CHAPTER

14

TURBOMACHINERY

In this chapter we discuss the basic principles of a common and important application of fluid mechanics, *turbomachinery*. First we classify turbomachines into two broad categories, *pumps* and *turbines*. Then we discuss both of these turbomachines in more detail, mostly qualitatively, explaining the basic principles of their operation. We emphasize preliminary design and overall performance of turbomachines rather than detailed design. In addition, we discuss how to properly match the requirements of a fluid flow system to the performance characteristics of a turbomachine. A significant portion of this chapter is devoted to *turbomachinery scaling laws*—a practical application of dimensional analysis. We show how the scaling laws are used in the design of new turbomachines that are geometrically similar to existing ones.

OBJECTIVES

When you finish reading this chapter, you should be able to

- Identify various types of pumps and turbines, and understand how they work
- Apply dimensional analysis to design new pumps or turbines that are geometrically similar to existing pumps or turbines
- Perform basic vector analysis of the flow into and out of pumps and turbines
- Use specific speed for preliminary design and selection of pumps and turbines

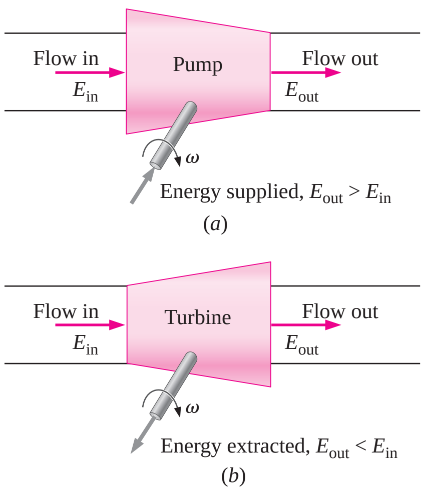


FIGURE 14–1
(*a*) A pump supplies energy to a fluid, while (*b*) a turbine extracts energy from a fluid.

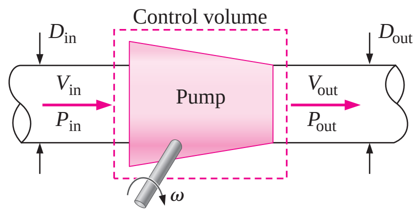


FIGURE 14–2
For the case of steady flow, conservation of mass requires that the mass flow rate out of a pump must equal the mass flow rate into the pump; for incompressible flow with equal inlet and outlet cross-sectional areas ($D_{out} = D_{in}$), we conclude that $V_{out} = V_{in}$, but $P_{out} > P_{in}$.

	Fan	Blower	Compressor
ΔP	Low	Medium	High
$\dot{V}$	High	Medium	Low

FIGURE 14–3
When used with gases, pumps are called *fans*, *blowers*, or *compressors*, depending on the relative values of pressure rise and volume flow rate.

14–1 ▪ CLASSIFICATIONS AND TERMINOLOGY

There are two broad categories of turbomachinery, **pumps** and **turbines.** The word *pump* is a general term for any fluid machine that *adds* energy to a fluid. Some authors call pumps **energy absorbing devices** since energy is supplied *to* them, and they transfer most of that energy to the fluid, usually via a rotating shaft (Fig. 14–1*a*). The increase in fluid energy is usually felt as an increase in the pressure of the fluid. Turbines, on the other hand, are **energy producing devices**—they extract energy *from* the fluid and transfer most of that energy to some form of mechanical energy output, typically in the form of a rotating shaft (Fig. 14–1*b*). The fluid at the outlet of a turbine suffers an energy loss, typically in the form of a loss of pressure.

An ordinary person may think that the energy supplied to a pump increases the speed of fluid passing through the pump and that a turbine extracts energy from the fluid by slowing it down. This is not necessarily the case. Consider a control volume surrounding a pump (Fig. 14–2). We assume steady conditions. By this we mean that neither the mass flow rate nor the rotational speed of the rotating blades changes with time. (The detailed flow field near the rotating blades inside the pump is *not* steady of course, but control volume analysis is not concerned with details inside the control volume.) By conservation of mass, we know that the mass flow rate into the pump must equal the mass flow rate out of the pump. If the flow is incompressible, the volume flow rates at the inlet and outlet must be equal as well. Furthermore, if the diameter of the outlet is the same as that of the inlet, conservation of mass requires that the average speed across the outlet must be identical to the average speed across the inlet. In other words, the pump does not necessarily increase the *speed* of the fluid passing through it; rather, it increases the *pressure* of the fluid. Of course, if the pump were turned off, there might be no flow at all. So, the pump *does* increase fluid speed compared to the case of no pump in the system. However, in terms of changes from the inlet to the outlet *across* the pump, fluid speed is not necessarily increased. (The output speed may even be *lower* than the input speed if the outlet diameter is larger than that of the inlet.)

> The purpose of a pump is to add energy to a fluid, resulting in an increase in fluid pressure, not necessarily an increase of fluid speed across the pump.

An analogous statement is made about the purpose of a turbine:

> The purpose of a turbine is to extract energy from a fluid, resulting in a decrease of fluid pressure, not necessarily a decrease of fluid speed across the turbine.

Fluid machines that move liquids are called **pumps,** but there are several other names for machines that move gases (Fig. 14–3). A **fan** is a gas pump with relatively low pressure rise and high flow rate. Examples include ceiling fans, house fans, and propellers. A **blower** is a gas pump with relatively moderate to high pressure rise and moderate to high flow rate. Examples include centrifugal blowers and squirrel cage blowers in automobile ventilation systems, furnaces, and leaf blowers. A **compressor** is a gas pump designed to deliver a very high pressure rise, typically at low to moderate flow rates. Examples include air compressors that run pneumatic tools and inflate tires at automobile service stations, and refrigerant compressors used in heat pumps, refrigerators, and air conditioners.

Pumps and turbines in which energy is supplied or extracted by a rotating shaft are properly called **turbomachines,** since the Latin prefix *turbo* means "spin." Not all pumps or turbines utilize a rotating shaft, however. The hand-operated air pump you use to inflate the tires of your bicycle is a prime example (Fig. 14–4*a*). The up and down reciprocating motion of a plunger or piston replaces the rotating shaft in this type of pump, and it is more proper to call it simply a **fluid machine** instead of a turbomachine. An old-fashioned well pump operates in a similar manner to pump water instead of air (Fig. 14–4*b*). Nevertheless, the words *turbomachine* and *turbomachinery* are often used in the literature to refer to *all* types of pumps and turbines regardless of whether they utilize a rotating shaft or not.

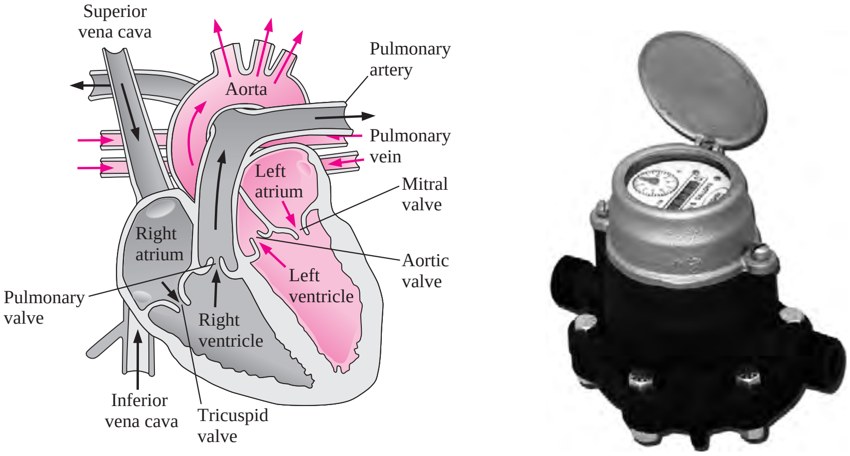
Fluid machines may also be broadly classified as either *positive-displacement* machines or *dynamic* machines, based on the manner in which energy transfer occurs. In **positive-displacement machines,** fluid is directed into a closed volume. Energy transfer to the fluid is accomplished by movement of the boundary of the closed volume, causing the volume to expand or contract, thereby sucking fluid in or squeezing fluid out, respectively. Your heart is a good example of a **positive-displacement pump** (Fig. 14–5*a*). It is designed with one-way valves that open to let blood in as heart chambers expand, and other one-way valves that open as blood is pushed out of those chambers when they contract. An example of a **positive-displacement turbine** is the common water meter in your house (Fig. 14–5*b*), in which water forces itself into a closed chamber of expanding volume connected to an

(*a*) (*b*)

FIGURE 14–4

Not all pumps have a rotating shaft; (*a*) energy is supplied to this manual tire pump by the up and down motion of a person's arm to pump air; (*b*) a similar mechanism is used to pump water with an old-fashioned well pump.

(a) Photo by Andrew Cimbala, with permission. (b) © The McGraw-Hill Companies, Inc./Ellen Behrman, photographer.



(*a*) (*b*)

FIGURE 14–5

(*a*) The human heart is an example of a *positive-displacement pump*; blood is pumped by expansion and contraction of heart chambers called *ventricles*. (*b*) The common water meter in your house is an example of a *positive-displacement turbine*; water fills and exits a chamber of known volume for each revolution of the output shaft.

Photo courtesy of Niagara Meters, Spartanburg, SC. Used by permission.

FIGURE 14–6
A wind turbine is a good example of a dynamic machine of the open type; air turns the blades, and the output shaft drives an electric generator.
The Wind Turbine Company. Used by permission.

output shaft that turns as water enters the chamber. The boundary of the volume then collapses, turning the output shaft some more, and letting the water continue on its way to your sink, shower, etc. The water meter records each 360° rotation of the output shaft, and the meter is precisely calibrated to the known volume of fluid in the chamber.

In **dynamic machines,** there is no closed volume; instead, rotating blades supply or extract energy to or from the fluid. For pumps, these rotating blades are called **impeller blades,** while for turbines, the rotating blades are called **runner blades** or **buckets.** Examples of **dynamic pumps** include **enclosed pumps** and **ducted pumps** (those with casings around the blades such as the water pump in your car's engine), and **open pumps** (those without casings such as the ceiling fan in your house, the propeller on an airplane, or the rotor on a helicopter). Examples of **dynamic turbines** include **enclosed turbines,** such as the hydroturbine that extracts energy from water in a hydroelectric dam, and **open turbines** such as the wind turbine that extracts energy from the wind (Fig. 14–6).

14–2 ▪ PUMPS

Some fundamental parameters are used to analyze the performance of a pump. The **mass flow rate** of fluid through the pump, $\dot{m}$, is an obvious primary pump performance parameter. For incompressible flow, it is more common to use **volume flow rate** rather than mass flow rate. In the turbomachinery industry, volume flow rate is called **capacity** and is simply mass flow rate divided by fluid density,

Volume flow rate (capacity):
$$\dot{V} = \frac{\dot{m}}{\rho} \tag{14–1}$$

The performance of a pump is characterized additionally by its **net head** H, defined as the change in **Bernoulli head** between the inlet and outlet of the pump,

Net head:
$$H = \left(\frac{P}{\rho g} + \frac{V^2}{2g} + z\right)_{\text{out}} - \left(\frac{P}{\rho g} + \frac{V^2}{2g} + z\right)_{\text{in}} \tag{14–2}$$

The dimension of net head is length, and it is often listed as an equivalent column height of water, even for a pump that is not pumping water.

For the case in which a *liquid* is being pumped, the Bernoulli head at the inlet is equivalent to the **energy grade line** at the inlet, EGL_{in}, obtained by aligning a Pitot probe in the center of the flow as illustrated in Fig. 14–7. The energy grade line at the outlet EGL_{out} is obtained in the same manner, as also illustrated in the figure. In the general case, the outlet of the pump may be at a different elevation than the inlet, and its diameter and average speed may not be the same as those at the inlet. Regardless of these differences, net head H is equal to the difference between EGL_{out} and EGL_{in},

Net head for a liquid pump:
$$H = \text{EGL}_{\text{out}} - \text{EGL}_{\text{in}}$$

Consider the special case of incompressible flow through a pump in which the inlet and outlet diameters are identical, and there is no change in elevation. Equation 14–2 reduces to

Special case with $D_{\text{out}} = D_{\text{in}}$ and $z_{\text{out}} = z_{\text{in}}$:
$$H = \frac{P_{\text{out}} - P_{\text{in}}}{\rho g}$$

For this simplified case, net head is simply the pressure rise across the pump expressed as a head (column height of the fluid).

Net head is proportional to the useful power actually delivered to the fluid. It is traditional to call this power the **water horsepower,** even if the fluid being pumped is not water, and even if the power is not measured in units of horsepower. By dimensional reasoning, we must multiply the net head of Eq. 14–2 by mass flow rate and gravitational acceleration to obtain dimensions of power. Thus,

Water horsepower: $$\dot{W}_{\text{water horsepower}} = \dot{m}gH = \rho g\dot{V}H \qquad \textbf{(14–3)}$$

All pumps suffer from irreversible losses due to friction, internal leakage, flow separation on blade surfaces, turbulent dissipation, etc. Therefore, the mechanical energy supplied to the pump must be *larger* than $\dot{W}_{\text{water horsepower}}$. In pump terminology, the external power supplied to the pump is called the **brake horsepower,** which we abbreviate as bhp. For the typical case of a rotating shaft supplying the brake horsepower,

Brake horsepower: $$\text{bhp} = \dot{W}_{\text{shaft}} = \omega T_{\text{shaft}} \qquad \textbf{(14–4)}$$

where ω is the rotational speed of the shaft (rad/s) and T_{shaft} is the torque supplied to the shaft. We define **pump efficiency** η_{pump} as the ratio of useful power to supplied power,

Pump efficiency: $$\eta_{\text{pump}} = \frac{\dot{W}_{\text{water horsepower}}}{\dot{W}_{\text{shaft}}} = \frac{\dot{W}_{\text{water horsepower}}}{\text{bhp}} = \frac{\rho g\dot{V}H}{\omega T_{\text{shaft}}} \qquad \textbf{(14–5)}$$

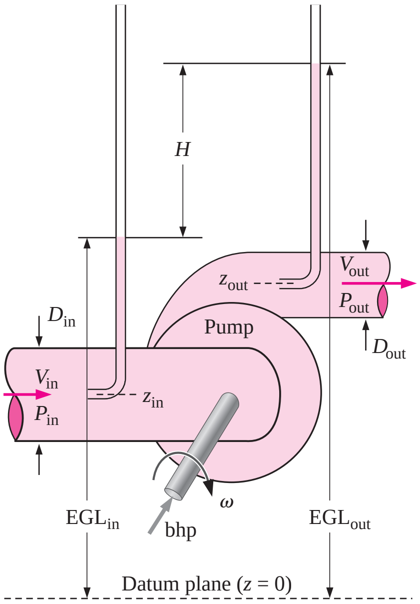


FIGURE 14–7
The *net head* of a pump, H, is defined as the change in Bernoulli head from inlet to outlet; for a liquid, this is equivalent to the change in the energy grade line, $H = \text{EGL}_{\text{out}} - \text{EGL}_{\text{in}}$, relative to some arbitrary datum plane; bhp is the *brake horsepower*, the external power supplied to the pump.

Pump Performance Curves and Matching a Pump to a Piping System

The maximum volume flow rate through a pump occurs when its net head is zero, $H = 0$; this flow rate is called the pump's **free delivery.** The free delivery condition is achieved when there is no flow restriction at the pump inlet or outlet—in other words when there is no **load** on the pump. At this operating point, $\dot{V}$ is large, but H is zero; the pump's efficiency is zero because the pump is doing no useful work, as is clear from Eq. 14–5. At the other extreme, the **shutoff head** is the net head that occurs when the volume flow rate is zero, $\dot{V} = 0$, and is achieved when the outlet port of the pump is blocked off. Under these conditions, H is large but $\dot{V}$ is zero; the pump's efficiency (Eq. 14–5) is again zero, because the pump is doing no useful work. Between these two extremes, from shutoff to free delivery, the pump's net head may increase from its shutoff value somewhat as the flow rate increases, but H must eventually decrease to zero as the volume flow rate increases to its free delivery value. The pump's efficiency reaches its maximum value somewhere between the shutoff condition and the free delivery condition; this operating point of maximum efficiency is appropriately called the **best efficiency point** (BEP), and is notated by an asterisk (H^*, $\dot{V}^*$, bhp*). Curves of H, η_{pump}, and bhp as functions of $\dot{V}$ are called **pump performance curves** (or *characteristic curves*, Chap. 8); typical curves at one rotational speed are plotted in Fig. 14–8. The pump performance curves change with rotational speed.

It is important to realize that *for steady conditions, a pump can operate only along its performance curve*. Thus, the operating point of a piping system is

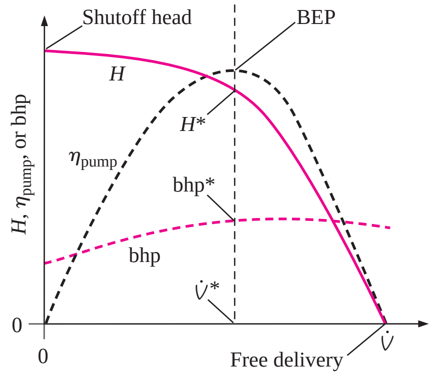


FIGURE 14–8
Typical *pump performance curves* for a centrifugal pump with backward-inclined blades; the curve shapes for other types of pumps may differ, and the curves change as shaft rotation speed is changed.

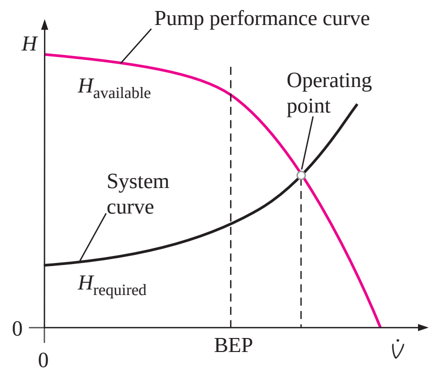


FIGURE 14–9
The *operating point* of a piping system is established as the volume flow rate where the system curve and the pump performance curve intersect.

determined by matching system requirements (*required* net head) to pump performance (*available* net head). In a typical application, $H_{required}$ and $H_{available}$ match at one unique value of flow rate—this is the **operating point** or **duty point** of the system.

The steady operating point of a piping system is established at the volume flow rate where $H_{required} = H_{available}$.

For a given piping system with its major and minor losses, elevation changes, etc., the required net head *increases* with volume flow rate. On the other hand, the available net head of most pumps *decreases* with flow rate, as in Fig. 14–8, at least over the majority of its recommended operating range. Hence, the system curve and the pump performance curve intersect as sketched in Fig. 14–9, and this establishes the operating point. If we are lucky, the operating point is at or near the best efficiency point of the pump. In most cases, however, as illustrated in Fig. 14–9, the pump does not run at its optimum efficiency. If efficiency is of major concern, the pump should be carefully selected (or a new pump should be designed) such that the operating point is as close to the best efficiency point as possible. In some cases it may be possible to change the shaft rotation speed so that an existing pump can operate much closer to its design point (best efficiency point).

There are unfortunate situations where the system curve and the pump performance curve intersect at more than one operating point. This can occur when a pump that has a dip in its net head performance curve is mated to a system that has a fairly flat system curve, as illustrated in Fig. 14-10. Although rare, such situations are possible and should be avoided, because the system may "hunt" for an operating point, leading to an unsteady-flow situation.

It is fairly straightforward to match a piping system to a pump, once we realize that the term for **useful pump head** ($h_{pump,\,u}$) that we used in the head form of the energy equation (Chap. 5) is the same as the *net head* (H) used in the present chapter. Consider, for example, a general piping system with elevation change, major and minor losses, and fluid acceleration (Fig. 14–11). We begin by solving the energy equation for the **required net head** $H_{required}$,

$$H_{required} = h_{pump,\,u} = \frac{P_2 - P_1}{\rho g} + \frac{\alpha_2 V_2^2 - \alpha_1 V_1^2}{2g} + (z_2 - z_1) + \cancel{h_{turbine}} + h_{L,\,total} \quad \textbf{(14–6)}$$

where we assume that there is no turbine in the system, although that term can be added back in, if necessary. We have also included the kinetic energy correction factors in Eq. 14–6 for greater accuracy, even though it is common practice in the turbomachinery industry to ignore them (α_1 and α_2 are often assumed to be unity since the flow is turbulent).

Equation 14–6 is evaluated from the inlet of the piping system (point 1, upstream of the pump) to the outlet of the piping system (point 2, downstream of the pump). Equation 14–6 agrees with our intuition, because it tells us that the useful pump head delivered to the fluid does four things:

- It increases the *static pressure* of the fluid from point 1 to point 2 (first term on the right).
- It increases the *dynamic pressure* (kinetic energy) of the fluid from point 1 to point 2 (second term on the right).

- It raises the *elevation* (potential energy) of the fluid from point 1 to point 2 (third term on the right).
- It overcomes *irreversible head losses* in the piping system (last term on the right).

In a general system, the change in static pressure, dynamic pressure, and elevation may be either positive or negative, while irreversible head losses are *always positive*. In many mechanical and civil engineering problems in which the fluid is a liquid, the elevation term is important, but when the fluid is a gas, such as in ventilation and air pollution control problems, the elevation term is almost always negligible.

To match a pump to a system, and to determine the operating point, we equate H_{required} of Eq. 14–6 to $H_{\text{available}}$, which is the (typically known) net head of the pump as a function of volume flow rate.

Operating point: $$H_{\text{required}} = H_{\text{available}} \qquad \textbf{(14–7)}$$

The most common situation is that an engineer selects a pump that is somewhat heftier than actually required. The volume flow rate through the piping system is then a bit larger than needed, and a valve or damper is installed in the line so that the flow rate can be decreased as necessary.

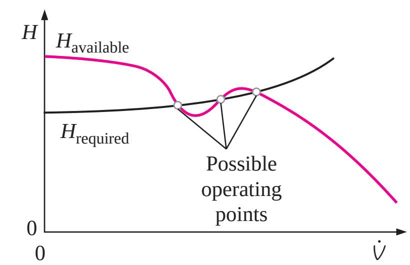


FIGURE 14–10
Situations in which there can be more than one unique operating point should be avoided. In such cases a different pump should be used.

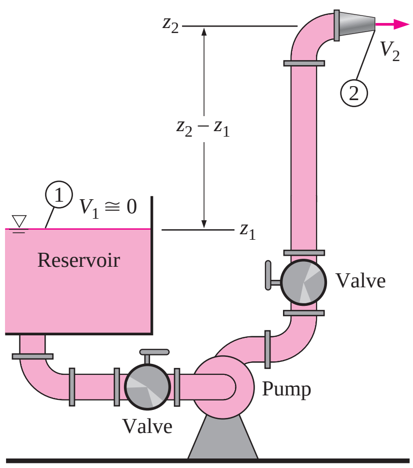


FIGURE 14–11
Equation 14–6 emphasizes the role of a pump in a piping system; namely, it increases (or decreases) the static pressure, dynamic pressure, and elevation of the fluid, and it overcomes irreversible losses.

EXAMPLE 14–1 Operating Point of a Fan in a Ventilation System

A *local ventilation system* (hood and exhaust duct) is used to remove air and contaminants produced by a dry-cleaning operation (Fig. 14–12). The duct is round and is constructed of galvanized steel with longitudinal seams and with joints every 30 in (0.76 m). The inner diameter (ID) of the duct is $D = 9.06$ in (0.230 m), and its total length is $L = 44.0$ ft (13.4 m). There are five CD3-9 elbows along the duct. The equivalent roughness height of this duct is 0.15 mm, and each elbow has a minor (local) loss coefficient of $K_L = C_0 = 0.21$. Note the notation C_0 for minor loss coefficient, commonly used in the ventilation industry (ASHRAE, 2001). To ensure adequate ventilation, the minimum required volume flow rate through the duct is $\dot{V} = 600$ cfm (cubic feet per minute), or 0.283 m^3/s at 25°C. Literature from the hood manufacturer lists the hood entry loss coefficient as 1.3 based on duct velocity. When the damper is fully open, its loss coefficient is 1.8. A centrifugal fan with 9.0-in inlet and outlet diameters is available. Its performance data are shown in Table 14–1, as listed by the manufacturer. Predict the operating point of this local ventilation system, and draw a plot of required and available fan pressure rise as functions of volume flow rate. Is the chosen fan adequate?

SOLUTION We are to estimate the operating point for a given fan and duct system and to plot required and available fan pressure rise as functions of volume flow rate. We are then to determine if the selected fan is adequate.

Assumptions **1** The flow is steady. **2** The concentration of contaminants in the air is low; the fluid properties are those of air alone. **3** The flow at the outlet is fully developed turbulent pipe flow with $\alpha = 1.05$.

Properties For air at 25°C, $\nu = 1.562 \times 10^{-5}$ m^2/s and $\rho = 1.184$ kg/m^3. Standard atmospheric pressure is $P_{\text{atm}} = 101.3$ kPa.

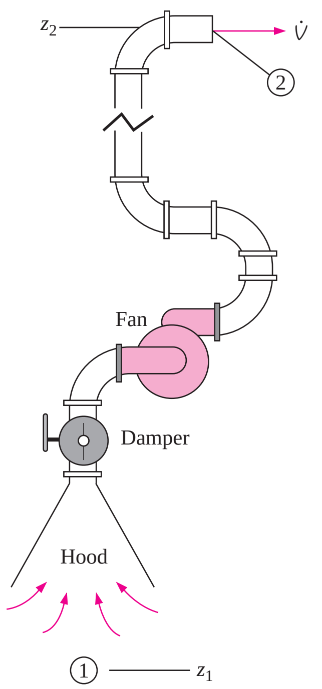


FIGURE 14–12
The local ventilation system for Example 14–1, showing the fan and all minor losses.

TABLE 14–1

Manufacturer's performance data for the fan of Example 14–1*

$\dot{V}$, cfm	$(\delta P)_{fan}$, inches H_2O
0	0.90
250	0.95
500	0.90
750	0.75
1000	0.40
1200	0.0

* Note that the pressure rise data are listed as inches of *water*, even though *air* is the fluid. This is common practice in the ventilation industry.

Analysis We apply the steady energy equation in head form (Eq. 14–6) from point 1 in the stagnant air region in the room to point 2 at the duct outlet,

$$H_{required} = \frac{P_2 - P_1}{\rho g} + \frac{\alpha_2 V_2^2 - \alpha_1 V_1^2}{2g} + \underbrace{(z_2 - z_1)}_{\text{negligible for gases}} + h_{L,\ total} \quad (1)$$

In Eq. 1 we may ignore the air speed at point 1 since it was chosen (wisely) far enough away from the hood inlet so that the air is nearly stagnant. At point 1, P_1 is equal to P_{atm}, and at point 2, P_2 is also equal to P_{atm} since the jet discharges into the outside air on the roof of the building. Thus the pressure terms cancel out and Eq. 1 reduces to

Required net head:
$$H_{required} = \frac{\alpha_2 V_2^2}{2g} + h_{L,\ total} \quad (2)$$

The total head loss in Eq. 2 is a combination of major and minor losses and depends on volume flow rate. Since the duct diameter is constant,

Total irreversible head loss:
$$h_{L,\ total} = \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g} \quad (3)$$

The dimensionless roughness factor is ε/D = (0.15 mm)/(230 mm) = 6.52×10^{-4}. The Reynolds number of the air flowing through the duct is

Reynolds number:
$$\text{Re} = \frac{DV}{\nu} = \frac{D}{\nu}\frac{4\dot{V}}{\pi D^2} = \frac{4\dot{V}}{\nu \pi D} \quad (4)$$

The Reynolds number varies with volume flow rate. At the minimum required flow rate, the air speed through the duct is $V = V_2$ = 6.81 m/s, and the Reynolds number is

$$\text{Re} = \frac{4(0.283\ \text{m}^3/\text{s})}{(1.562 \times 10^{-5}\ \text{m}^2/\text{s})\pi(0.230\ \text{m})} = 1.00 \times 10^5$$

From the Moody chart (or the Colebrook equation) at this Reynolds number and roughness factor, the friction factor is f = 0.0209. The sum of all the minor loss coefficients is

Minor losses:
$$\sum K_L = 1.3 + 5(0.21) + 1.8 = 4.15 \quad (5)$$

Substituting these values at the minimum required flow rate into Eq. 2, the required net head of the fan at the minimum flow rate is

$$H_{required} = \left(\alpha_2 + f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g}$$

$$= \left(1.05 + 0.0209\frac{13.4\ \text{m}}{0.230\ \text{m}} + 4.15\right)\frac{(6.81\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)} = 15.2\ \text{m of air} \quad (6)$$

Note that the head is expressed naturally in units of equivalent column height of the pumped fluid, which is air in this case. We convert to an equivalent column height of *water* by multiplying by the ratio of air density to water density,

$$H_{\text{required, inches of water}} = H_{\text{required, air}} \frac{\rho_{\text{air}}}{\rho_{\text{water}}}$$

$$= (15.2 \text{ m}) \frac{1.184 \text{ kg/m}^3}{998.0 \text{ kg/m}^3} \left(\frac{1 \text{ in}}{0.0254 \text{ m}}\right)$$

$$= 0.709 \text{ inches of water} \qquad \textbf{(7)}$$

We repeat the calculations at several values of volume flow rate, and compare to the available net head of the fan in Fig. 14–13. The operating point is at a volume flow rate of about **650 cfm**, at which both the required and available net head equal about **0.83 inches of water**. We conclude that **the chosen fan is more than adequate for the job**.

Discussion The purchased fan is somewhat more powerful than required, yielding a higher flow rate than necessary. The difference is small and is acceptable; the butterfly damper valve could be partially closed to cut back the flow rate to 600 cfm if necessary. For safety reasons, it is clearly better to oversize than undersize a fan when used with an air pollution control system.

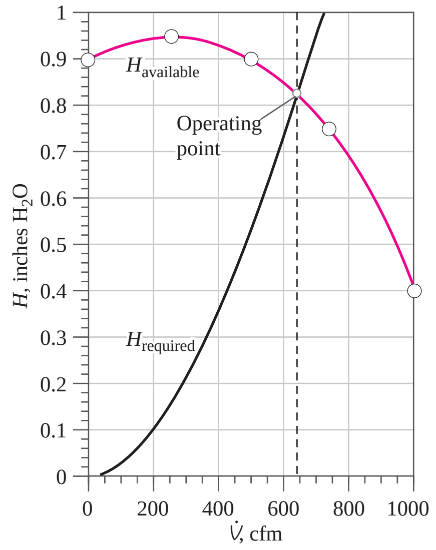


FIGURE 14–13

Net head as a function of volume flow rate for the ventilation system of Example 14–1. The point where the available and required values of H intersect is the operating point.

It is common practice in the pump industry to offer several choices of impeller diameter for a single pump casing. There are several reasons for this: (1) to save manufacturing costs, (2) to enable capacity increase by simple impeller replacement, (3) to standardize installation mountings, and (4) to enable reuse of equipment for a different application. When plotting the performance of such a "family" of pumps, pump manufacturers do not plot separate curves of H, η_{pump}, and bhp for each impeller diameter in the form sketched in Fig. 14–8. Instead, they prefer to combine the performance curves of an entire family of pumps of different impeller diameters onto a single plot (Fig. 14–14). Specifically, they plot a curve of H as a function of $\dot{V}$ for each impeller diameter in the same way as in Fig. 14–8, but create *contour lines* of constant efficiency, by drawing smooth curves through points that have the same value of η_{pump} for the various choices of impeller diameter. Contour lines of constant bhp are often drawn on the same plot in similar fashion. An example is provided in Fig. 14–15 for a family of centrifugal pumps manufactured by Taco, Inc. In this case, five impeller diameters are available, but the identical pump casing is used for all five options. As seen in Fig. 14–15, pump manufacturers do not always plot their pumps' performance curves all the way to free delivery. This is because the pumps are usually not operated there due to the low values of net head and efficiency. If higher values of flow rate and net head are required, the customer should step up to the next larger casing size, or consider using additional pumps in series or parallel.

It is clear from the performance plot of Fig. 14–15 that for a given pump casing, the larger the impeller, the higher the maximum achievable efficiency. Why then would anyone buy the smaller impeller pump? To answer this question, we must recognize that the customer's application requires a certain combination of flow rate and net head. If the requirements match a particular impeller diameter, it may be more cost effective to sacrifice pump efficiency in order to satisfy these requirements.

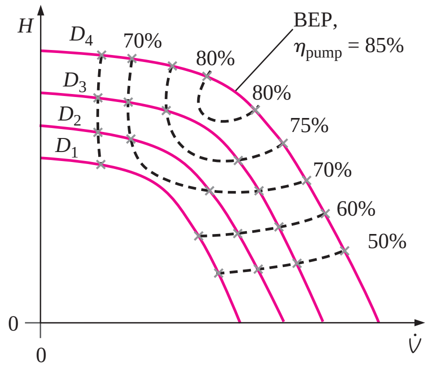


FIGURE 14–14

Typical pump performance curves for a *family* of centrifugal pumps of the same casing diameter but different impeller diameters.

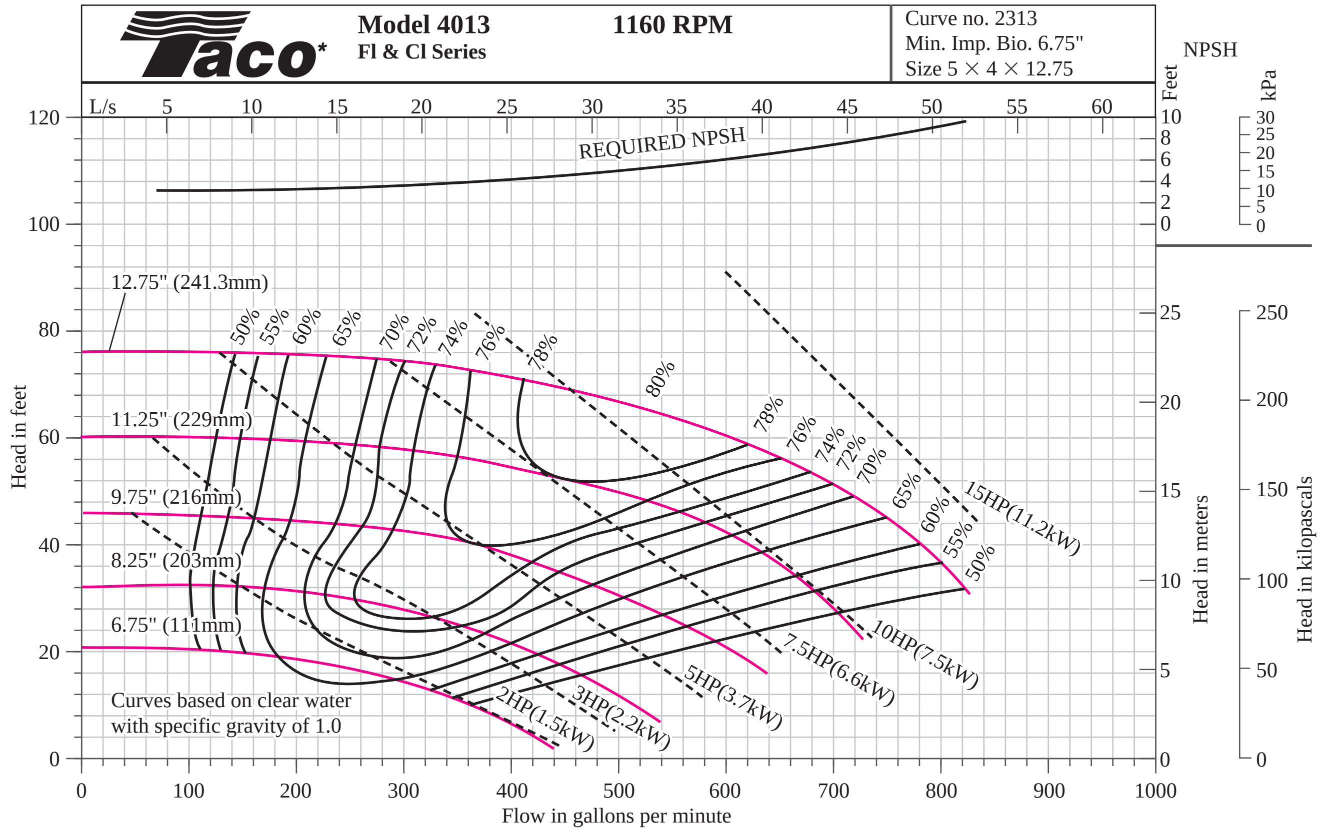


FIGURE 14–15

Example of a manufacturer's performance plot for a family of centrifugal pumps. Each pump has the same casing, but a different impeller diameter.

Courtesy of Taco, Inc., Cranston, RI. Used by permission.

EXAMPLE 14–2 Selection of Pump Impeller Size

A washing operation at a power plant requires 370 gallons per minute (gpm) of water. The required net head is about 24 ft at this flow rate. A newly hired engineer looks through some catalogs and decides to purchase the 8.25-in impeller option of the Taco Model 4013 FI Series centrifugal pump of Fig. 14–15. If the pump operates at 1160 rpm, as specified in the performance plot, she reasons, its performance curve intersects 370 gpm at H = 24 ft. The chief engineer, who is very concerned about efficiency, glances at the performance curves and notes that the efficiency of this pump at this operating point is only 70 percent. He sees that the 12.75-in impeller option achieves a higher efficiency (about 76.5 percent) at the same flow rate. He notes that a throttle valve can be installed downstream of the pump to increase the required net head so that the pump operates at this higher efficiency. He asks the junior engineer to justify her choice of impeller diameter. Namely, he asks her to calculate which impeller option

(8.25-in or 12.75-in) would need the least amount of electricity to operate (Fig. 14–16). Perform the comparison and discuss.

SOLUTION For a given flow rate and net head, we are to calculate which impeller size uses the least amount of power, and we are to discuss our results.

Assumptions **1** The water is at 70°F. **2** The flow requirements (volume flow rate and head) are constant.

Properties For water at 70°F, ρ = 62.30 lbm/ft³.

Analysis From the contours of brake horsepower that are shown on the performance plot of Fig. 14–15, the junior engineer estimates that the pump with the smaller impeller requires about 3.2 hp from the motor. She verifies this estimate by using Eq. 14–5,

Required bhp for the 8.25-in impeller option:

$$\text{bhp} = \frac{\rho g \dot{V} H}{\eta_{\text{pump}}} = \frac{(62.30\ \text{lbm/ft}^3)(32.2\ \text{ft/s}^2)(370\ \text{gal/min})(24\ \text{ft})}{0.70}$$

$$\times \left(\frac{0.1337\ \text{ft}^3}{\text{gal}}\right)\left(\frac{\text{lbf}}{32.2\ \text{lbm}\cdot\text{ft/s}^2}\right)\left(\frac{1\ \text{min}}{60\ \text{s}}\right)\left(\frac{\text{hp}\cdot\text{s}}{550\ \text{ft}\cdot\text{lbf}}\right) = 3.20\ \text{hp}$$

Similarly, the larger-diameter impeller option requires

Required bhp for the 12.75-in impeller option: $\text{bhp} = 8.78\ \text{hp}$

using the operating point of that pump, namely, $\dot{V}$ = 370 gpm, H = 72.0 ft, and η_{pump} = 76.5 percent (Fig. 14–15). Clearly, **the smaller-diameter impeller option is the better choice in spite of its lower efficiency, because it uses less than half the power**.

Discussion Although the larger impeller pump would operate at a somewhat higher value of efficiency, it would deliver about 72 ft of net head at the required flow rate. This is overkill, and the throttle valve would be required to make up the difference between this net head and the required flow head of 24 ft of water. A throttle valve does nothing more than waste mechanical energy, however; so the gain in efficiency of the pump is more than offset by losses through the throttle valve. If the flow head or capacity requirements increase at some time in the future, a larger impeller can be purchased for the same casing.

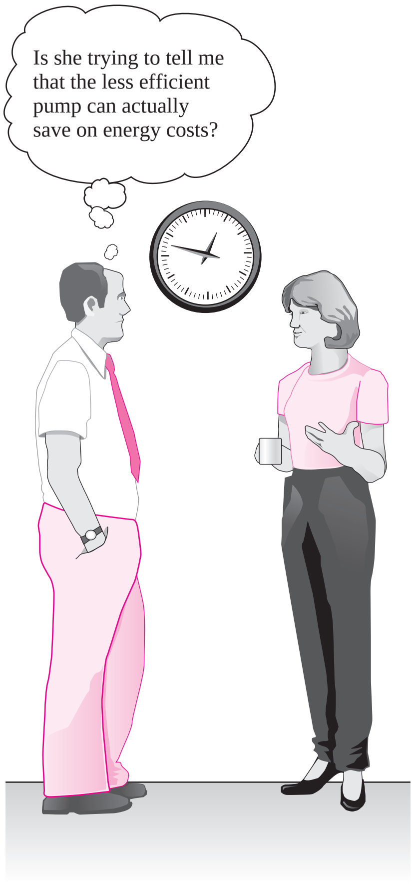


FIGURE 14–16
In some applications, a less efficient pump from the same family of pumps may require less energy to operate. An even better choice, however, would be a pump whose best efficiency point occurs at the required operating point of the pump, but such a pump is not always commercially available.

Pump Cavitation and Net Positive Suction Head

When pumping liquids, it is possible for the local pressure inside the pump to fall below the **vapor pressure** of the liquid, P_v. (P_v is also called the **saturation pressure** P_{sat} and is listed in thermodynamics tables as a function of saturation temperature.) When $P < P_v$, vapor-filled bubbles called **cavitation bubbles** appear. In other words, the liquid *boils* locally, typically on the suction side of the rotating impeller blades where the pressure is lowest (Fig. 14–17). After the cavitation bubbles are formed, they are transported through the pump to regions where the pressure is higher, causing rapid collapse of the bubbles. It is this *collapse* of the bubbles that is undesirable, since it causes noise, vibration, reduced efficiency, and most importantly, damage to the impeller blades. Repeated bubble collapse near a blade surface leads to pitting or erosion of the blade and eventually catastrophic blade failure.

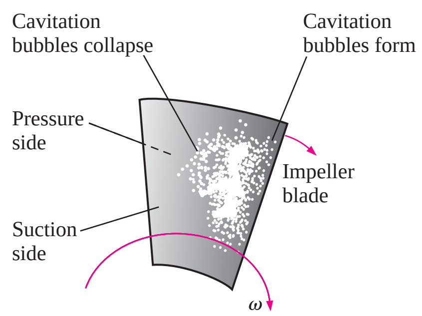


FIGURE 14–17
Cavitation bubbles forming and collapsing on the suction side of an impeller blade.

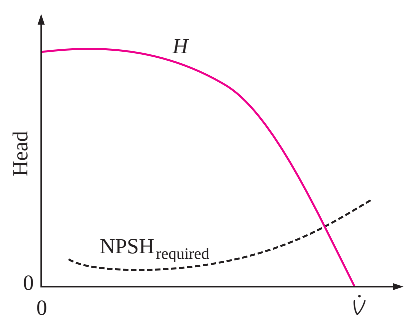


FIGURE 14–18
Typical pump performance curve in which net head and required net positive suction head are plotted versus volume flow rate.

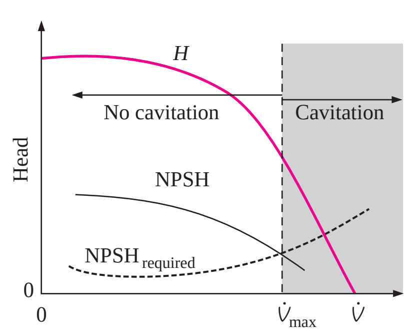


FIGURE 14–19
The volume flow rate at which the actual NPSH and the required NPSH intersect represents the maximum flow rate that can be delivered by the pump without the occurrence of cavitation.

To avoid cavitation, we must ensure that the local pressure everywhere inside the pump stays *above* the vapor pressure. Since pressure is most easily measured (or estimated) at the inlet of the pump, cavitation criteria are typically specified *at the pump inlet*. It is useful to employ a flow parameter called **net positive suction head (NPSH),** defined as *the difference between the pump's inlet stagnation pressure head and the vapor pressure head*,

Net positive suction head:
$$\text{NPSH} = \left(\frac{P}{\rho g} + \frac{V^2}{2g}\right)_{\text{pump inlet}} - \frac{P_v}{\rho g} \tag{14–8}$$

Pump manufacturers test their pumps for cavitation in a pump test facility by varying the volume flow rate and inlet pressure in a controlled manner. Specifically, at a given flow rate and liquid temperature, the pressure at the pump inlet is slowly lowered until cavitation occurs somewhere inside the pump. The value of NPSH is calculated using Eq. 14–8 and is recorded at this operating condition. The process is repeated at several other flow rates, and the pump manufacturer then publishes a performance parameter called the **required net positive suction head ($\text{NPSH}_{\text{required}}$),** defined as the *minimum NPSH necessary to avoid cavitation in the pump*. The measured value of $\text{NPSH}_{\text{required}}$ varies with volume flow rate, and therefore $\text{NPSH}_{\text{required}}$ is often plotted on the same pump performance curve as net head (Fig. 14–18). When expressed properly in units of head of the liquid being pumped, $\text{NPSH}_{\text{required}}$ is independent of the type of liquid. However, if the required net positive suction head is expressed for a particular liquid in pressure units such as pascals or psi, the engineer must be careful to convert this pressure to the equivalent column height of the actual liquid being pumped. Note that since $\text{NPSH}_{\text{required}}$ is usually much smaller than H over the majority of the performance curve, it is often plotted on a separate expanded vertical axis for clarity (see Fig. 14–15) or as contour lines when being shown for a family of pumps. $\text{NPSH}_{\text{required}}$ typically increases with volume flow rate, although for some pumps it decreases with $\dot{V}$ at low flow rates where the pump is not operating very efficiently, as sketched in Fig. 14–18.

In order to ensure that a pump does not cavitate, the actual or available NPSH must be greater than $\text{NPSH}_{\text{required}}$. It is important to note that the value of NPSH varies not only with flow rate, but also with liquid temperature, since P_v is a function of temperature. NPSH also depends on the type of liquid being pumped, since there is a unique P_v versus T curve for each liquid. Since irreversible head losses through the piping system upstream of the inlet *increase* with flow rate, the pump inlet stagnation pressure head *decreases* with flow rate. Therefore, the value of NPSH *decreases* with $\dot{V}$, as sketched in Fig. 14–19. By identifying the volume flow rate at which the curves of actual NPSH and $\text{NPSH}_{\text{required}}$ intersect, we estimate the maximum volume flow rate that can be delivered by the pump without cavitation (Fig. 14–19).

EXAMPLE 14–3 Maximum Flow Rate to Avoid Pump Cavitation

The 11.25-in impeller option of the Taco Model 4013 FI Series centrifugal pump of Fig. 14–15 is used to pump water at 25°C from a reservoir whose surface is 4.0 ft above the centerline of the pump inlet (Fig. 14–20). The piping system from the reservoir to the pump consists of 10.5 ft of cast iron pipe with an ID of 4.0 in and an average inner roughness height of 0.02 in.

There are several minor losses: a sharp-edged inlet (K_L = 0.5), three flanged smooth 90° regular elbows (K_L = 0.3 each), and a fully open flanged globe valve (K_L = 6.0). Estimate the maximum volume flow rate (in units of gpm) that can be pumped without cavitation. If the water were warmer, would this maximum flow rate increase or decrease? Why? Discuss how you might increase the maximum flow rate while still avoiding cavitation.

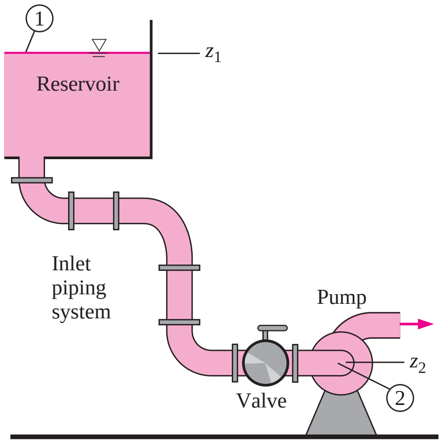


FIGURE 14–20
Inlet piping system from the reservoir (1) to the pump inlet (2) for Example 14–3.

SOLUTION For a given pump and piping system we are to estimate the maximum volume flow rate that can be pumped without cavitation. We are also to discuss the effect of water temperature and how we might increase the maximum flow rate.

Assumptions **1** The flow is steady. **2** The liquid is incompressible. **3** The flow at the pump inlet is turbulent and fully developed, with α = 1.05.

Properties For water at T = 25°C, ρ = 997.0 kg/m^3, μ = 8.91 $\times$ 10^{-4} kg/m · s, and P_v = 3.169 kPa. Standard atmospheric pressure is P_{atm} = 101.3 kPa.

Analysis We apply the steady energy equation in head form along a streamline from point 1 at the reservoir surface to point 2 at the pump inlet,

$$\frac{P_1}{\rho g} + \cancel{\frac{\alpha_1 V_1^2}{2g}} + z_1 + \cancel{h_{pump,\,u}} = \frac{P_2}{\rho g} + \frac{\alpha_2 V_2^2}{2g} + z_2 + \cancel{h_{turbine,\,e}} + h_{L,\,total} \quad (1)$$

In Eq. 1 we have ignored the water speed at the reservoir surface ($V_1 \cong 0$). There is no turbine in the piping system. Also, although there is a pump in the system, there is no pump between points 1 and 2; hence the pump head term also drops out. We solve Eq. 1 for $P_2/\rho g$, which is the pump inlet pressure expressed as a head,

Pump inlet pressure head:
$$\frac{P_2}{\rho g} = \frac{P_{atm}}{\rho g} + (z_1 - z_2) - \frac{\alpha_2 V_2^2}{2g} - h_{L,\,total} \quad (2)$$

Note that in Eq. 2, we have recognized that $P_1 = P_{atm}$ since the reservoir surface is exposed to atmospheric pressure.

The available net positive suction head at the pump inlet is obtained from Eq. 14–8. After substitution of Eq. 2, we get

Available NPSH:
$$\text{NPSH} = \frac{P_{atm} - P_v}{\rho g} + (z_1 - z_2) - h_{L,\,total} - \frac{(\alpha_2 - 1)V_2^2}{2g} \quad (3)$$

Since we know P_{atm}, P_v, and the elevation difference, all that remains is to estimate the total irreversible head loss through the piping system, which depends on volume flow rate. Since the pipe diameter is constant,

Irreversible head loss:
$$h_{L,\,total} = \left(f\frac{L}{D} + \sum K_L\right)\frac{V^2}{2g} \quad (4)$$

The rest of the problem is most easily solved on a computer. For a given volume flow rate, we calculate speed V and Reynolds number Re. From Re and the known pipe roughness, we use the Moody chart (or the Colebrook equation) to obtain friction factor f. The sum of all the minor loss coefficients is

Minor losses:
$$\sum K_L = 0.5 + 3 \times 0.3 + 6.0 = 7.4 \quad (5)$$

We make one calculation by hand for illustrative purposes. At $\dot{V}$ = 400 gpm (0.02523 m^3/s), the average speed of water through the pipe is

$$V = \frac{\dot{V}}{A} = \frac{4\dot{V}}{\pi D^2} = \frac{4(0.02523\ \text{m}^3/\text{s})}{\pi(4.0\ \text{in})^2}\left(\frac{1\ \text{in}}{0.0254\ \text{m}}\right)^2 = 3.112\ \text{m/s} \quad (6)$$

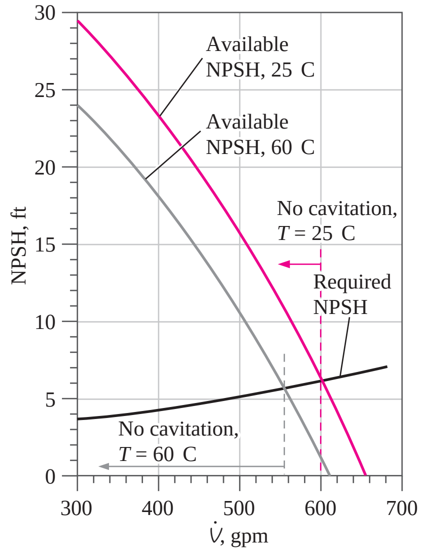


FIGURE 14–21
Net positive suction head as a function of volume flow rate for the pump of Example 14–3 at two temperatures. Cavitation is predicted to occur at flow rates greater than the point where the available and required values of NPSH intersect.

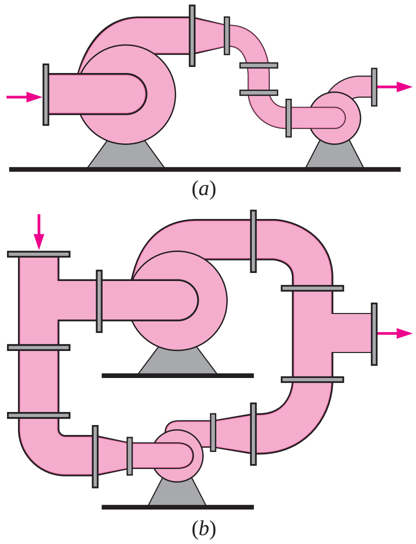


FIGURE 14–22
Arranging two very dissimilar pumps in (*a*) series or (*b*) parallel can sometimes lead to problems.

which produces a Reynolds number of Re = $\rho VD/\mu$ = 3.538×10^5. At this Reynolds number, and with roughness factor ε/D = 0.005, the Colebrook equation yields f = 0.0306. Substituting the given properties, along with f, D, L, and Eqs. 4, 5, and 6, into Eq. 3, we calculate the available net positive suction head at this flow rate,

$$\text{NPSH} = \frac{(101{,}300 - 3169)\ \text{N/m}^2}{(997.0\ \text{kg/m}^3)(9.81\ \text{m/s}^2)}\left(\frac{\text{kg}\cdot\text{m/s}^2}{\text{N}}\right) + 1.219\ \text{m}$$
$$- \left(0.0306\,\frac{10.5\ \text{ft}}{0.3333\ \text{ft}} + 7.4 - (1.05 - 1)\right)\frac{(3.112\ \text{m/s})^2}{2(9.81\ \text{m/s}^2)}$$
$$= 7.148\ \text{m} = 23.5\ \text{ft} \qquad \textbf{(7)}$$

The required net positive suction head is obtained from Fig. 14–15. At our example flow rate of 400 gpm, $\text{NPSH}_{\text{required}}$ is just above 4.0 ft. Since the actual NPSH is much higher than this, we need not worry about cavitation at this flow rate. We use EES (or a spreadsheet) to calculate NPSH as a function of volume flow rate, and the results are plotted in Fig. 14–21. It is clear from this plot that at 25°C, **cavitation occurs at flow rates above approximately 600 gpm**—close to the free delivery.

If the water were warmer than 25°C, the vapor pressure would increase, the viscosity would decrease, and the density would decrease slightly. The calculations are repeated at T = 60°C, at which ρ = 983.3 kg/m³, μ = 4.67×10^{-4} kg/m · s, and P_v = 19.94 kPa. The results are also plotted in Fig. 14–21, where we see that **the maximum volume flow rate without cavitation *decreases* with temperature** (to about 555 gpm at 60°C). This decrease agrees with our intuition, since warmer water is already closer to its boiling point from the start.

Finally, how can we increase the maximum flow rate? *Any modification that increases the available NPSH helps.* We can raise the height of the reservoir surface (to increase the hydrostatic head). We can reroute the piping so that only one elbow is necessary and replace the globe valve with a ball valve (to decrease the minor losses). We can increase the diameter of the pipe and decrease the surface roughness (to decrease the major losses). In this particular problem, the minor losses have the greatest influence, but in many problems, the major losses are more significant, and increasing the pipe diameter is most effective. That is one reason why many centrifugal pumps have a larger inlet diameter than outlet diameter.

Discussion Note that $\text{NPSH}_{\text{required}}$ does not depend on water temperature, but the actual or available NPSH decreases with temperature (Fig. 14–21).

Pumps in Series and Parallel

When faced with the need to increase volume flow rate or pressure rise by a small amount, you might consider adding an additional smaller pump in series or in parallel with the original pump. While series or parallel arrangement is acceptable for some applications, arranging *dissimilar* pumps in series or in parallel may lead to problems, especially if one pump is much larger than the other (Fig. 14–22). A better course of action is to increase the original pump's speed and/or input power (larger electric motor), replace the impeller with a larger one, or replace the entire pump with a larger one. The logic for this decision can be seen from the pump performance curves, realizing that *pressure rise and volume flow rate are related.* Arranging dissimilar pumps in series may create problems because the volume flow rate through each pump must be the same, but the overall pressure rise is equal to the pressure rise of one pump plus that of the other. If the pumps have widely different performance curves, the smaller pump may be forced to operate beyond its free delivery flow rate, whereupon it acts like a head *loss,* reducing the total volume flow rate. Arranging dissimilar pumps in parallel may create problems because the overall pressure rise must be the same, but the net volume flow rate is the sum of that through each branch. If the pumps are not sized properly, the smaller pump may not be able to handle the large head imposed on it, and the flow in its branch could actually be *reversed;* this would inadvertently reduce the overall pressure rise. In either case, the power supplied to the smaller pump would be wasted.

Keeping these cautions in mind, there are many applications where two or more similar (usually identical) pumps are operated in series or in parallel. When operated in *series,* the combined net head is simply the sum of the net heads of each pump (at a given volume flow rate),

Combined net head for n pumps in series: $$H_{\text{combined}} = \sum_{i=1}^{n} H_i \qquad \textbf{(14–9)}$$

Equation 14–9 is illustrated in Fig. 14–23 for three pumps in series. In this example, pump 3 is the strongest and pump 1 is the weakest. The shutoff head of the three pumps combined in series is equal to the sum of the shutoff head of each individual pump. For low values of volume flow rate, the net head of the three pumps in series is equal to $H_1 + H_2 + H_3$. Beyond the free delivery of pump 1 (to the right of the first vertical dashed gray line in Fig. 14–23), *pump 1 should be shut off and bypassed*. Otherwise it would be running beyond its maximum designed operating point, and the pump or its motor could be damaged. Furthermore, the net head across this pump would be *negative* as previously discussed, contributing to a net loss in the system. With pump 1 bypassed, the combined net head becomes $H_2 + H_3$. Similarly, beyond the free delivery of pump 2, that pump should also be shut off and bypassed, and the combined net head is then equal to H_3 alone, as indicated to the right of the second vertical dashed gray line in Fig. 14–23. In this

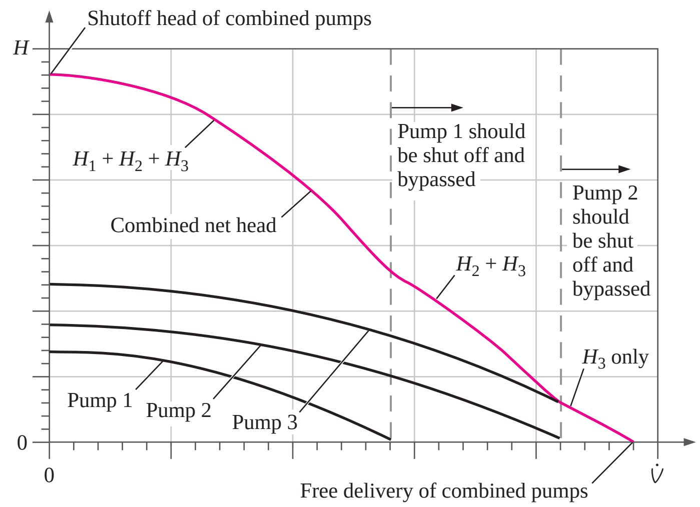


FIGURE 14–23
Pump performance curve (dark blue) for three dissimilar pumps in *series*. At low values of volume flow rate, the combined net head is equal to the sum of the net head of each pump by itself. However, to avoid pump damage and loss of combined net head, any individual pump should be shut off and bypassed at flow rates larger than that pump's free delivery, as indicated by the vertical dashed gray lines. If the three pumps were identical, it would not be necessary to turn off any of the pumps, since the free delivery of each pump would occur at the same volume flow rate.

case, the combined free delivery is the same as that of pump 3 alone, assuming that the other two pumps are bypassed.

When two or more identical (or similar) pumps are operated in *parallel*, their individual volume flow rates (rather than net heads) are summed,

Combined capacity for n pumps in parallel: $$\dot{V}_{\text{combined}} = \sum_{i=1}^{n} \dot{V}_i \qquad \textbf{(14–10)}$$

As an example, consider the *same* three pumps, but arranged in parallel rather than in series. The combined pump performance curve is shown in Fig. 14–24. The free delivery of the three combined pumps is equal to the sum of the free delivery of each individual pump. For low values of net head, the capacity of the three pumps in parallel is equal to $\dot{V}_1 + \dot{V}_2 + \dot{V}_3$. Above the shutoff head of pump 1 (above the first horizontal dashed gray line in Fig. 14–24), *pump 1 should be shut off and its branch should be blocked* (with a valve). Otherwise it would be running beyond its maximum designed operating point, and the pump or its motor could be damaged. Furthermore, the volume flow rate through this pump would be *negative* as previously discussed, contributing to a net loss in the system. With pump 1 shut off and blocked, the combined capacity becomes $\dot{V}_2 + \dot{V}_3$. Similarly, above the shutoff head of pump 2, that pump should also be shut off and blocked. The combined capacity is then equal to $\dot{V}_3$ alone, as indicated above, the second horizontal dashed gray line in Fig. 14–24. In this case, the combined shutoff head is the same as that of pump 3 alone, assuming that the other two pumps are shut off and their branches are blocked.

In practice, several pumps may be combined in parallel to deliver a large volume flow rate (Fig. 14–25). Examples include banks of pumps used to circulate water in cooling towers and chilled water loops (Wright, 1999). Ideally all the pumps should be identical so that we don't need to worry

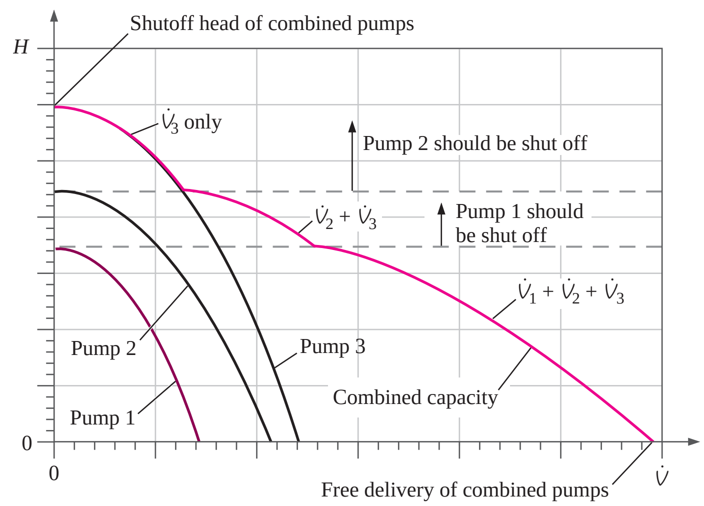


FIGURE 14–24
Pump performance curve (dark blue) for three pumps in *parallel*. At a low value of net head, the combined capacity is equal to the sum of the capacity of each pump by itself. However, to avoid pump damage and loss of combined capacity, any individual pump should be shut off at net heads larger than that pump's shutoff head, as indicated by the horizontal dashed gray lines. That pump's branch should also be blocked with a valve to avoid reverse flow. If the three pumps were identical, it would not be necessary to turn off any of the pumps, since the shutoff head of each pump would occur at the same net head.