17

PRIME MOVERS

17.1 INTRODUCTION

If a pressure difference is required between two points in a system, a prime mover such as a fan, pump, or compressor is usually used to provide the necessary pressure and/or flow impetus. Engineers are often called on to specify prime movers more frequently than any other piece of processing equipment, particularly in the chemical industry. In a general sense, these prime movers are to a process plant what the engine is to one's automobile. Whether one is processing petrochemicals, caustic soda, acids, and so on, the fluid must usually be transferred from one point to another somewhere in the process. At a chemical plant, chemicals must be loaded or unloaded, sent to heat exchangers or cooling coils, transferred from one processing unit to another, or packaged for shipment.

To move material (either a fluid or slurry) through the various pieces of equipment at a facility (including piping and duct work) requires mechanical energy, not only to impart an initial velocity to the material, but more importantly, to overcome pressure losses that occur throughout the flow path. This energy may be imparted to the moving stream in one or more of three modes: an increase in the stream's velocity, an increase in the stream pressure, or an increase in stream height. In the first case, the additional energy takes the form of increased external kinetic energy as the bulk stream velocity increases. In the second, the internal energy (mainly potential energy, but usually some kinetic as well) of the stream increases. This pressure increase may also cause a stream temperature rise, which represents an internal

Fluid Flow for the Practicing Chemical Engineer. By J. Patrick Abulencia and Louis Theodore

energy increase. In the third case, which may be relatively small for some operations, the bulk fluid experiences an increase in external potential energy in the Earth's gravitational field.

Three devices which convert electrical energy into the mechanical energy that is to be applied to various streams are discussed in this chapter. These devices are fans, which move low pressure gases; pumps, which move liquids and liquid–solid mixtures such as slurries, suspensions, and sludges; and, compressors, which move high pressure gases.

There are three general process classifications of prime movers—centrifugal, rotary, and reciprocating—that can be selected. Except for special applications, centrifugal units are normally employed. Basically, a centrifugal unit consists of an impeller, which is a series of radial-vanes of various shapes and curvatures, spinning in a circular casing. Fluid enters the "eye" or axis of rotation and discharges radially into a peripheral chamber at a higher pressure that corresponds to the sum of the centrifugal force of rotation and the kinetic energy given to the fluid by the turning vanes. The only moving part in the unit is the impeller. The vanes of the impeller extend from the center of rotation to the periphery and the shrouds are the disks on each side of the vanes enclosing them. The vanes may be radial, may curve slightly "forward" (in the same direction as that of rotation), or may curve "backward," which is the usual case.

As indicated above, there are three classes of prime movers—fans, pumps, and compressors. The three sections that follow will treat each individually. These units are normally rated in terms of four characteristics:

1. Capacity: the quantity of fluid discharged per unit time.
2. Increase in pressure, often reported for pumps as head: head can be expressed as the energy supplied to the fluid per unit weight and is obtained by dividing the increase in pressure by the fluid density.
3. Power: the energy consumed by the mover per unit time.
4. Efficiency: the energy supplied to the fluid divided by the energy supplied to the unit.

The net effect of most prime movers is to increase the pressure of the fluid. However, as described earlier, some provide the fluid with an increase in kinetic energy (velocity) or an increase in potential energy (elevation) or both.

Noise generation is a problem common to all these devices. The noise level is strongly related to the rotative speed of the unit. Noise control can include:

1. Isolation.
2. Proper maintenance.
3. Encapsulation.
4. Piping insulation (to and from the unit).

Additional information is provided in the following section "Fans".

17.2 FANS

The term *fans* and *blowers* are often used interchangeably, and no distinction will be made between the two in the following discussion. Whatever is stated about fans applies equally to blowers. Strictly speaking, however, fans are used for low pressure (drop) operation, generally below 2 psi. Blowers are generally employed when generating pressure heads in the 2.0–14.7-psi range. Higher pressure operations require compressors.

Fans are usually classified as centrifugal or axial-flow type. In centrifugal fans (as noted earlier), the gas is introduced into the center of the revolving wheel (the eye) and discharges at right angles to the rotating blades. In axial-flow fans, the gas moves directly (forward) through the axis of rotation of the fan blades. Both types are used in industry, but it is the centrifugal fan that is employed at most facilities.

The gas in a centrifugal fan is subjected to the same centrifugal forces described earlier. These forces compress the gas giving it additional static pressure. Centrifugal fans are enclosed in a scroll-shaped housing that helps convert kinetic energy to static pressure. Gas rotating between the fan blades is compressed in the fan scroll, which increases the static pressure. Centrifugal fans are classified by blade configuration as not only radial, forward curved, backward curved, but also as air foil and radial tip-forward curved heel.

Radial or straight blade fans physically resemble a paddle wheel with long radial blades attached to the rotor and are the simplest design of all centrifugal fans. This enables most radial blade fans to be built with great mechanical strength and to be easily repaired. These fans can be used in a variety of situations, especially heavy duty applications. This type of fan can handle erosive and corrosive gases as well as very viscous gases. It is particularly well-suited for high static pressure operations and can generate pressures in excess of 50 in. H_2O. When operated properly, the horse-power efficiency range is 55–69%, with 65% as a typical value. *Forward curved* fans are the most popular for general ventilation purposes (high flow rates and low static pressures). These fans have both the heel and the tip of the blade curved forward in the direction of rotation. Blades are smaller and spaced much closer together than in other blade designs. They are generally not used with dirty gases when dust or sticky materials are present because contaminants easily accumulate on the blades and cause imbalance. Efficiencies range from 52–71%, with 65% being typical. *Backward curved* or backward inclined fans have blades inclined in a direction opposite to that of the direction of rotation. This feature causes the gas to leave the tip of the blade at a lower velocity than the wheel-tip speed, a factor that improves the mechanical efficiency. These types of fans are not suitable for a heavily particulate-laden gas, sticky material, or abrasive dust. Centrifugal forces tend to build up particulate matter on the backside of the fan blades. The airfoil is similar to the backward curved fan, except that the blade has been contoured to increase stability and operating efficiency. These fans are more expensive to construct than backward curved fans, but have lower power requirements. They are rarely used in air pollution control where the gas must be clean and noncorrosive. Since the blades are hollow, abrasion and wear could allow dust, water vapor, and so on, to enter the blade and cause imbalance. This is the most efficient of the various fan types, with typical efficiencies at the 85% level. A modification of the radial fan is the *radial tip-forward curved heel*. The blades are curved forward with this unit. This fan is reportedly more dependable than the radial or high tip speed applications (i.e., high static pressures) due to better vibrational characteristics and its ability to resist fatigue. It finds application in the processing of large flow rates (>200,000 acfm) with light to medium particulate loadings. However, sticky material and particulates, in general, can accumulate in the slight curvature of the blades and cause imbalance. Efficiencies here typically range from 52–74%, with 70% being common.

Generally, centrifugal fans are easier to control, more robust in construction, and less noisy than axial units. They have a broader operating range at their highest efficiencies. Centrifugal fans are better suited for operations in which there are flow variations and they can handle dust and fumes better than axial fans.

Fan laws are equations that enable the results of a fan test (or operation) at one set of conditions to be used to calculate the performance at another set of conditions, including differently sized but geometrically similar models of the same fan design. The fan laws can be written in many different ways. The three key laws are provided in the following equations:

$$q_a = k_1(\text{rpm})D^3 \tag{17.1}$$

$$P_s = k_2(\text{rpm})^2 D^2 \rho \tag{17.2}$$

$$\text{hp} = k_3(\text{rpm})^3 D^5 \rho \tag{17.3}$$

where q_a = volumetric flow rate
P_s = static pressure
hp = horsepower
rpm = revolutions per minute
D = wheel diameter
ρ = gas density
k_1, k_2, k_3 = proportionality constants

Thus, these three laws may be used to determine the effect of fan speed, fan size, and gas density on flow rate, developed static pressure head, and horsepower. For two conditions, where the constants k remain unchanged, Equations (17.1) and (17.3) become:

$$\left(\frac{q_a'}{q_a}\right) = \left(\frac{\text{rpm}'}{\text{rpm}}\right)\left(\frac{D'}{D}\right)^3 \tag{17.4}$$

$$\left(\frac{P_s'}{P_s}\right) = \left(\frac{\text{rpm}'}{\text{rpm}}\right)^2\left(\frac{D'}{D}\right)^2\left(\frac{\rho'}{\rho}\right) \tag{17.5}$$

$$\left(\frac{\text{hp}'}{\text{hp}}\right) = \left(\frac{\text{rpm}'}{\text{rpm}}\right)^3\left(\frac{D'}{D}\right)^5\left(\frac{\rho'}{\rho}\right) \tag{17.6}$$

Note: The *prime* refers to the new condition. It is also important to note that the fan laws are approximations and should not be used over wide ranges or changes of flow rate, size, etc.

Illustrative Example 17.1 Fan A has a blade diameter of 46 inches. It is operating at about 1575 rpm while transporting 16,240 acfm of flue gas and requires 47.5 brake horse power (bhp). Fan B is to replace Fan A. It is to operate at 1625 rpm with a blade diameter of 42 inches and is of the same homologous series as Fan A. What is the power requirement of Fan B?

Solution The power requirement for Fan B is calculated, using the fan law, Equation (17.6):

$$\frac{\text{hp}_\text{B}}{\text{hp}_\text{A}} = \left(\frac{\text{rpm}_\text{B}}{\text{rpm}_\text{A}}\right)^3 \left(\frac{D_\text{B}}{D_\text{A}}\right)^5 \left(\frac{\rho_\text{B}}{\rho_\text{A}}\right) = \left(\frac{1625}{1575}\right)^3 \left(\frac{42}{46}\right)^5 (1) = 0.697$$

$$\begin{aligned} \text{hp}_\text{B} &= (0.697)(47.5) \\ &= 33.1 \text{ bhp} \end{aligned}$$

Illustrative Example 17.2 A fan operating at a speed of 1694 rpm delivers 12,200 acfm of flue gas at 5.0 in. H_2O static pressure and requires 9.25 bhp. What will be the new operating conditions if the fan speed is increased to 2100 rpm?

Solution The new fan flow rate (superscript prime) is calculated using Equation (17.4):

$$\begin{aligned} q_a' &= q_a \frac{\text{rpm}'}{\text{rpm}} \left(\frac{D'}{D}\right)^3 = (12{,}200)\frac{2100}{1694}(1) \\ &= 15{,}124 \text{ acfm} \end{aligned}$$

Using Equation (17.5), the new static pressure is calculated:

$$\begin{aligned} P_s' &= P_s \left(\frac{\text{rpm}'}{\text{rpm}}\right)^2 \left(\frac{D'}{D}\right)^2 \left(\frac{\rho'}{\rho}\right) = 5.0\left(\frac{2100}{1694}\right)^2 (1)(1) \\ &= 7.68 \text{ in. } H_2O \end{aligned}$$

The required horsepower is calculated using Equation (17.6):

$$\begin{aligned} \text{hp}' &= \text{hp}\left(\frac{\text{rpm}'}{\text{rpm}}\right)^3 \left(\frac{D'}{D}\right)^5 \left(\frac{\rho'}{\rho}\right) = 9.25\left(\frac{2100}{1694}\right)^3 (1)(1) \\ &= 17.62 \text{ bhp} \end{aligned}$$

A rigorous, extensive treatment on fan selection is beyond the scope of this text. It is common practice among fan vendors to publish voluminous data in tabular form providing flow rate, static pressure, speed, and horsepower at a standard temperature and gas density. These are often referred to as *multirating tables.*

Note: These tables should not be used for fan selection except by those who have experience in this area. For those who do not, the proper course of action to follow is to provide the fan manufacturer with a complete description of the system and allow the manufacturer to select and guarantee the optimum fan choice.

To help in the actual selection of fan size, a typical fan rating table is given in Table 17.1. The fan size and dimensions are usually listed at the top of the table. Values of static pressure are arranged as columns that contain the fan speed and brake horsepower required to produce various volume flows. The point of maximum efficiency at each static pressure is usually underlined or printed in special type. In order to select a fan for the exact condition desired, it is sometimes necessary to interpolate between values presented in the multirating tables. Straight-line interpolation can be used with negligible error for multirating tables based on a single fan size. Some multirating tables attempt to show ratings for a whole series of geometrically similar (homologous) fans in one table; in this case, interpolation is not advised.[1]

The selection procedure is, in part, an examination of the fan curve and the system curve. A fan curve, relating static pressure with flow rate, is provided in Fig. 17.1. Note that each type of fan has its own characteristic curve. Also note that fans are usually tested in the factory or laboratory with open inlets and long smooth straight discharge ducts. Since these conditions are seldom duplicated in the field, actual operation often results in lower efficiency and reduced performance. A system curve is also shown in Fig. 17.1. This curve is calculated prior to the purchase of the fan and provides a best estimate of the pressure drop across the system through which the fan must deliver the gas. (This curve should approach a straight line with an approximate slope of 1.8 on log–log coordinates.) The system pressure (drop) is defined as the resistance through ducts, fittings, equipment, contractions, expansions, etc.

A number of methods are available to estimate the total system pressure change. These vary from very crude approximations to detailed, rigorous calculations. The simplest procedure is to obtain estimates of the pressure change associated with the movement of the gas through all of the resistances described previously. The sum of these pressure changes represents the total pressure drop across the system, and represents the total pressure (change) that must be developed by the fan. This calculation becomes more complex if branches (i.e., combining flows) are involved. However, the same stepwise procedure should be employed.

When the two curves are superimposed, the intersection is defined as the *point of operation*. The fan should be selected so that it operates just to the right of the peak on the fan curve. The fan operates most efficiently and with maximum stability at this condition. If a fan is selected for operation too close to its peak, it will surge and oscillate. Thus, the point of intersection of the two curves determines the actual volumetric flow rate. If the system resistance has been accurately specified

Table 17.1 Typical fan rating table[a–e]

	Static Pressure (in. H_2O)											
	$\frac{1}{2}$		1		$1\frac{1}{2}$		2		$2\frac{1}{2}$		3	
q_a (acfm)	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
5727	216	0.74	278	1.34	330	2.01	378	2.75	421	3.50	460	4.28
6873	236	0.97	291	1.67	339	2.42	384	3.24	424	4.06	462	4.91
8018	250	1.27	305	2.05	352	2.87	393	3.76	430	4.69	467	5.66
9164	271	1.63	320	2.49	366	3.42	405	4.35	441	5.36	475	6.40
10,309	293	2.12	338	3.05	381	4.06	419	5.06	453	6.14	486	7.26
11,455	315	2.72	356	3.65	396	4.76	432	5.88	468	7.03	499	8.19
12,600	337	3.46	377	4.39	413	5.58	448	6.81	482	8.04	514	9.31
13,746	360	4.39	399	5.25	430	6.48	465	7.82	496	9.16	527	10.53
14,891	382	5.43	421	6.25	451	7.52	481	8.93	512	10.35	542	11.80
16,037	405	6.66	442	7.48	473	8.67	501	10.16	529	11.69	557	13.25
17,182	429	8.08	463	8.97	496	10.05	521	11.54	547	13.18	574	14.81
18,328	451	9.64	486	10.61	517	11.61	543	12.99	566	14.78	591	16.49
19,473	474	11.46	510	12.47	539	13.47	565	14.85	587	16.56	610	18.35
20,619	497	13.51	532	14.55	560	15.60	587	16.82	610	18.54	630	20.40
21,764	520	15.82	556	16.82	584	17.98	608	19.17	632	20.77	652	22.59

Source: Bayler Blower Co.
[a]Wheel style: backward inclined
[b]Wheel diameter: 50 $\frac{1}{2}$ inches
[c]Maximum fan speed: 1134 rpm
[d]Performances underlined are those at maximum efficiency
[e]Brake horsepower = bhp

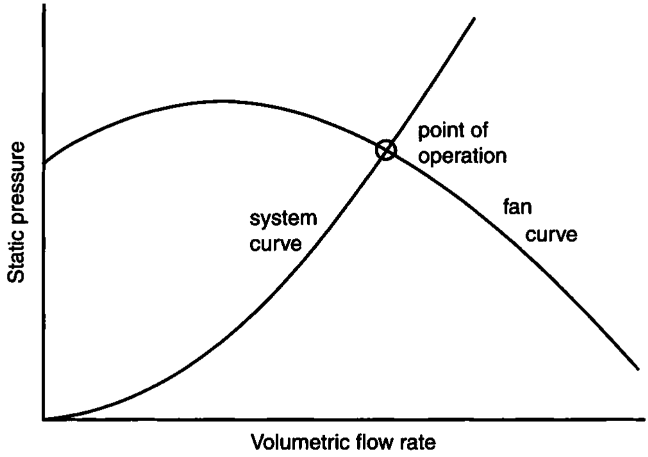

Figure 17.1 System and fan characteristic curves.

and the fan properly selected, the two performance curves will intersect at the design flow rate. If system pressure losses have not been accurately specified or if undesirable inlet and outlet conditions exist, design conditions will not be obtained. Dampers and fan speed changes can provide some variability on operating conditions.

There are a number of process and equipment variables that are classified as part of a fan specification. These include:

Flow rate (acfm)
Temperature
Density
Gas stream characteristics
Static pressure that needs to be developed
Motor type
Drive type
Materials of construction
Fan location
Noise controls (briefly discussed earlier)

With respect to drive type, belt drives are usually employed if the power is <200 hp. Direct drives are preferred for large horsepower systems. Direct drive units have lower maintenance costs and lower power transmission losses. However, the speed cannot be varied; if a speed change is required, the cost can be expensive. For materials of construction, mild carbon steel is commonly employed when treating dry air up to temperatures approaching 1000°F. Fiberglass may be used for corrosive conditions but the temperature should not exceed 250°F.

With respect to fan location, the fan may be located downstream or upstream of a particular piece of equipment in the process. Fans located downstream are referred to as *induced draft* or *negative pressure* fans. Leakage occurs into, rather than out of, the unit. Possible lower flow rates and lower temperatures may lead to a smaller fan and reduced operating costs. Equipment costs could be higher because of the need for heavier construction material when operating under negative (vacuum) pressure. Fans located upstream may have erosion problems due to possible particulate loading; in addition, there may be an accumulation of particulates. A larger volumetric flow rate is possible, which would require a larger fan and high horsepower costs.

With respect to noise controls, fans can create objectionable noise problems in the work area or neighboring residences. To minimize noise effects, the fan should, of course, be properly designed and properly operated. A fan should therefore be operated close to the point of maximum efficiency to reduce noise levels. Putting a fan in proper balance can also be an effective solution. If noise problems are expected or are already present, acoustical insulation should be applied to the fan housing. If the noise level is severe, insulation should also be added to the ductwork.(2)

Gas (or air) horsepower (ghp) and brake horsepower (bhp) are the two terms of interest. These may be calculated from Equations (17.7) and (17.8):

$$\text{ghp} = 0.0001575 q_a \Delta P = q_a \Delta P / 6356 \tag{17.7}$$

$$\text{bhp} = 0.0001575 q_a \Delta P / \eta_f \tag{17.8}$$

where q_a = volumetric flow rate (acfm)
ΔP = SP = static pressure head developed (in. H_2O)
η_f = fractional fan efficiency

For cold start-ups, the static pressure can be significantly higher than the normal operating pressure drop, and the fan must be able to handle the increased resistance. The correction for the cold static pressure may be calculated from Equation (17.9):

$$\Delta P_c = \Delta P(\rho_c / \rho) \tag{17.9}$$

where ρ = gas density (lb/ft^3)
c = subscript denoting cold gas

Illustrative Example 17.3 Calculate the hp required to process a 6500-acfm gas stream from a process. The pressure drop across various pieces of equipment has been estimated to be 6.4 in. H_2O. The pressure loss for duct work, elbows, valves, expansion–contraction losses, etc., are estimated at 4.4 in. H_2O. Assume an overall fan-motor efficiency of 63%.

Solution The total pressure drop, ΔP (in. H_2O), is

$$\Delta P = (6.4 + 4.4) = 10.8 \text{ in. } H_2O$$

The brake horsepower required is calculated by Equation (17.7).

$$\text{bhp} = \frac{(1.575 \times 10^{-4})q_0\Delta P}{\eta_f} = \frac{(1.575 \times 10^{-4})(6500)(10.8)}{(0.63)}$$
$$= 17.55\,\text{bhp}$$

Note that the term 1.575×10^{-4} is a conversion factor to obtain units of horsepower.

17.3 PUMPS

Pumps are required to transport liquids, liquid–solid mixtures such as slurries and sludges, auxiliary fuel, etc. Pumps are also needed to transport water to and/or from such peripheral devices as boilers, quenchers, scrubbers, and so on.

As indicated earlier, pumps may be classified as reciprocating, rotary, or centrifugal. The reciprocating and rotary types are referred to as *positive displacement* pumps because unlike the centrifugal type, the liquid or semiliquid flow is broken up into small portions as it passes through the pump. These three classes of units are described below.

Reciprocating pumps operate by the direct action of a piston on the liquid contained in a cylinder. As the liquid is compressed by the piston, the higher pressure forces it through discharge valves to the pump outlet. As the piston retracts, the next batch of low pressure liquid is drawn into the cylinder and the cycle is repeated. The piston may be either directly steam driven or moved by a rotating crankshaft through a crosshead. The rate of liquid delivery is a function of the volume swept out by the piston and the number of strokes per unit time. A fixed volume is delivered for each stroke but the actual delivery may be less because of both leakage past the piston and failure to fill the cylinder when the piston retracts. The volumetric efficiency of the pump is defined as the ratio of the actual volumetric discharge to the pump displacement. For well maintained pumps, the volumetric efficiency is at least 95%. Reciprocating pumps are used for some applications.

Reciprocating pumps can deliver the highest pressure of any type of pump (20,000 psig); however, their capacities are relatively small compared to the centrifugal pump. Also, because of the nature of the operation of the reciprocating pump, the discharge flow rate tends to be somewhat pulsating. Liquids containing abrasive solids can damage the machined surfaces of the piston and cylinder. Because of its positive displacement operation, reciprocating pumps can be used to measure liquid volumetric flow rates.

The *rotary* pump combines rotation of the liquid with positive displacement. The rotating elements mesh with elements of the stationary casing in much the same way that two gears mesh. As the rotating elements come together, a pocket is created that first enlarges, drawing in liquid from the inlet or suction line. As rotation continues, the pocket of liquid is trapped, reduced in volume, and then forced into the discharge line at a higher pressure.

The flow rate of liquid from a rotary pump is a function of size and speed of rotation and is slightly dependent on the discharge pressure. Unlike reciprocating pumps, rotary pumps deliver nearly constant flow rates. Rotary pumps are used on liquids of almost any viscosity as long as the liquids do not contain abrasive solids. For this reason, they are very effective with many high viscosity mixtures. They operate in moderate pressure ranges (5000 psig), have small-to-medium capacities, and like the reciprocating pump, can be used for metering liquids.

Centrifugal pumps are the most widely used in the process industry because of simplicity of design, low initial cost, low maintenance, and flexibility of application. Centrifugal pumps have been built to move as little as a few gallons per minute against a small head, and as much as several thousand gallons per minute against a pressure of several hundred pounds force per square inch (psi). In its simplest form, this type of pump consists of an impeller rotating within a casing. Fluid enters the pump near the center of the rotating impeller and is thrown outwards by centrifugal force. The kinetic energy of the fluid increases from the center of the impeller to the tips of the impeller vanes. This high velocity is converted to a high pressure as the fast-moving fluid leaves the impeller and is driven into slower moving fluid in the volute or diffuser.[3]

Not all centrifugal pumps produce the radial flow (directed away from the axis of rotation) described previously; many produce an axial flow (directed along the axis of rotation) and others a combination of the two. The turbine type of centrifugal pump has smaller, straighter vanes, is driven at high speed, and generates a highly radial flow to produce higher pressures at lower flow rates. The axial-flow type employs multibladed propellers that generate a highly axial flow, resulting in large flow rates at lower pressures. For a specific balance between flow rate and pressure, impellers can be shaped to provide results between that of the turbine and axial-flow type; these are referred to as *mixed-flow* impellers.

The impeller is the heart of the centrifugal pump. It consists of a number of curved vanes or blades that are shaped in such a way as to give smooth fluid flow between the blades. In the *straight-vane, single-suction closed* impeller, the surfaces of the vanes are defined by straight lines parallel to the axis of rotation. The *double-suction* impeller is, in fact, two single-suction impellers arranged back-to-back in a single casing. Centrifugal pump casings may be of several designs but their main function is to convert kinetic energy imparted to the fluid by the impeller into a higher pressure. In addition, the casing provides an inlet and an outlet for the pump and contains the fluid. Casings may be either the *volute* or the *diffuser* type. The *volute* type has a continuous flow area that allows the velocity to decrease gradually, thereby reducing eddy formation; this minimizes the loss of energy due to turbulence. The *diffuser*-type casing has stationary guides that offers the liquid a widening path from impeller to casing; this also keeps turbulence to a minimum.[3]

In pumping fluids at a facility, the operation of a pump may result in either the loss of liquid or the release of air contaminants, or both. The reciprocating and centrifugal pumps can also be sources of hazardous emissions. The opening in the cylinder through which the connecting rod drives the piston is the major source of contaminant release from a reciprocating pump. In centrifual pumps, normally the only potential source of leakage occurs where the drive shaft passes through the casing.

Several methods have been devised for sealing the clearance between the pump shaft and fluid casing. For most applications, packed seals and mechanical seals are widely used. Packed seals can be used on both positive displacement and centrifugal pumps. A typical packed seal consists of a stuffing box filled with a sealing material or packing that encases the moving shaft. The stuffing box is fitted with a take-up ring that compresses the packing and causes it to tighten around the shaft. Materials used for packing vary with the fluid's temperature, physical and chemical properties, pressure, and pump type. Some commonly used materials are metal, rubber, leather, and plastics. For cases where the use of mechanical seals is not feasible, specialized pumps such as canned-motor, diaphragm, or electromagnetic pumps are used. These specialty pumps are used where no leakage can be tolerated and are available in a limited range of sizes; most are for low flow rates and all are of single- or two-stage construction. They have been used to handle both high temperature and very low temperature liquids. These pumps follow the same hydraulic principles as the traditional centrifugal pump. Because of their small size, they operate with rather low efficiencies, but in dangerous applications, efficiency must be sacrificed for safety.[4]

Illustrative Example 17.4 A pump produces 25 kPa pressure when a valve is closed to shut down the flow completely, and 5 kPa when the valve is opened to allow a flow rate of 2 m^3/s. Assuming the pressure-flow rate curve is parabolic, what is the pressure when the flow rate is 1 m^3/s?

Solution The parabolic pump pressure-flow rate curve can be written as:

$$P = a - bq^2$$

Enter the given data to obtain two equations for the constants a and b.

$$25 = a - b(0)$$

$$5 = a - b(2)^2$$

Solve the equations simultaneously for a and b.

$$a = 25$$

$$5 = a - 4b$$

$$b = \frac{a-5}{4} = \frac{25-5}{4} = 5$$

Write the equation for the pressure P with the values for the constants.

$$P = 25 - 5q^2$$

Substitute the known value for q and compute P.

$$P = 25 - 5(1)^2$$
$$= 20\,\text{kPa}$$

Pumps are employed in industry for one and/or a combination of the following reasons:

1. The transportation of liquids from trucks or delivery vehicles to storage tanks.
2. The transportation of liquids from storage tanks to process units.
3. The transportation of slurries and/or sludges to/from storage tanks to/from process units.
4. The transportation of liquids from one process unit to another.
5. The delivery of water to the quench units and heat boilers (where applicable).
6. The delivery and circulation of water and/or caustic solutions in absorbers and scrubbers (where applicable).

Pump discharge pressures for (2) and (3) are roughly 75 and 50 psig, respectively. Actual discharge pressures will vary, however, with fluid viscosity and line pressure drop (which is a function of site-specific logistics).

A general relationship that may be used to determine pump hp requirements is[4]

$$\text{hp} = \frac{7.27 \times 10^{-5}\dot{m}\Delta P}{\rho \times \eta_p} \tag{17.10}$$

where $\dot{m}$ is the liquid flow rate, ΔP is the developed pressure drop, ρ is the liquid density and η_p is the pump-motor efficiency (on a fraction basis).

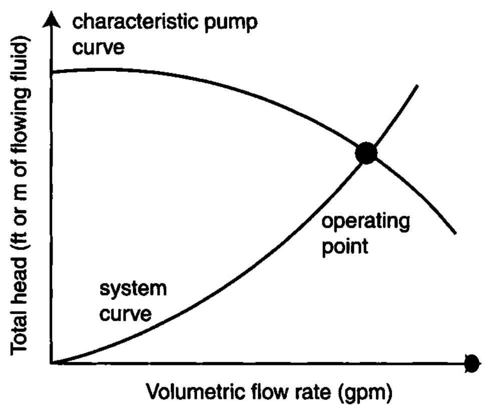

Figure 17.2 Pump characteristic curve.

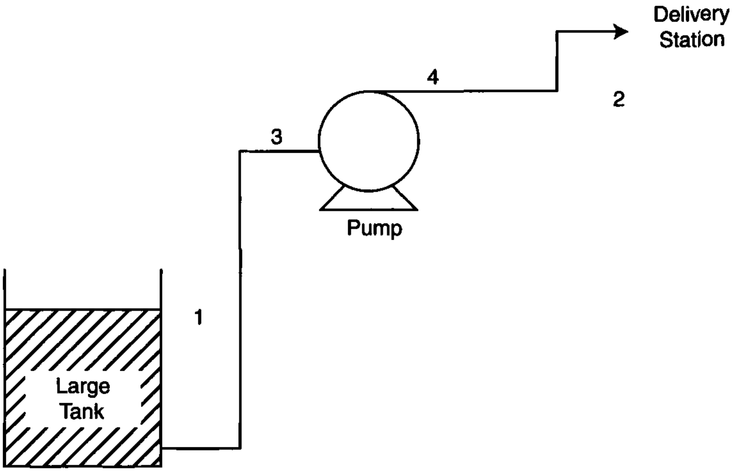

Figure 17.3 Pump system.

The pressure increase (head) developed by a centrifugal pump is generally a function of the discharge rate. In a real pump, as the flow rate becomes large, the total head delivered by the pump decreases. To obtain the discharge rate for a given system, the engineer has to solve, simultaneously, the system curve of head, h_p, versus discharge with the pump curve head, h_c. The system h_p is usually obtained by applying Bernoulli's equation (or the equivalent). The pump characteristic curve of h_c is usually supplied by the manufacturer. The point where the two curves intersect (as indicated earlier—see Fig. 17.1) yields the operating conditions for the system, as shown in Fig. 17.2.

Consider the flow of fluid in Fig. 17.3. The liquid is pumped from the large tank (point 1) to the delivery station (point 2). Applying a modified form of Bernoulli's equation to the liquid system between stations 1 and 2 gives:

$$\frac{P_1}{\rho} + \frac{v_1^2}{2g_c} + z_1 \frac{g}{g_c} = \frac{P_2}{\rho} + \frac{v_2^2}{2g_c} + z_2 \frac{g}{g_c} - h_s + h_f$$

or

$$h_s = \frac{P_2 - P_1}{\rho} + \frac{v_2^2 - v_1^2}{2g_c} + (z_2 - z_1)\frac{g}{g_c} + h_f \qquad (17.11)$$

where h_s is the input work (pump head) and h_f is the head loss due to fluid friction, with both treated as positive numbers. The units of each term as written in the above equation are energy/mass.

For any flow rate, the pump head required to maintain that flow is calculated from Equation (17.11). In this procedure, one constructs a head-versus-discharge curve. This curve of h_p vs q is termed the system curve. For a given centrifugal pump, the pump manufacturer supplies a characteristic head-versus-discharge curve. Once again, where the two curves intersect is the operating point of the pump.

When a centrifugal pump is operating at high flow rates, the high velocities occurring at certain points in the eye of the impeller or at the vane tips cause local pressures to fall below the vapor pressure of the liquid. This is predicted by the Bernoilli equation. Vaporization occurs at these points, forming bubbles that collapse violently upon moving to a region of higher pressure or lower velocity. This momentary vaporization and destructive collapse of the bubbles is called cavitation. This should be avoided if maximum capacity is to be obtained and damage to the pump prevented. The shock of bubble collapse causes severe pitting of the impeller and creates considerable noise and vibration. Cavitation may be reduced or eliminated by reducing the pumping rate or by slight alterations in the impeller design to give better streamlining. Note, however, that cavitation usually does not occur at low flow rates on any given pump.(5)

Both vapor binding and cavitation may be eliminated by maintaining a pressure at the pump inlet that is significantly higher than the vapor pressure of the liquid being pumped. The required margin of pressure is called the net positive suction head (NPSH). It is a function of the pump design and is usually specified by the manufacturer for pumps that handle liquids such as preheated boiler feed, steam condensate, or volatile liquids. Usually, the NPSH is of the order of 2 to 10 ft-lb_f/lb of fluid and increases with increasing throughput.(5)

From an engineering point-of-view, the question that needs answering is will the pump work or will it cavitate. The NPSH determines the answer. Cavitation will not occur if the sum of the pressure plus the velocity and dynamic heads in the suction line is greater than the vapor pressure head. The NPSH is therefore defined as:

$$\text{NPSH} = \frac{P_i}{\rho}\frac{g_c}{g} + \frac{v_i^2}{2g} - \frac{p'}{\rho}\frac{g_c}{g} \tag{17.12}$$

where P_i and v_i are the pressure and velocity at the pump inlet and p' is the liquid vapor pressure. If the pump inlet is at a height, z_i, and if the reservoir free surface is at pressure, P_{ac}, and height, z_a, one may use Bernoulli's equation to rewrite NPSH as:

$$\frac{P_a}{\rho}\frac{g_c}{g} + z_a = \frac{P_i}{\rho}\frac{g_c}{g} + \frac{v_i^2}{2g} + z_i + h_{fi}' \tag{17.13}$$

where h_{fi}' is the friction head loss between the reservoir and the pump inlet, with the prime representing units of height of flowing fluid. Substituting from Equation (17.13) into Equation (17.12) gives:

$$\text{NPSH} = \left(\frac{P_a - p'}{\rho}\right)\frac{g_c}{g} + (z_a - z_i) - h_{fi}' \tag{17.14}$$

Pump manufacturers will usually specify the NPSH requirement of the pump. The reader should note that the units change from energy/mass to height of flowing fluid by multiplying each of the terms in the energy equations by (g/g_c).

The sensitivity of a device to cavitation occurrence is measured by the dimensionless cavitation number, Ca. It is the ratio of available pressure above the vapor pressure to the *dynamic pressure* of the flow. When a fluid moves at a velocity, v, its dynamic pressure, P_{dyn}, is defined as the kinetic energy of flow per unit volumetric flow rate of the fluid, i.e.,

$$P_{\text{dyn}} = \frac{\rho v^2}{2g_c} \tag{17.15}$$

Therefore, the cavitation number, Ca, is expressed by the following equation:

$$\text{Ca} = \frac{P_a - p'}{P_{\text{dyn}}} = \frac{P_a - p'}{\rho v^2/2g_c} \tag{17.16}$$

where P_a is the ambient pressure, p' is the vapor pressure, v is the fluid velocity, and ρ is the fluid density. In some liquid flow applications, a pump is placed above the liquid being pumped. The pump has to draw in the liquid in order to pump it out. It is highly undesirable to have the pump drawing vapor rather than the liquid. An example is the use of a pump to draw water from a well. Clearly, the vapor pressure of a liquid limits the maximum height above a sump at which a pump may be located.

There are seven items of process data that must be specified for pumps.[6] The first four are basic, and can be obtained from the process flowsheet (which should include a heat and material balance) or from tables of physical properties. The basic four process items are:

1. Fluid being pumped
2. Operating temperature
3. Specific gravity, both at 60°F and at operating conditions
4. Viscosity at operating conditions—this is particularly important for positive displacement pumps

The next three items usually require calculation. These are:

5. Capacity
6. Head (or pressure)
7. Net Positive Suction Head (NPSH)

Centrifugal pumps are very widely used in industry. The use of a pump is required any time a liquid needs to overcome pressure or static constraints. Pump applications range from the small-scale every-day uses as those in household piping systems to large machinery, plants, and pipelines. Pumps can be found as integral parts of systems in laboratories, automobiles, and anything else that necessitates a liquid flow system.

Centrifugal pumps are dependable and long-lasting. They can tolerate internal corrosion and erosion without substantially decreasing performance. Flexible operation provides good flow-control characteristics over a wide capacity range at constant speed. Capacities can be conveniently varied by throttling the discharge. These pumps are usually quiet, need little attention, and operate pulsation free. Due to the long life of the pumps, maintenance costs are also low. They can be easily disassembled, few parts have close tolerances, and worn parts can be quickly replaced. Centrifugal pumps pay for themselves in a short time. The low investment cost is due to simple designs, direct-coupled motor arrangements, and a wide range of material, size, performance and operating characteristics. Such pumps occupy small space without shelter. Piping arrangements are usually simple.

Pumps rarely influence plant layout except where a common standby for two services might require the rearrangement of process equipment. Pumps are generally placed close to process vessels and arranged esthetically.

On the negative side, capacity, head, and efficiency rapidly decrease with increasing viscosity. In addition, the centrifugal pump normally cannot transfer liquid having vapor content. At low flows (below 15–20% of design capacity), the centrifugal pump can become unstable. Thus, a minimum flow is often specified.

Pumps are selected by specialists and the piping designer has little influence on the basic selection. However, the layout designer can request preferred orientation for suction and discharge and the NPSH limitations (to meet required equipment elevations).[7]

17.3.1 Parallel Pumps

Pumps can also be set up in parallel. This configuration is used to administer systems where higher capacity is desired. Pumps in parallel work as two single pumps with merging outlets. In addition to creating greater capacity, pumps in parallel can be helpful for systems that cannot be shut down for long periods of time, as each pump works independently of the other.

The calculations used for pumps in parallel are much like those used for pumps in series. The capacity and volumetric flow rate of the pair of pumps are found in the same way as that of the single pump and series pumps. The total head of the system is simply the greater head produced from each of the two pumps; these values are calculated as the head is calculated for a single pump. The power and total efficiency are calculated in the same way as those for two pumps in series.

Illustrative Example 17.5 It is necessary to deliver a liquid with density and viscosity similar to that of water to a process unit at a rate of 80 gal/min. The pump must operate against a pressure of 180 psi. An external gear pump with characteristics shown in Fig. 17.4 is available with variable speed drive. At what speed should the pump be operated? What horsepower is needed to maintain flow?

Solution Plotting this information on Fig. 17.4 indicates that the speed is between the 400 and 600 rpm lines (point A). Interpolation gives a speed of about 425 rpm. Interpolating again on the hp curve gives a value of about 17 hp (point B). Using this data with Equation (17.11), an efficiency of 50% results in a pump with a horsepower of approximately 15 hp.

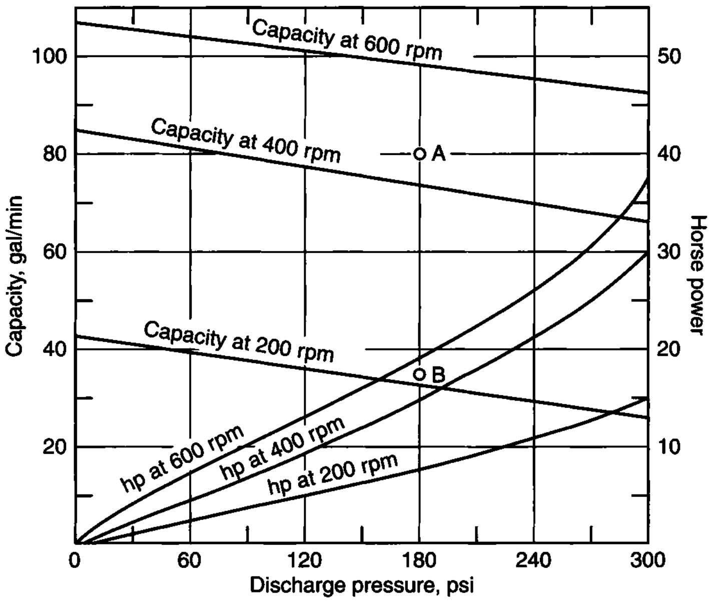

Figure 17.4 Performance characteristics of an external gear pump.

Illustrative Example 17.6 The characteristic curve, h'_c, of a centrifugal pump is approximated as

$$h'_c = 42 - 0.0047q^2$$

where h'_c is the total (or developed) head in ft of water and q is the liquid discharge in gpm. The pump efficiency, η, is about 60% and is independent of the discharge rate. The pump is to be used in a water flow system in which the pump head in feet of water is calculated as:

$$h'_p = 12 + 0.0198q^2$$

The density of water is 62.4 lb/ft^3. Determine the water flow rate, the total pump head, the ideal (theoretical or fluid) pump power and the brake horsepower of the pump.

Solution The equation to be solved for the flow rate is obtained by equating h_c and h_p

$$h'_c = h'_p$$

$$42 - 0.0047q^2 = 12 + 0.0198q^2$$

$$30 = 0.0245q^2$$

$$q = 35 \text{ gpm} = 0.078 \text{ cfs}$$

Calculate the pump head

$$h'_c = 42 - 0.0047(35)^2$$
$$= 36.24 \text{ ft of water}$$

Calculate the mass flow rate of water

$$\dot{m} = \rho q = 62.4(0.078)$$
$$= 4.87 \text{ lb/s}$$

The fluid power requirement, $\dot{W}_{s,id}$, may now be calculated

$$\dot{W}_{s,id} = \dot{m}h_c' = 4.87(36.24) = 176 \text{ lbf} \cdot \text{ft/s}$$
$$= 0.32 \text{ hp}$$

Finally, the brake horsepower, $\dot{W}_s$, or bhp is

$$\dot{W}_s = \text{bhp} = \frac{\dot{W}_{s,id}}{\eta_p} = \frac{0.32}{0.6}$$
$$= 0.53 \text{ hp}$$

Illustrative Example 17.7 As indicated in pump flow, the lowest pressure usually occurs at the pump inlet. For a pump handling a known liquid at a specified temperature, determine the inlet absolute pressure (in Pa, lb_f/ft^2, and psi) that is liable to cause cavitation, given that the fluid is water at 40°C and 1 atm.

Solution The vapor (or saturation) pressures of several common liquids at 20°C are listed in Table A.2 (in the Appendix). The vapor pressure of water at different temperatures is listed in Table A.4 in the Appendix.

Obtain the vapor pressure of water at 40°C from Table A.4.

$$p' = 7.375 \text{ kPa} = 0.0728 \text{ atm} = 1.069 \text{ psi} = 153.97 \text{ psf}$$
$$= 0.752 \text{ m of water} = 2.47 \text{ ft of water}$$

Noting that atmospheric pressure is equivalent to 33.91 ft of water, then the pump elevation should not exceed 33.91 ft. Since the water vapor pressure at 40°C is 2.47 ft of water, then the maximum elevation at which the pump may be placed is (33.91 − 2.47) = 31.44 ft (9.58 m); otherwise the pump will draw water vapor instead of liquid water. Therefore an inlet pressure greater than 31.44 ft of water is liable to cause cavitation.

17.4 COMPRESSORS

Compressors, unlike fans and pumps, find only limited and specialized application. They are primarily employed to increase the pressure of a fluid in some types of systems, e.g., when liquids are broken up into tiny droplets (atomized) before entering a unit. This can be accomplished through the use of a high pressure stream of air or steam that impinges on the liquid stream and atomizes it. The pressurizing of the air or steam is accomplished through the use of compressors. Compressors are also used for atomization on certain types of air pollution control devices such as venturi scrubbers, which depend on fine water droplets to remove particulates from the flue gas stream.

Compressors operate in a similar fashion to pumps and have the same classifications: rotary, reciprocating, and centrifugal. An obvious difference between the two operations is the large decrease in volume resulting from the compression of a gaseous stream compared to the negligible change in volume caused by the pumping of a liquid stream.

Centrifugal compressors are employed when large volumes of gases are to be handled at low to moderate pressure increases (0.5–50 psi). *Rotary* compressors have smaller capacities and can achieve discharge pressures up to 100 psi. *Reciprocating* compressors are the most common type used in industry and are capable of compressing small gas flows to as much as 3500 psig. With specially designed compressors, discharge pressures as high as 25,000 psig can be reached, but these devices are capable of handling only very small volumes, and do not work well for all gases. For the applications mentioned earlier, atomizing of liquids for combustion or of venturi scrubber water for gas cleaning, reciprocating compressors are normally used.

The following equation may be used to calculate compressor power requirements when the compressor operation is adiabatic and the gas (usually air) follows ideal gas behavior[(2)]

$$W_s = \left(\frac{\gamma RT}{\gamma - 1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right] \tag{17.17}$$

where W_s = compressor work required per lbmol of air
R = 1.987 Btu/(lbmol · °R)
T = air temperature at compressor inlet conditions (°R)
P_1, P_2 = air inlet and discharge pressures
γ = ratio of the heat capacity at constant pressure to that at constant volume—typically 1.3 for air.

Illustrative Example 17.8 Compressed air is to be employed in the nozzle to assist the atomization of a liquid. The air requirement for the nozzle is 7.5 lb/min at 40 psia. If atmospheric air is available at 60°F and 1.0 atm, calculate the power requirement.

Solution The ideal gas law will apply at these conditions. Set the coefficient γ for air equal to 1.3. The compressed energy requirement (delivered to the air) is given by Equation (17.17):

$$W_s = -\frac{\gamma RT_1}{\gamma - 1}\left[\left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma} - 1\right] = -\frac{(1.3)(1.987)(520)}{1.3 - 1}\left[\left(\frac{40}{14.7}\right)^{(1.3-1)/1.3} - 1\right]$$

$$= -1163\ \text{Btu/lbmol of air}$$

The power is

$$\text{hp} = -(1163)(7.5/29)(778)$$
$$= -234.000\ \text{ft-lb}_f/\text{min}$$

This result may be divided by 33,000 to yield a hp requirement of 7.1 hp. The power required has the opposite sign. The reader should also note that this represents the *minimum* power required to accomplish this job.

The temperature ratio across a compression stage is

$$T_2/T_1 = (P_2/P_1)^{(k-1)/k} \quad \text{adiabatic operation} \tag{17.18}$$

$$T_2/T_1 = (P_2/P_1)^{(N-1)/N} \quad \text{polytropic operation} \tag{17.19}$$

where $k =$ adiabatic exponent, C_p/C_v
$N =$ polytropic exponent, $(N-1)/N = (K-1)/KE_p$
$P_1, P_2 =$ suction, discharge pressures, psia
$T_1, T_2 =$ suction, discharge temperatures, °R
$E_p =$ polytropic efficiency, fraction

It is normally assumed that the usual centrifugal compressor is uncooled internally and thus follows a polytropic path. The temperature of the gas is often limited to protect against temperature excursions. Intercooling can be employed to retain temperatures at reasonable levels during high overall compression ratio applications.

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