CHAPTER 55
PUMPS AND FANS

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55.1 PUMP AND FAN SIMILARITY

The performance characteristics of centrifugal pumps and fans (i.e., rotating fluid machines) are described by the same basic laws and derived equations and, therefore, should be treated together and not separately. Both fluid machines provide the input energy to create flow and a pressure rise in their respective fluid systems and both use the principle of fluid acceleration as the mechanism to add this energy. If the pressure rise across a fan is small (5000 Pa), then the gas can be considered as an incompressible fluid, and the equations developed to describe the process will be the same as for pumps.

Compressors are used to obtain large increases in a gaseous fluid system. With such devices the compressibility of the gas must be considered, and a new set of derived equations must be developed to describe the compressor's performance. Because of this, the subject of gas compressors will be included in a separate chapter.

Mechanical Engineers' Handbook, 2nd ed., Edited by Myer Kutz.
ISBN 0-471-13007-9

55.2 SYSTEM DESIGN: THE FIRST STEP IN PUMP OR FAN SELECTION

55.2.1 Fluid System Data Required

The first step in selecting a pump or fan is to finalize the design of the piping or duct system (i.e., the "fluid system") into which the fluid machine is to be placed. The fluid machine will be selected to meet the flow and developed head requirements of the fluid system. The developed head is the energy that must be added to the fluid by the fluid machine, expressed as the potential energy of a column of fluid having a height H_p (meters). H_p is the "developed head." Consequently, the following data must be collected before the pump or fan can be selected:

1. Maximum flow rate required and variations expected
2. Detailed design (including layout and sizing) of the pipe or duct system, including all elbows, valves, dampers, heat exchangers, filters, etc
3. Exact location of the pump or fan in the fluid system, including its elevation
4. Fluid pressure and temperature available at start of system (suction)
5. Fluid pressure and temperature required at end of system (discharge)
6. Fluid characteristics (density, viscosity, corrosiveness, and erosiveness)

55.2.2 Determination of Fluid Head Required

The fluid head required is calculated using both the Bernoulli and D'Arcy equations from fluid mechanics. The Bernoulli equation represents the total mechanical (nonthermal) energy content of the fluid at any location in the system:

$$E_{T(1)} = P_1 v_1 + Z_1 g + V_1^2/2 \tag{55.1}$$

where $E_{T(1)}$ = total energy content of the fluid at location (1), J/kg
P_1 = absolute pressure of fluid at (1), Pa
v_1 = specific volume of fluid at (1), m^3/kg
Z_1 = elevation of fluid at (1), m
g = gravity constant, m/sec^2
V_1 = velocity of fluid at (1), m/sec

The D'Arcy equation expresses the loss of mechanical energy from a fluid through friction heating between any two locations in the system:

$$\bar{v}\Delta P_f(i,j) = f\, L_e(i-j)\, V^2/2D \qquad \text{J/kg·m} \tag{55.2}$$

where $\bar{v}$ = average fluid specific volume between two locations (i and j) in the system, m^3/kg
$\Delta P_f(i,j)$ = pressure loss due to friction between two locations (i and j) in the system, Pa
f = Moody's friction factor, an empirical function of the Reynolds number and the pipe roughness, nondimensional
$L_e(i-j)$ = equivalent length of pipe, valves, and fittings between two locations i and j in the system, m
D = pipe internal diameter (i.d.), m

An example best illustrates the method.

Example 55.1

A piping system is designed to provide 2.0 m^3/sec of water (Q) to a discharge header at a pressure of 200 kPa. Water temperature is 20°C. Water viscosity is 0.0013 N·sec/m^2. Pipe roughness is 0.05 mm. The gravity constant (g) is 9.81 m/sec^2. Water suction is from a reservoir at atmospheric pressure (101.3 kPa). The level of the water in the reservoir is assumed to be at elevation 0.0 m. The pump will be located at elevation 1.0 m. The discharge header is at elevation 50.0 m. Piping from the reservoir to the pump suction flange consists of the following:

1 20 m length of 1.07 m i.d. steel pipe
3 90° elbows, standard radius
2 gate valves
1 check valve
1 strainer

Piping from the pump discharge flange to the discharge header inlet flange consists of the following:

1 100 m length of 1.07 m i.d. steel pipe
4 90° elbows, standard radius
1 gate valve
1 check valve

Determine the "total developed head," H_p (m), required of the pump.

Solution:

Let location (1) be the surface of the reservoir, the system "suction location."
Let location (2) be the inlet flange of the pump.
Let location (3) be the outlet flange of the pump.
Let location (4) be the inlet flange to the discharge header, the system "discharge location."

By energy balances

$$E_{T(1)} - \bar{v}\Delta P_f(1 - 2) = E_{T(2)}$$
$$E_{T(2)} + E_p = E_{T(3)}$$
$$E_{T(3)} - \bar{v}\Delta P_f(3 - 4) = E_{T(4)}$$

where E_p is the energy input required by the pump. When E_p is described as the potential energy equivalent of a height of liquid, this liquid height is the "total developed head" required of the pump.

$$H_p = E_p / g \text{ m}$$

where H_p = total developed head, m.
For the data given, assuming incompressible flow:

$P_1 = 101.3$ kPa $\qquad Z_2 = +1.0$ m
$v_1 = 0.001$ m^3/kg = constant $\qquad Z_3 = +1.0$ m
$Z_1 = 0.0$ m $\qquad Z_4 = +50.0$ m
$V_1 = 0.0$ m/sec $\qquad P_4 = 200$ kPa

A_p = internal cross sectional area of the pipe, m^2
$V_2 = Q/A = (2.0)(4)/\pi(1.07)^2 = 2.22$ m/sec
Assume $V_3 = V_4 = V_2 = 2.22$ m/sec
Viscosity (μ) = 0.0013 N · sec/m^2
Reynolds number = $D\,V/v\mu$
$= (1.07)(2.22)/(0.001)(0.0013) = 1.82 \times 10$
Pipe roughness (ϵ) = 0.05 mm
$\epsilon/D = 0.05/(1000)(1.07) = 0.000047$
From Moody's chart, $f = 0.009$ (see references on fluid mechanics)

From tables of equivalent lengths (see references on fluid mechanics):

Fitting	Equivalent Length, L_e (m)
Elbow	1.6
Gate valve (open)	0.3
Check valve	0.3
Strainer	1.8

$$
\begin{aligned}
L_e(1\text{–}2) &= 20 + (3)(1.6) + 2(0.3) + 0.3 + 1.8 = 27.5 \text{ m} \\
L_e(3\text{–}4) &= 100 + (4)(1.6) + 0.3 + 0.3 = 107.0 \text{ m} \\
\bar{v}\Delta P(1\text{–}2) &= (0.009)(27.5)(2.22)^2/(2)(1.07) = 0.57 \text{ J/kg} \\
\bar{v}\Delta P(3\text{–}4) &= (0.009)(107.0)(2.22)^2/(2)(1.07) = 2.21 \text{ J/kg} \\
E_{T(1)} &= P_1 v_1 + Z_1 g + V_1^2/2 \\
&= (101{,}300)(0.001) + 0 + 0 = 101.30 \text{ J/kg} \\
E_{T(2)} &= E_{T(1)} - \bar{v}\Delta P_f(1\text{–}2) \\
&= 101.3 - 0.57 = 100.7 \text{ J/kg} \\
E_{T(4)} &= P_4 v_4 + Z_4 g + V_4^2/2 \\
&= (200{,}000)(0.001) + (50.0)(9.81) + (2.22)^2/2 \\
&= 692.9 \text{ J/kg} \\
E_{T(3)} &= E_{T(4)} + \bar{v}\Delta P_f(3\text{–}4) \\
&= 692.9 + 2.21 = 695.1 \text{ J/kg} \\
E_p &= E_{T(3)} - E_{T(2)} \\
&= 695.1 - 100.7 = 594.4 \text{ J/kg} \\
H_p = E_p/g &= 594.4/9.81 = 60.6 \text{ m of water}
\end{aligned}
$$

It is seen that a pump capable of providing 2.0 m^3/sec flow with a developed head of 60.6 m of water is required to meet the demands of this fluid system.

55.2.3 Total Developed Head of a Fan

The procedure for finding the total developed head of a fan is identical to that described for a pump. However, the fan head is commonly expressed in terms of a height of water instead of a height of the gas being moved, since water manometers are used to measure gas pressures at the inlet and outlet of a fan. Consequently,

$$H_{fw} = (\rho_g/\rho_w)H_{fg}$$

where H_{fw} = developed head of the fan, expressed as a head of water, m
H_{fg} = developed head of the fan, expressed as a head of the gas being moved, m
ρ_g = density of gas, kg/m^3
ρ_w = density of water in manometer, kg/m^3

As an example, if the head required of a fan is found to be 100 m of air by the method described in Section 55.2.2, the air density is 1.21 kg/m^3, and the water density in the manometer is 1000 kg/m^3, then the developed head, in terms of the column of water, is

$$H_{fw} = (1.21/1000)(100) = 0.121 \text{ m of water}$$

In this example the air is assumed to be incompressible, since the pressure rise across the fan was small (only 0.12 m of water, or 1177 Pa).

55.2.4 Engineering Data for Pressure Loss in Fluid Systems

In practice, only rarely will an engineer have to apply the D'Arcy equation to determine pressure losses in fluid systems. Tables and figures for pressure losses of water, steam, and air in pipe and duct systems are readily available from a number of references. (See Figs. 55.1 and 55.2.)

55.2.5 Systems Head Curves

A systems head curve is a plot of the head required by the system for various flow rates through the system. This plot is necessary for analyzing system performance for variable flow application and is desirable for pump and fan selection and system analysis for constant flow applications.

The curve to be plotted is H versus Q, where

$$H = [E_{T(3)} - E_{T(2)}]/g \tag{55.3}$$

Assume that $V_1 = 0$ and $V = V_4$ in Eqs. (55.1) and (55.2), and letting $V = Q/A$, then Eq. (55.3) reduces to

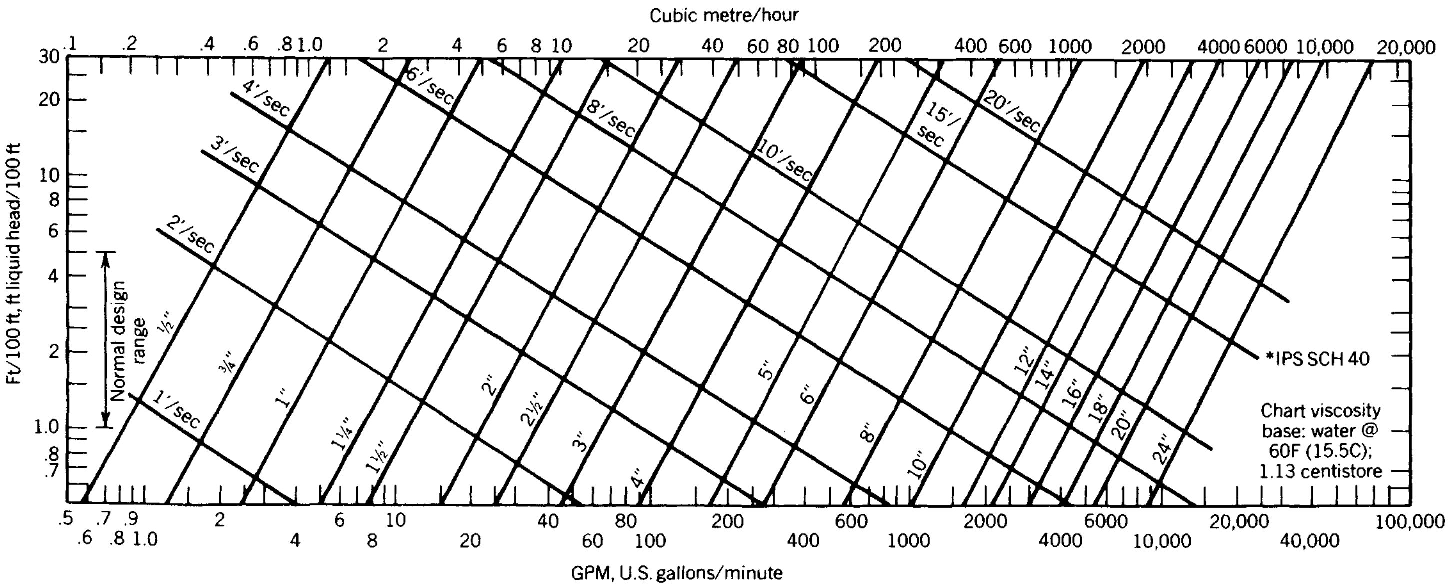

Fig. 55.1 Friction loss for water in commercial steel pipe (schedule 40). (Courtesy of American Society of Heating, Refrigerating and Air Conditioning Engineers.)

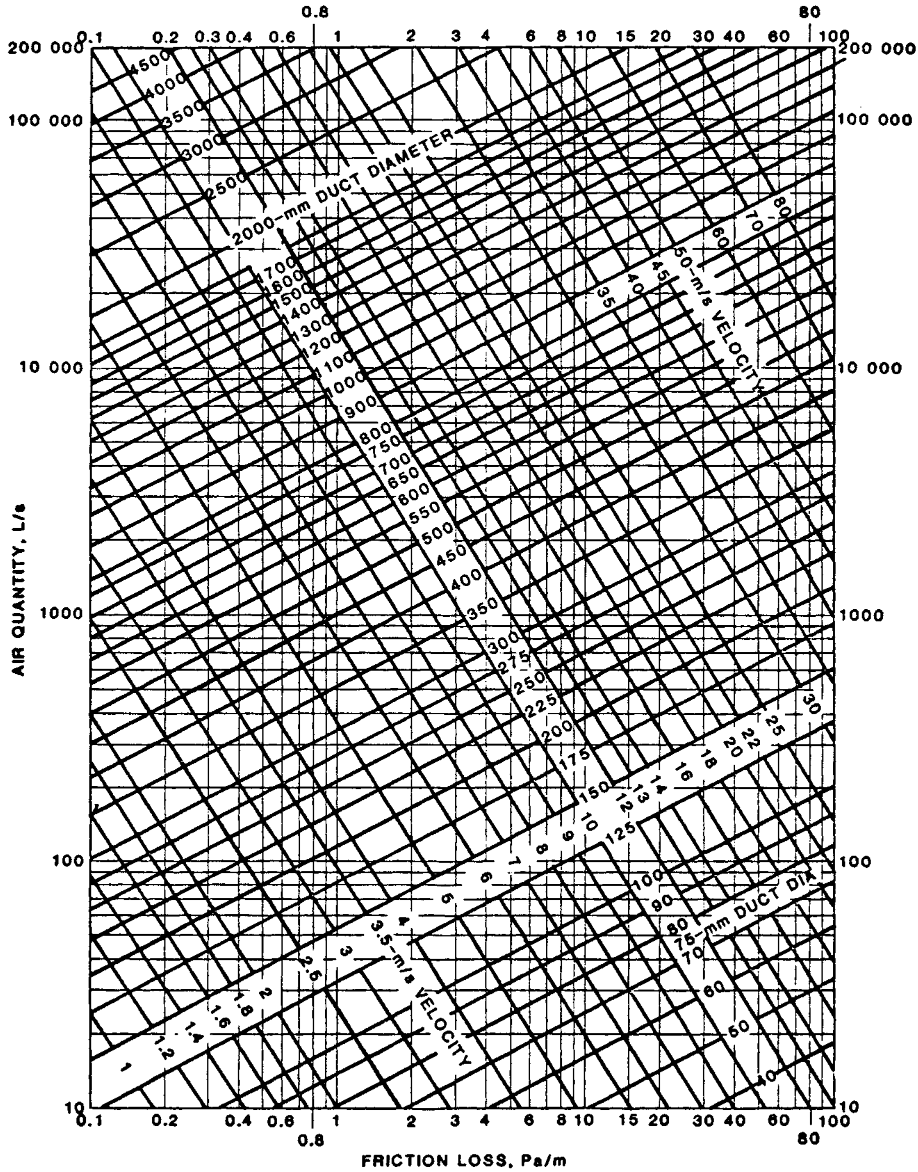

Fig. 55.2 Friction loss of air in straight ducts. (Courtesy of American Society of Heating, Refrigerating and Air Conditioning Engineers.)

$$H = K_1 + K_2Q^2 \tag{55.4}$$

where

$$K_1 = (P_4v_4/g + Z_4) - (P_1v_1/g + Z_1)$$
$$K_2 = [fL_e(1\text{–}4)A^2Dg + 1/A^2g](0.5)$$

However, K_2 is more easily calculated from

$$K_2 = (H - K_1)/Q^2$$

since both H and Q are known from previous calculations.

For example 55.1:

$$\begin{aligned} K_1 &= (200{,}000)(0.001)/9.81 + 50 - (101{,}300)(0.001)/9.81 + 0 \\ &= 60.0 \text{ m} \\ K_2 &= (60.6 - 60.0)/(7200)^2 = 0.012 \times 10^{-6} \text{ hr}^2/\text{m} \end{aligned}$$

A plot of this curve [Eq. (55.4)] would show a shallow parabola displaced from the origin by 60.0 m. (This will be shown in Fig. 55.10. Its usefulness will be discussed in Sections 55.6 and 55.7.)

55.3 CHARACTERISTICS OF ROTATING FLUID MACHINES

55.3.1 Energy Transfer in Rotating Fluid Machines

Most pumps and fans are of the rotating type. In a centrifugal machine the fluid enters a rotor at its eye and is accelerated radially by centrifugal force until it leaves at high velocity. The high velocity is then reduced by an area increase (either a volute or diffuser ring of a pump, or scroll of a fan) in which, by Bernoulli's law, the pressure is increased. This pressure rise causes only negligible density changes, since liquids (in pumps) are nearly incompressible and gases (in fans) are not compressed significantly by the small pressure rise (up to 0.5 m of water, or 5000 Pa, or 0.05 bar) usually encountered. For fan pressure rises exceeding 0.5 m of water, compressibility effects should be considered, especially if the fan is a large one (above 50 kW).

The principle of increasing a fluid's velocity, and then slowing it down to get the pressure rise, is also used in mixed flow and axial flow machines. A mixed flow machine is one where the fluid acceleration is in both the radial and axial directions. In an axial machine, the fluid acceleration is intended to be axial but, in practice, is also partly radial, especially in those fans (or propellors) without any constraint (shroud) to prevent flow in the radial direction.

The classical equation for the developed head of a centrifugal machine is that given by Euler:

$$H = (C_{t_2}U_2 - C_{t_1}U_1)/g \quad \text{m} \tag{55.5}$$

where H is the developed head, m, of fluid in the machine; C_t is the tangential component of the fluid velocity C in the rotor; subscript 2 stands for the outer radius of the blade, r_2, and subscript 1 for the inner radius, r_1, m/sec; U is the tangential velocity of the blade, subscript 2 for outer tip and subscript 1 for the inner radius; and U_2 is the "tip speed," m/sec. The velocity vector relationships are shown in Fig. 55.3.

The assumptions made in the development of the theory are:

1. Fluid is incompressible
2. Angular velocity is constant
3. There is no rotational component of fluid velocity while the fluid is between the blades, that is, the velocity vector W exactly follows the curvature of the blade
4. No fluid friction

The weakness of the third assumption is such that the model is not good enough to be used for design purposes. However, it does provide a guidepost to designers on the direction to take to design rotors for various head requirements.

If it is assumed that C_{t_1} is negligible (and this is reasonable if there is no deliberate effort made to cause prerotation of the fluid entering the rotor eye), then Eq. (55.5) reduces to

$$gH = \pi^2N^2D^2 - NQ\cot(\beta/b) \tag{55.6}$$

where Q = the flow rate, m^3/sec
D = the outer diameter of the rotor, m
b = the rotor width, m
N = the rotational frequency, Hz

55.3.2 Nondimensional Performance Characteristics of Rotating Fluid Machines

Equation (55.6) can also be written as

$$(H/N^2D^2) = \pi^2/g - [D\cot(\beta/gb)](Q/ND^3) \tag{55.7}$$

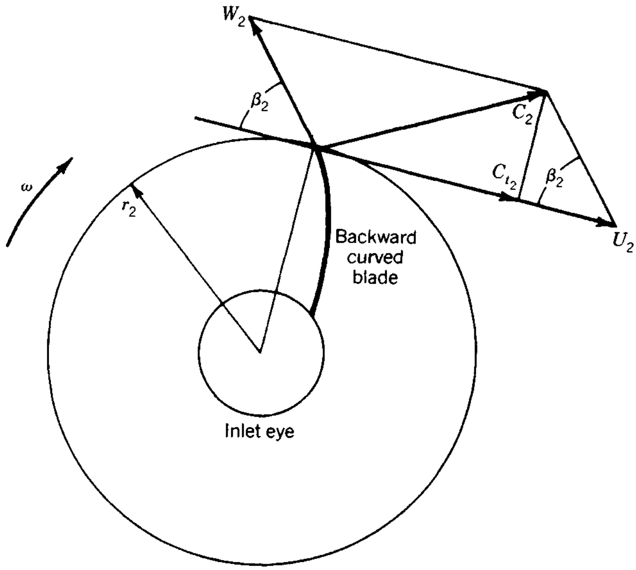

Fig. 55.3 Relationships of velocity vectors used in Euler's theory for the developed head in a centrifugal fluid machine; *W* is the fluid's velocity with respect to the blade; β is the blade angle, ω is the angular velocity, 1/sec.

In Eq. (55.7) H/N^2D^2 is called the "head coefficient" and Q/ND^3 is the "flow coefficient." The theoretical power, P (W), to drive the unit is given by $P = QgH$, and this reduces to

$$(P/\rho N^3D^5) = (\pi^2)\,(Q/ND^3) - [D\cot(\beta/b)](Q/ND^3)^2 \tag{55.8}$$

where $P/\rho N^3D^5$ is called the "power coefficient." Plots of Eqs. (55.7) and (55.8) for a given D/b ratio are shown in Fig. 55.4.

Analysis of Fig. 55.4 reveals that:

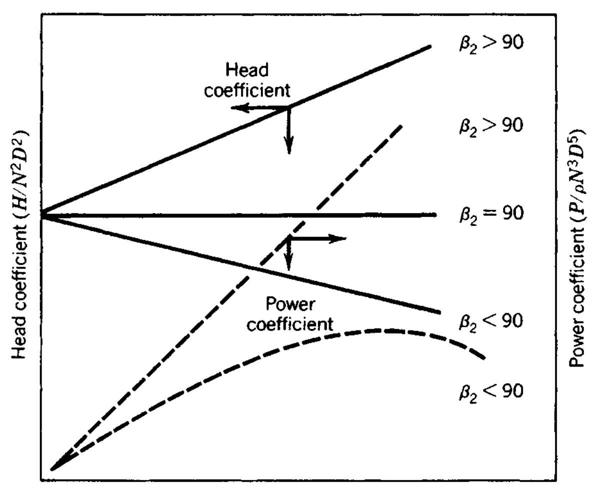

Fig. 55.4 Theoretical (Euler's) head and power coefficients plotted against the flow coefficient for constant D/b ratio and for values of $\beta < 90°$, equal to 90°, and $>90°$.

1. For a given Q, N, and D, the developed head increases as β gets larger, that is, as the blade tips are curved more into the direction of rotation
2. For a given N and D, the head either rises, stays the same, or drops as Q increases, depending on the value of β
3. For a given N and D, the power required continuously increases as Q increases for β's of 90° or larger, but has a peak value if β is less than 90°

The practical applications of these guideposts appear in the designs offered by the fluid machine industry. Although there is a theoretical reason for using large values of β, there are practical reasons why β must be constrained. For liquids, β's cannot be too large or else there will be excessive turbulence, vibration, and erosion. Blades in pumps are always backward curved ($\beta < 90°$). For gases, however, β's can be quite large before severe turbulence sets in. Blade angles are constrained for fans not only by the turbulence but also by the decreasing efficiency of the fan and the negative economic effects of this decreasing efficiency. Many fan sizes utilize β's $> 90°$.

One important characteristic of fluid machines with blade angles less than 90° is that they are "limit load"; that is, there is a definite maximum power they will draw regardless of flow rate. This is an advantage when sizing a motor for them. For fans with radial (90°) or forward curved blades, the motor size selected for one flow rate will be undersized if the fan is operated at a higher flow rate. The result of undersizing a motor is overheating, deterioration of the insulation, and, if badly undersized, cutoff due to overcurrent.

55.3.3 Importance of the Blade Inlet Angle

While the outlet angle, β_2, sets the head characteristic the inlet angle, β_1, sets the flow characteristic, and by setting the flow characteristic, β_1 also sets the efficiency characteristic.

The inlet vector geometry is shown in Fig. 55.5.

If the rotor width is b at the inlet and there is no prerotation of the fluid prior to its entering the eye (i.e., $C_{t_1} = 0$), then the flow rate into the vector is given by $Q = D_1\ b_1\ c_1$ and β_1 is given by:

$$\beta_1 = \arctan(C_1/U_1) = \tan^{-1}(Q/ND_1^3)(D_1/b_1)(1/\pi^2) \tag{55.9}$$

It is seen that β_1 is fixed by any choice of Q, N, D, and b_1. Also, a machine of fixed dimensions (D_1b_1, β_1) and operated at one angular frequency (N) is properly designed for only one flow rate, Q. For flow rates other than its design value, the inlet geometry is incorrect, turbulence is created, and efficiency is reduced. A typical efficiency curve for a machine of fixed dimensions and constant angular velocity is shown in Fig. 55.6.

A truism of all fluid machines is that they operate at peak efficiency only in a narrow range of flow conditions (H and Q). It is the task of the system designer to select a fluid machine that operates at peak efficiency for the range of heads and flows expected in the operation of the fluid system.

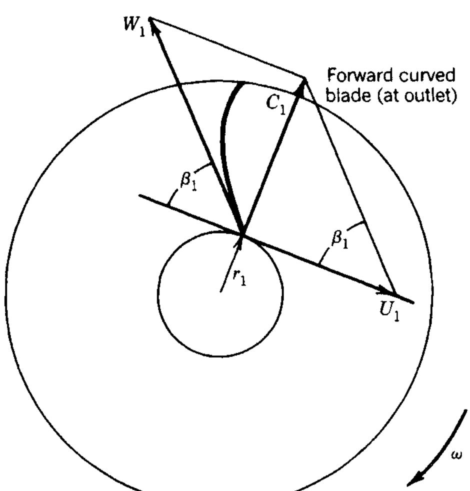

Fig. 55.5 Relationship of velocity vectors at the inlet to the rotor. Symbols are defined in Section 55.3.1.

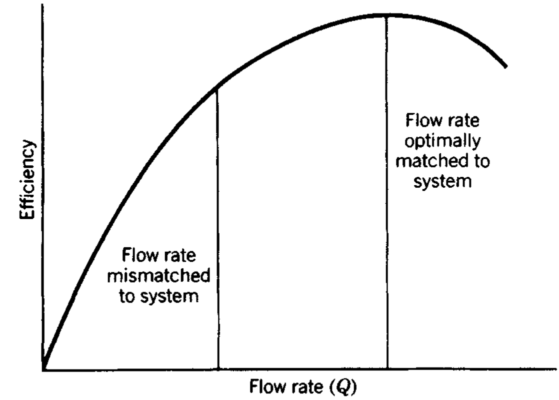

Fig. 55.6 Typical efficiency curve for fluid machines of fixed geometry and constant angular frequency.

55.3.4 Specific Speed

Besides the flow, head, and power coefficients, there is one other nondimensional coefficient that has been found particularly useful in describing the characteristics of rotating fluid machines, namely the specific speed N_s. Specific speed is defined as $NQ^{0.5}/H^{0.75}$ at peak efficiency. It is calculated by using the Q and H that a machine develops at its peak efficiency (i.e., when operated at a condition where its internal geometry is exactly right for the flow conditions required). The specific speed coefficient has usefulness when applying a fluid machine to a particular fluid system. Once the flow and head requirements of the system are known, the best selection of a fluid machine is that which has a specific speed equal to $NQ^{0.5}/H^{0.75}$, where the N, Q, and H are the actual operating parameters of the machine.

Since the specific speed of a machine is dependent on its structural geometry, the physical appearance of the machine as well as its application can be associated with the numerical value of its specific speed. Figure 55.7 illustrates this for a variety of pump geometries. The figure also gives approximate efficiencies to be expected from these designs for a variety of system flow rates (and pump sizes).

It is observed that centrifugal machines with large D/b ratios have low specific speeds and are suitable for high-head and low-flow applications. At the other extreme, the axial flow machines are suitable for low-head and large-flow applications. This statement holds for fans as well as pumps.

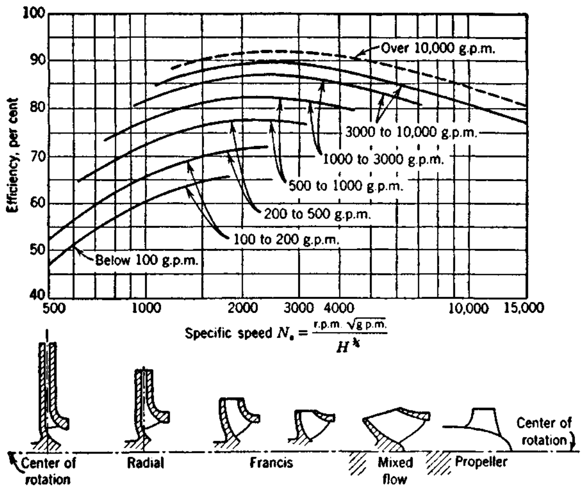

Fig. 55.7 Variation of physical appearance and expected efficiency with specific speed for a variety of pump designs and sizes. (Courtesy of Worthington Corporation.)

As an example of the use of specific speed, consider the pump application of Example 55.1. The head required was found to be 60.6 m. The flow was 7200 m^3/hr. If a pump selected for this service is to have a rotational frequency of 14.75 Hz, then it should have a (nondimensional) specific speed of

$$N_s = (14.75)(2\pi)(2.0)^{0.5}/(9.81)^{0.75}(60.6)^{0.75} = 1.089$$

Its dimensional equivalent in the English system of units (rpm, gpm, ft) is 2972. Looking at Fig. 55.7 it is seen that a pump for this service would be of the centrifugal type, with an impeller that is wide and not very large in diameter. It is a large pump (31,700 gpm) and its efficiency is expected to be high (90%).

Assuming an efficiency of 90%, then the power requirement (P) would be

$$\begin{aligned} P &= \rho QgH/\text{eff} = (1000)(2.0)(9.81)(60.6/(0.9)(1000) \\ &= 1321 \text{ kW (or 1770 hp)} \end{aligned}$$

55.3.5 Modeling of Rotating Fluid Machines

A "family" of fluid machines is one in which each member has the same geometric proportions (and physical appearance) as every other member, except for overall size. The largest member is merely a blown-up version of the smallest member.

Since the geometric proportions of each are the same, all members of a family have the same specific speed. They also have (theoretically) the same performance characteristics (Q, H, P, N) when the performance characteristics are expressed nondimensionally. Practically, the performance characteristics between members of a family differ slightly owing to changes in clearance distances, relative roughness, and Reynolds number that occur between sizes. These differences are called "secondary effects."

Ignoring secondary effects (and structural effects such as vibrations) the performance of an as yet unbuilt, large prototype can be predicted from tests on a small-scale model. Assume that the test data on a pump model, expressed nondimensionally, are as given in Fig. 55.8. It can be assumed that these results will be identical to those obtained on the prototype. If the prototype is to have a diameter of 0.81 m and a rotational frequency of 14.75 Hz, then, at peak efficiency, it can be predicted that the prototype will have the following flow, head, and power characteristics:

$$\begin{aligned} (Q/ND^3)_{\text{prototype}} &= (Q/ND^3)_{\text{model}} \\ Q_p &= (Q/ND^3)_{\text{model}}\,(ND^3)_{\text{prototype}} \\ &= (0.0406)(2\pi)(14.75)(0.81)^3 = 2.0 \text{ m}^3/\text{sec} \\ (H/N^2D^2)_{\text{prototype}} &= (H/N^2D^2)_{\text{model}} \\ H_p &= (H/N^2D^2)_{\text{model}}\,(N^2D^2)_{\text{prototype}} \\ &= (0.1054)(2\pi)^2(14.75)^2(0.81)^2/9.81 = 60.6 \text{ m} \\ P &= \rho QgH/\mathit{eff} = \mathit{1321\ kW\ (from\ the\ previous\ section)} \end{aligned}$$

If the model had a diameter of 0.1 m and a rotational frequency of 29 Hz, then, at peak efficiency, its flow, head, and power were:

$$\begin{aligned} Q_m &= (Q/ND^3)(ND^3) = (2.0)(29/14.75)(0.1/0.81)^3 \\ &= 0.0074 \text{ m}^3/\text{sec} \\ H_m &= (H/N^2D^2)p(N^2D^2)m = 60.6\ (29/14.75)^2(0.1/0.81)^2 \\ &= 3.57 \text{ m} \\ P_m &= (1000)(0.0074)(9.81)(3.57)/(0.9)(1000) = 0.29 \text{ kW} \end{aligned}$$

Manufacturers of fluid machines often do not have facilities large enough (fluid quantities and power) to test their largest products. Consequently, the performance of such large machines is estimated from model tests.

55.3.6 Summary of Modeling Laws

Neglecting secondary effects (changes in Reynolds number, size, and clearance distances) the nondimensional performance relationships between a model and a prototype, for any single point of operation (i.e., one point on a common nondimensional curve of performance characteristics), can be summarized as follows:

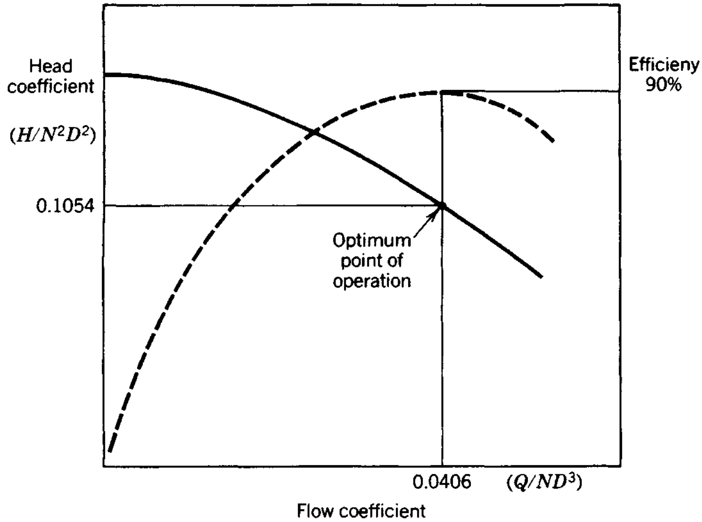

Fig. 55.8 Performance characteristics of a model pump, expressed nondimensionally.

Nondimensional Characteristic	Model	Prototype
Flow coefficient	Q/ND^3	$= Q/ND^3$
Head coefficient	H/N^2D^2	$= H/N^2/D^2$
Power coefficient	$P/\rho N^3D^5$	$= P/\rho N^3D^5$
Efficiency	eff	= eff
Specific speed	$NQ^{0.5}/H^{0.75}$	$= NQ^{0.5}/H^{0.75}$

The same relationships can be used to determine the changes in flow, head, or power of a single-fluid machine whenever its diameter or angular frequency is changed in an unchanging fluid system. For a machine of constant diameter, the flow, head, and power will vary with angular frequency as follows:

$$Q \propto N$$
$$H \propto N^2$$
$$P \propto N^3$$

55.4 PUMP SELECTION

55.4.1 Basic Types: Positive Displacement and Centrifugal (Kinetic)

Positive displacement pumps are best suited for systems requiring high heads and low flow rates, or for use with very viscous fluids. The common types are reciprocating (piston and cylinder) and rotary (gears, lobes, vanes, screws). Centrifugal pumps are well suited for the majority of pumping services. The common types are radial, centrifugal, mixed flow, and propellor (axial flow).

55.4.2 Characteristics of Positive Displacement Pumps

Some advantages of positive displacement pumps, besides their inherent ability to provide high discharge pressures at low flow rates, are: ability to provide metered quantities of fluid at a wide range of viscosities; can handle non-Newtonian fluids (sludge, syrup, mash); can operate at slow speeds. Some disadvantages are: flow is pulsating; costs (initial and maintenance) are higher than for centrifugals; must have pressure relief valves in the discharge piping; tight seals and close tolerances are essential to prevent leak-back.

Overall efficiencies usually vary with pump size, being lowest (50%) for small pumps (2 kW) and highest (90%) for large pumps (250 kW). Efficiencies do not vary significantly with flow rate.

Pulsating flows of reciprocating pumps can be smoothed out somewhat by installing air chambers in the discharge. The volume of the air chamber should be recommended by the manufacturer, but is approximately three or four times the displacement volume of the piston. Pulsating flows can be further smoothed out by using double-acting reciprocating pumps that discharge fluid at both ends of the stroke. Rotary pumps have smooth flows.

55.4.3 Characteristics of Centrifugal Pumps

Centrifugal pumps are used in most pumping services. They can deliver small to large flow rates and operate against pressures up to 3000 psi when several impellers are staged in series. They do not work well on highly viscous or non-Newtonian fluids; they operate at high speeds; flow is smooth; clearances between impeller tip and casing are not critical; they do not develop dangerously high head pressures when the discharge valve is closed; and their initial and maintenance costs are lower than that for positive displacement pumps.

Efficiencies of centrifugal pumps are about the same as their corresponding-sized positive displacement pumps if they are carefully matched to their systems. However, their efficiencies vary significantly with flow rate when operated at constant speed, and their efficiencies can be very poor if mismatched to their system, as seen in Figs. 55.6 and 55.8.

55.4.4 Net Positive Suction Head (NPSH)

The liquid static pressure at the suction of both positive displacement and centrifugal pumps must be higher than the liquid's vapor pressure to prevent vaporization at the inlet. Vaporization at the inlet, called "cavitation," causes a drop in developed head, and, in severe cases, a complete loss of flow. Cavitation also causes pitting of the impeller that, in time and if severe enough, can destroy the impeller.

Net positive suction head (NPSH) is the difference between the static pressure and the vapor pressure of the liquid at the pump inlet flange, expressed in meters:

$$\text{NPSH} = (P_s - P_v)/\rho g \quad \text{m} \tag{55.10}$$

where P_s is the static pressure at the pump inlet flange, Pa; P_v is the liquid's vapor pressure, Pa; and ρ is the liquid density, kg/m^3.

There are two NPSHs that a system designer must consider. One is the NPSH available (NPSHA), which is dependent on the design of the piping system (most importantly the relative elevations of the pump and the source of liquid being pumped). The second is the NPSH required (NPSHR) by the pump selected for the service. There is a static pressure loss within the pump as the liquid passes through the inlet casing and enters the blades. The severity of this loss is dependent on the design of the casing and the amount of acceleration (and turbulence) that the liquid experiences as it enters the blading. Manufacturers test for the NPSHR for each model of pump and report these requirements on the engineering performance specification sheets for the model. The task of the system designer is to ensure that the NPSHA exceeds the NPSHR. Using the data in Example 55.1, the NPSHA is calculated as follows:

P_v at 20°C = 2237 Pa

P_s is found from the calculation for the total energy at the pump suction flange, $E_{t(2)}$

$E_{t(2)} = 100.7$ J/kg $= (P_s/\rho + Zg + V^2/2)$ at (2)

$P_s = (100.7)(1000) - (1)(9.81)(1000) - (2.21)^2(1000)/2 = 88{,}400$ Pa

$\text{NPSHA} = (P_s - P_v)/\rho g = (88{,}400 - 2{,}237)/(9.81)(1000) = 8.8$ m

This NPSHA (8.8 m) is considered large and quite adequate for most pump models. However, if, after a survey of available pumps, it is found that none can operate with this net positive suction head, then the design of the piping system will have to be changed: the pump will have to be placed at a lower elevation to ensure adequate suction static pressure.

55.4.5 Selection of Centrifugal Pumps

The pump selected for a fluid system must deliver the specified flow and required head at or near the pump's maximum efficiency, and have a NPSHR less than the NPSHA. However, only rarely will one find a pump model, even from a survey of several manufacturers, that exactly matches the system; that is, a pump whose flow and head at maximum efficiency exactly match the flow and head required.

The first step in pump selection is to contact several pump manufacturers and obtain the performance curves of the pumps they recommend for the specified service. A typical pump curve is shown in Fig. 55.9.

It is seen that on this one curve data are presented giving flow, head, efficiency, power, and NPSHR for a variety of impeller sizes (diameters). The curves for impeller sizes in between those shown can be estimated by extrapolation. Since clearance distances between the impeller tip and the

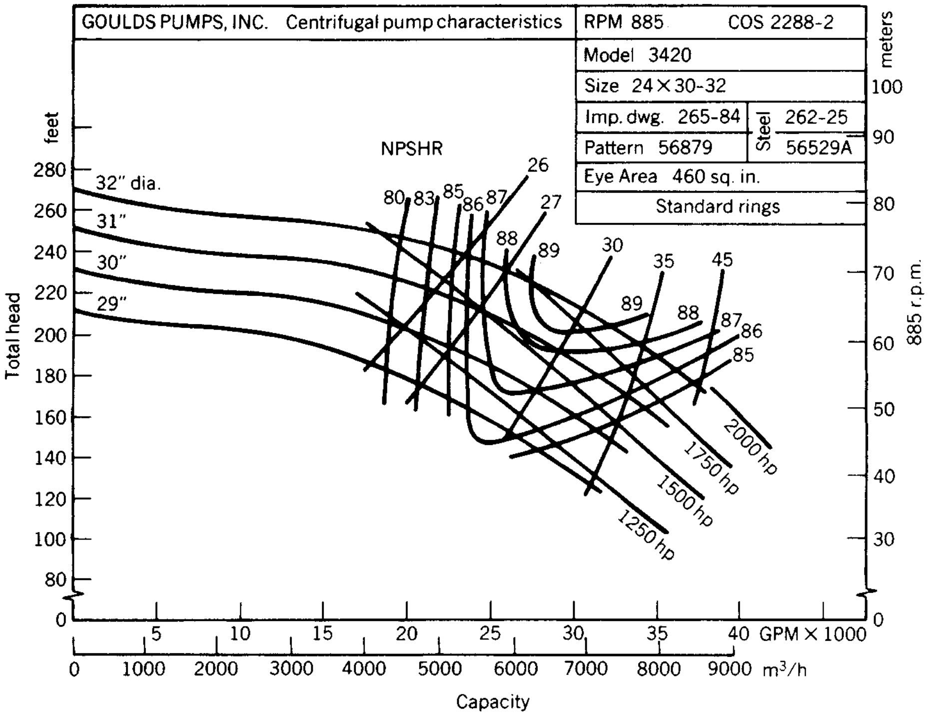

Fig. 55.9 Example of pump curve provided by manufacturer. (Courtesy Goulds Pumps, Inc., Seneca Falls, NY.)

pump casing are not critical, it is possible to install any of several different size impellers in one casing. It is also possible to cut down an existing impeller to a smaller size if it is found advantageous to do so after delivery of a pump and installation in its system.

An important selection parameter is the motor size. Note that there is a maximum power that the pump will require regardless of flow. It is advisable to specify a motor with a power rating at least equal to this maximum power required, since, in most applications, there will be times when the pump is called upon to deliver higher flow rates than originally expected.

For the pump in Example 55.1, the purchase specifications would be:

Flow	7200 m^3/hr (2.0 m^3/sec)
Head	60.6 m
NPSHA	8.8 m (28.9 ft)

It is seen, in Fig. 55.9, that the 32 in.-diameter impeller in the model 3420 would be adequate for the head and flow. However, the NPSHR is 10.29 m (33 ft), which is more than is available, and therefore not acceptable. The efficiency is 89%, which is close to what was expected. The usual procedure is to survey more manufacturers in hopes of finding a better match, a higher efficiency, and one requiring less NPSH. If the model 3420 were finally selected, it is recommended that the 2000 hp (1492 kW) motor be specified. Also, the piping system would have to be altered to lower the pump elevation and provide more NPSHA.

Referring again to Fig. 55.9, it is seen that the size is given by the numbers 24 × 30–32. It is standard practice in the industry to use a size designation number that gives, in order, the diameter of the discharge flange, the diameter of the suction flange, and then the impeller diameter, all in inches.

55.4.6 Operating Performance of Pumps in a System

The actual point of operation (head and flow) of a pump in a piping system is found from the intersection of the pump curve (Fig. 55.9) and the system head curve [Eq. (55.4)]. Both curves represent the energy required to cause a specified flow rate. By the law of conservation of energy the energy input to the fluid by the pump must equal the energy required by the piping system for a specified flow. Figure 55.10 shows both curves plotted on the same coordinate system and establishes the point of operation of the pump in the system.

The actual point of operation of the model 3420 pump and the system of Example 55.1 is 7450 m^3/hr at 60.7 m. If this flow is too large, the system will have to be throttled (by closing in a valve) until the flow is reduced to the desired value. If the system is throttled to 7200 m^3/hr, the actual developed head will be 63.1 m. This means the head loss created in the valve, at 7200 m^3/hr, is 63.1–60.6 or 2.5 m. The power wasted in this throttling process is $\rho QHg/\text{eff}$ = (1000)(2)(2.5)(9.81)/(0.89)(1000) = 55.1 kW, which is converted into heat.

In large pumps (100 kW) even small differences in operating power (2%) can make large differences in operating economy. For this reason it is important for the purchaser of a pump to seek the best possible match of the pump to the system to optimize efficiency and avoid having to throttle the flow. For example, if the difference in operating power between two pumps capable of meeting a specified service (head and flow) is as small as 2 kW but the pump is operated continuously (8760 hours per year), then the energy difference is 17,520 kWhr which, at 50.05/kWhr, has a value of \$876 per year. If the additional cost (if any) of the more economical pump can be amortized over its financial lifetime for less than \$876 per year, then the better pump should be purchased.

55.4.7 Throttling versus Variable Speed Drive

If the pump is to be operated at reduced flow rates for extended periods of time, it may be economically justifiable to use a variable speed drive.

As an example, assume that the system in Example 55.1 is operated at 5000 m^3/hr for 2500 hours per year. If throttled to 5000 m^3/hr, the pump head (from Fig. 55.10) would be 73.5 m and the efficiency would be (about) 84%. The energy consumed at this point of operation would be $\rho QHgh/\text{eff}$ = (1000)(5000/3600)(93.5)(9.81)(2500)/(0.84)(1000) = 2.98 × 10^6 kWhr per year.

The operating points for the variable speed drive are determined by using the modeling laws (Section 55.3.5). If the diameter is constant, the H/N^2 and Q/N are constant and, for variable N, $H = KQ^2$, which is a parabola through the origin, as shown in Fig. 55.10. The operating points (1) and (2) on this parabola are related by the equations $H_1/N_1^2 = H_2/N_2^2$ and $Q_1/N_1 = Q_2/N_2$, where $H_2 = 60.0 + 0.012 \times 10^{-6}\,(5000)^2 = 60.3$ m. The K of the parabola in $H_2/Q_2^2 = 60.3/(5000)^2$. The intersection of this parabola with the original pump curve, point (1), is $H_1 = 72.0$ m and $Q_1 = 5400$ m^3/hr. The reduced speed $N_1 = N_2Q_2/Q_1$ = (14.75)(5000)/5400 = 13.66 Hz. The efficiency at (2), 86%, equals the efficiency at (1) since all nondimensional parameters at (1) and (2) are the same.

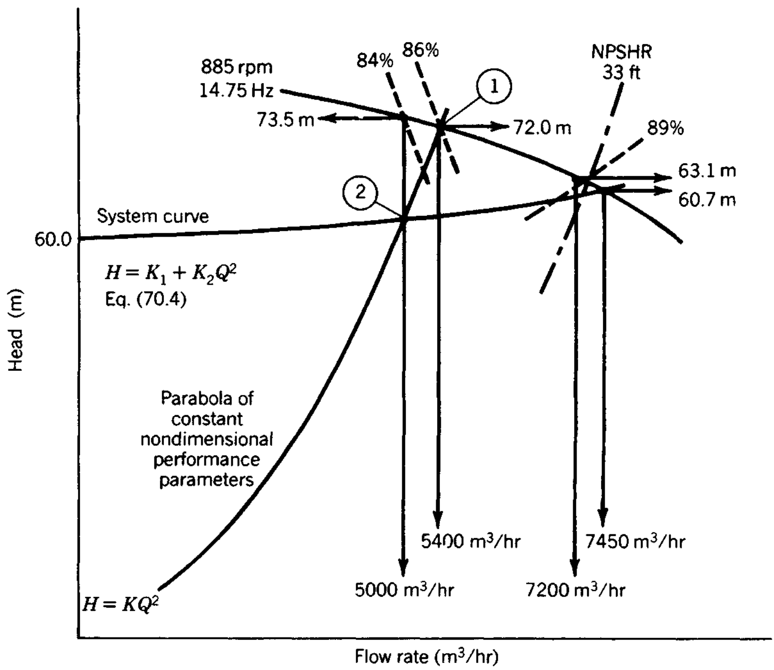

Fig. 55.10 Point of operation of a pump in a system.

The energy consumed at the reduced pump speed (13.66 Hz) to provide 5000 m^3/hr for 2500 hours per year is (1000)(5000)/(3600)(60.3)(9.81)(2500)/(0.86)(1000) = 2.38 × 10^6 kWhr per year. The saving of 600,000 kWhr, at \$0.05/kWhr, is worth \$30,000 per year. If the cost of a variable speed drive in this example can be amortized over its financial lifetime for less than \$30,000 per year, it should be purchased.

55.5 FAN SELECTION

55.5.1 Types of Fans; Their Characteristics

Fans, the same as pumps, are made in a large variety of types in order to serve a large variety of applications. There are also options in both cost and efficiency for applications requiring low power (5 kW). High-power applications require high efficiencies.

Fan types with low specific speeds (0.17) are suitable for high-head, low-flow application. These fans are usually centrifugal, with both forward and backward curved blades. Fan types with high specific speeds (16.75) are suitable for low-head, large-flow application. These fans are of the axial flow type (propellor blades). Higher heads can be achieved with axial flow fans if provision is made to recover, into head, the swirl (rotational) component of velocity imparted by the blades. Two methods of recovering this energy component are: (1) a set of fixed blades located either up- or downstream from the rotating blades (vane axial type); and (2) for maximum recovery, two sets of rotating blades, one turning in a reverse direction to the other (contrarotating propellors).

Characteristics of fans are similar to those of centrifugal pumps: they must be carefully matched to their system in order to achieve their best efficiencies; the basic modeling laws are used to predict their performance; clearances between the wheel tip and casing (cutoff) are not critical; their discharge ducts can be closed without causing high heads to develop; their flow is smooth; they can be used on gases and gas–particle mixtures (powders, dusts, lints); and their maintenance costs are low.

55.5.2 Fan Selection

The steps to follow in fan selection are the same as those for pump selection with two exceptions: (1) there is no net positive suction head to be concerned with; (2) a variety of speeds are usually available for each fan through the use of different size sheaves (using belt drives). This latter exception causes some inconvenience in determining optimum efficiency matches since the method of presenting performance data, called "multirating tables," does not include an efficiency parameter. However, the tables do list the power requirement so that a system designer can seek the best efficiency by seeking the lowest power requirement. If efficiencies are wanted, they can either be calculated or requested from the manufacturer in the form of performance curves (rather than tables).

An example best illustrates the method. Assume a fan is to be selected to exhaust 18,170 m^3/hr of air at 90°C and atmospheric pressure from a drying kiln. The design of the ductwork is such that the developed head of the fan must be 204 mm (water gauge). The task is to select the fan with the least power requirement. Multirating tables will be obtained from several manufacturers. They appear as shown in Table 55.1 for a size 60AW fan.

The data presented in multirating tables are based on an air density of 1.201 kg/m^3 (air at 760 mm mercury pressure and 21.11°C). It is easiest to adjust the system head required (at 1 atm and 90°C for the example) to an equivalent head based on the standard density (at STP) used in the tables. The relationship is: $H_{STP} = (H_{req})(21.11 + 273)(\text{mmHG})/(°\text{C} + 273)(760\text{ mm})$. Therefore $H_{STP} = (204)(294.11)(760)/(363)(760) = 165.1$ mm (6.5 in.) water gauge. The flow rate is unaffected by density changes since fans are constant volume devices. However, the power, as well as the head, is affected by density changes so that the power listed in Table 55.1 must be adjusted by the same factor that was used to adjust the head (0.81). Efficiency is independent of density, and 18,170 m^3/hr is 10,000 cfm (cubic feet per minute) From the data in Table 55.1, one selection of fan would be the size 60AW operated at 823 rpm (13.72 Hz). The power required would be (15.46)(0.81) hp (9.34 kW). The efficiency can be calculated from the STP data by eff = $\rho QHg/P$ = (1000)(18,170)(0.1651)(9.81)/(11.5)(1000)(3600) = 0.71, where the density of water in the gauge is assumed to be 1000 kg/m^3.

The multirating tables of different size fans of the same manufacturer as well as those of other manufacturers should be surveyed to find the one with the lowest power requirement. As an example, for the manufacturer of the fan in Table 55.1, the power requirement and efficiency of other sizes, all of which meet the head and flow requirements, are as follows:

Model	Size Wheel (m)	Power (kW)	Efficiency
45AW	0.83	11.48	0.58
50AW	0.92	10.43	0.64
55AW	1.01	9.67	0.69
60AW	1.10	9.34	0.71
70AW	1.29	8.57	0.77
80AW	1.47	9.00	0.74
90AW	1.66	9.76	0.68

Table 55.1 Example of Multirating Table for Fans[a]

Capacity (cfm)	Outlet Velocity (fpm)	5½″ S.P. rpm	5½″ S.P. bhp	6″ S.P. rpm	6″ S.P. bhp	6½″ S.P. rpm	6½″ S.P. bhp
3000	921						
3700	1136	715	5.83				
4400	1351	715	6.37	746	7.10	777	7.84
5100	1566	715	6.98	747	7.73	777	8.48
5800	1781	717	7.63	748	8.41	778	9.24
6500	1996	722	8.40	752	9.21	781	10.04
7200	2211	730	9.26	759	10.11	787	10.98
7900	2426	738	10.17	766	11.07	794	12.00
8600	2641	748	11.12	776	12.12	803	13.08
9300	2856	759	12.18	786	13.21	812	14.21
10000	3071	771	13.29	798	14.37	823	15.46
10700	3286	783	14.45	809	15.60	835	16.76
11400	3501	796	15.69	822	16.87	847	18.11
12100	3716	810	16.99	836	18.28	860	19.52
12800	3931	825	18.40	850	19.71	874	21.07
13500	4146	841	19.91	865	21.27	888	22.64
14200	4361	857	21.45	881	22.92	904	24.35
14900	4576	873	23.03	896	24.60	919	26.16
15600	4791	890	24.75	913	26.34	935	27.98
16300	5006	907	26.48	930	28.22	952	29.88
17000	5221	924	28.22	947	30.10	969	31.92
17700	5436	941	30.18	963	32.00	985	33.96
18400	5651	960	32.30	981	34.13	1002	36.02
19100	5866	978	34.43	999	36.41	1020	38.31
19800	6081	997	36.87	1017	38.73	1038	40.77

[a]Courtesy of Buffalo Forge Co., Buffalo, NY.

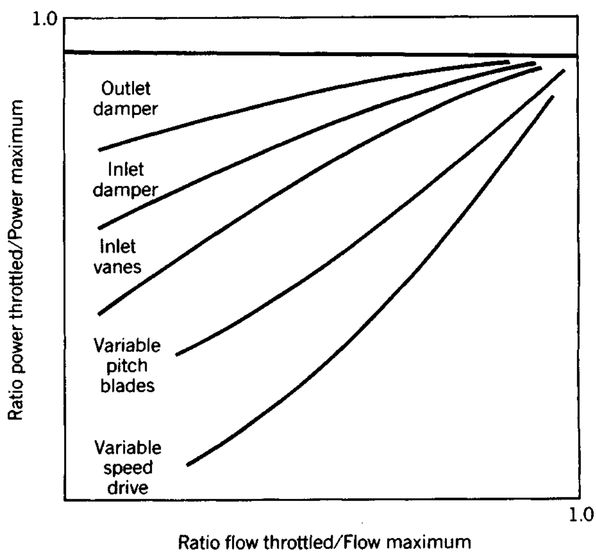

Fig. 55.11 Effectiveness of various methods of controlling fans in variable volume service.

The 70AW model is seen to be the best choice for this service.

55.5.3 Control of Fans for Variable Volume Service

Two common applications of fans requiring variable volume operation are combustion air to boilers and conditioned air to rooms in a building. Common methods of controlling air volume are outlet dampers; inlet dampers; inlet vanes (which impart a prerotation or swirl velocity to the air entering the wheel); variable pitch blades (on axial fans); and variable speed drives on the fan motor.

Figure 55.11 illustrates the effectiveness of these methods by comparing power ratios with flow ratios at reduced flows. The variable speed drive is the most effective method and, with the commercialization of solid-state motor controls (providing variable frequency and variable voltage electrical service to standard induction motors), is becoming the most popular method for fan speed control.

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