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FANS AND BLOWERS

Fan is the generic term for low pressure air- and gas-moving devices using rotary motion. Fans are subdivided into centrifugal and axial-flow types, depending on the direction of air flow through the impeller. In centrifugal fans, the air is introduced into the center of a revolving wheel or rotor with peripheral blades. Air is drawn through the blades and forced out in centrifugal flow into a scroll or volute housing where a portion of the kinetic energy is converted to pressure or static head. In axial-flow fans the air continues to move directly forward through the fan along the axis of the shaft. Kinetic energy is imparted to the air by the shape and arrangement of the blades. After discharge through the blades, although the general flow direction is still forward, a spiral component of velocity generally has been added to the air. A propeller-type fan is the most common axial-flow fan but more complicated types are in use where the blades resemble vanes in a turbine.

Blower is a term applied to a centrifugal fan generally when it is used to force air through a system under positive pressure. It generally implies a fan developing a reasonably high static pressure of at least 500 Pa (several inches of water). High speed centrifugal blowers ($\geq$3600 rpm) are also available in one or more stages to compress air to pressures of 108–150 kPa (1–7 psig). The term blower is also applied to relatively low pressure positive displacement compressors of the rotary lobe, screw, or sliding vane types where the discharge pressure is usually less than 205 kPa (15 psig). Positive displacement blowers are outside the scope of this article.

When a fan is placed at the end of a system so that most of the system pressure drop is on the suction side of the fan, it is commonly called an exhaust fan or an exhauster. This term may also be applied to a ventilating fan where the primary function is to exhaust air from a room or an open hood.

Centrifugal compressors or turbocompressors are high volume centrifugal devices capable of gas compression varying from 105 to $>$1500 kPa (0.5 to several hundred psig). These generally consist of a number of stages of alternating rotating and stationary turbine blades and turn at very high speeds (see High pressure technology).

The total pressure produced by a fan can be measured with an impact probe pointed directly upstream. The pressure so measured is a combination of both the static pressure and the kinetic energy pressure equivalent. Static pressure can be measured using a properly designed static wall tap or using the static pressure parts of a pitot tube. The latter represents the true pressure head exclusive of velocity effects. The difference between the total (impact) pressure and the static pressure is the velocity pressure or velocity head. Pressure readings are normally expressed in millimeters or inches of water (1 mm water = 9.807 Pa; 1 in. water = 248.8 Pa) and are referred to atmospheric pressure (101.3 kPa) as the reference base. Thus barometric pressure must be added to obtain absolute pressure. The total pressure rise produced by a fan is the difference in total pressure between the fan outlet and inlet. The fan static pressure is the total pressure rise for the fan reduced by the discharge velocity pressure. Inlet velocity head is assumed to be zero for fan rating purposes.

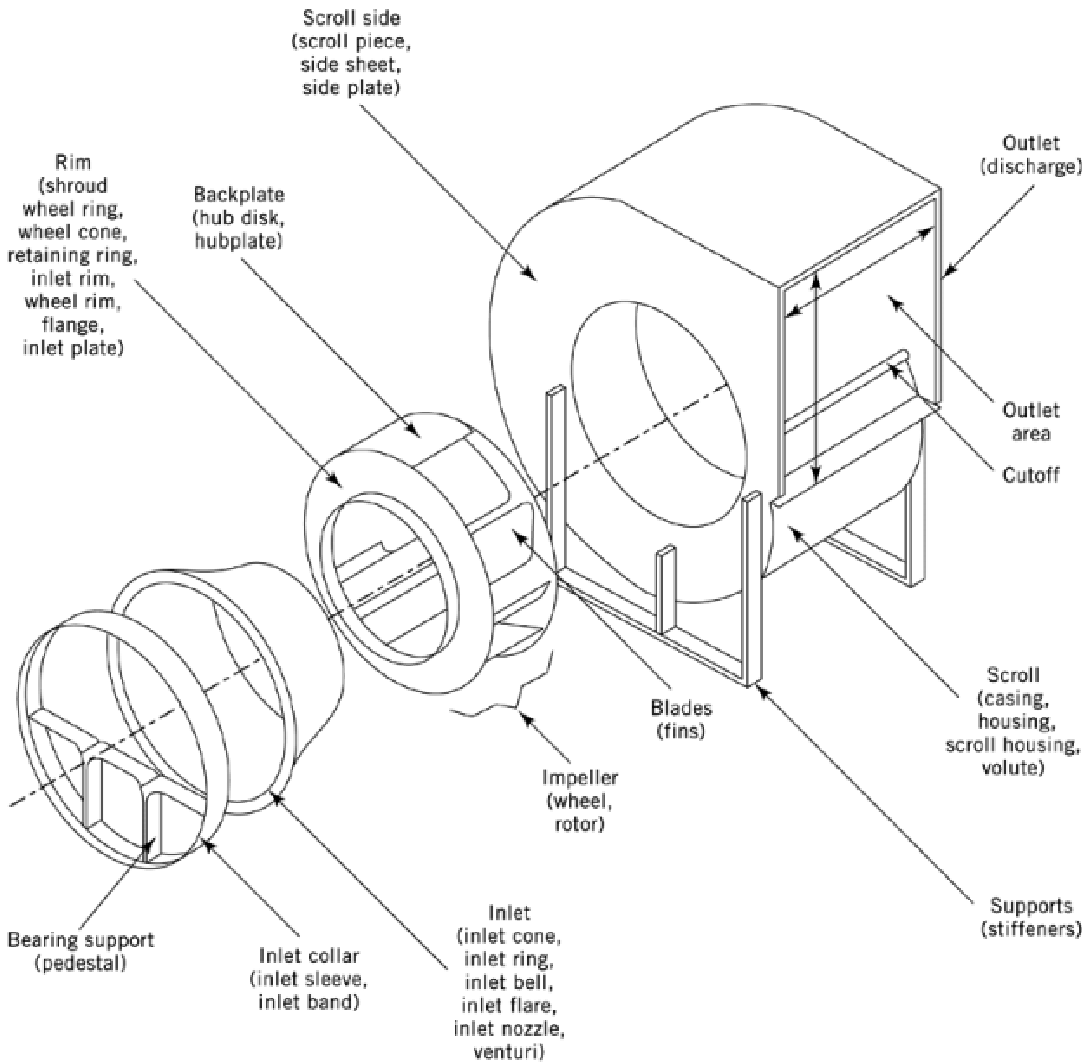

Fig. 1. Component parts of a centrifugal fan (1).

1. Centrifugal Fans

Figure 1 shows parts and names commonly associated with centrifugal fan components. The rotation of the wheel causes air between the blades to be rotated. The resulting centrifugal force causes this air to be compressed and ejected radially from the wheel. The compression results in an increase in static pressure in the fan scroll. The static pressure produced at the blade tips depends on the ratio of the velocity of air leaving the tips to the velocity of air entering at the heel of the blades. Thus the longer the blades, the greater the static pressure developed by the fan at a constant speed.

As the air leaves the blade tips, it contains kinetic energy by virtue of its velocity. The directional component of this velocity is both rotative and radial. When the fan blades are inclined forward, these components are cumulative. With backward-inclined blades, the components are in opposition. The purpose of the fan volute or scroll-shaped casing is to convert a portion of the kinetic energy of the air leaving the blades into static pressure.

Design operating efficiencies of fans under test conditions are in the range of 40–80%. Actual efficiency can be affected appreciably by the arrangement of inlet and outlet duct connections.

The air power in watts of a fan is given by equation 1:

$$\text{air power} = Q\,\Delta p \qquad (1)$$

where Q is the volume of gas handled in m^3/s, and Δp is the pressure rise across the fan, Pa. (In units of hp, the air power = $144Q'\Delta p'/33,000$; Q' is in ft^3/min; and $\Delta p'$ in psi.) In many fan installations, the velocity head of the fan discharge is wasted. In such cases, the fan static pressure may be used in equation 1 instead of the total pressure. Fan efficiency is expressed by equation 2:

$$\text{efficiency} = \frac{\text{air power}}{\text{shaft power}} \qquad (2)$$

1.1. Performance Testing

Although fan performance characteristics can be roughly estimated during the early stages of design, fan efficiency losses and slip cannot be estimated accurately from theory alone. Therefore, the exact performance characteristics of a new fan design must be determined by testing. Test conditions must be carefully controlled, such as provision for steady and uniform flow of air approaching the fan inlet, because any inlet disturbances can affect performance. For this reason, fan field tests are seldom reliable, and most testing is performed in a laboratory on a test block following procedures set forth in standards for performance (2) including sound (3).

Figure 2 illustrates one of the several available test methods (2) and a typical performance curve. Fans designed for a duct as illustrated have a section of straight discharge duct attached. Straightening vanes are provided to eliminate swirl, reduce turbulence, and aid flow equalization across the duct. Air flow is determined using a pitot traverse while the fan is operated at a constant speed. The measured pressures are corrected for duct losses back to fan outlet conditions. A fan performs in accordance with the performance curve only if there is an equivalent duct present to convert velocity head efficiently to static head. At the end of the test, the duct is blanked to measure discharge pressure and shaft power at a shutoff (no flow) condition. The opposite extreme of the curve, free delivery (equivalent to duct removal), is extrapolated from nearly wide-open conditions. Intermediate points at sufficiently close intervals to define the curve would be measured by replacing the blank at the end of the duct with restricting orifices of varying cross section.

1.2. Types and Characteristics

The four basic fan wheel and blade designs and the corresponding performance curves are illustrated in Figure. 3.

1.2.1. Forward-Curved Blades

In the forward-curved design (Fig. 3**a**), both the heel and tip of the blade are curved forward in the direction of rotation. Air leaves the tip of the wheel at a velocity greater than the wheel-tip speed. Blades are generally quite shallow and spaced much closer together than in other blade designs; 24–64 blades are typical. For a given fan duty, the wheel would have the smallest diameter and operate at the lowest speed of the various blade types. Such fans are commonly used for low pressure, high volume ventilating applications. As such, the wheel is often constructed of lightweight, low cost materials. Its mechanical efficiency is generally somewhat lower than that of the backward-curved blade fan. The pressure curve has a dip to the left of the peak which can

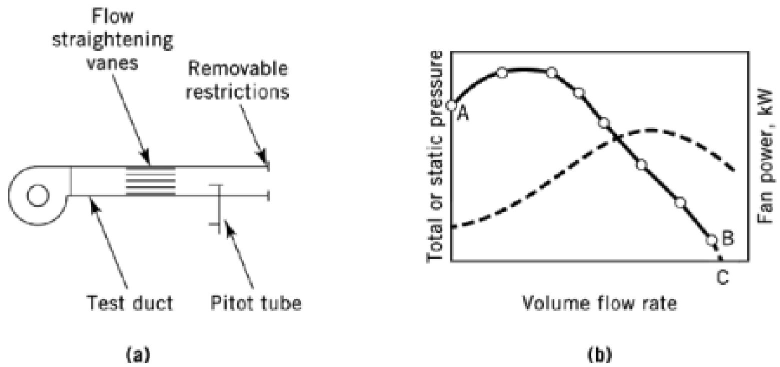

Fig. 2. Fan performance: (**a**) a typical test arrangement; (**b**) performance curve where the solid line showing data points is the pressure; the dashed line is the power. At point A the duct is blanked off, and at point B the flow is wide open; the data points in between represent progressively less restricted flow. Point C represents free delivery.

cause operating problems (fan instability). Flow control in this region is difficult. Highest efficiency is reached to the right of the pressure peak, usually at 40–50% of wide-open flow. The fan is usually operated and rated to the right of the pressure peak. Power rises continually toward free delivery, which must be considered in motor selection.

1.2.2. Backward-Curved Blades

In the backward-curved design (Fig. 3**b**), the blades incline backward (opposite to rotation direction) from the point of heel attachment on the wheel. The single-thickness blades may be either straight or curved, usually 12–16 blades to a wheel. Air leaves the blade at a velocity less than wheel-tip speed because the increasing flow passage through the blade provides for expansion of the air. This feature improves the mechanical efficiency over that of the forward-curved blade. The deep blades lend themselves to developing a high static pressure. Wheel diameter and speed are generally higher for a given performance than the forward-curved blade. Close clearance and alignment of the wheel with the inlet bell are important aspects of this design to obtain maximum efficiency, especially at high static pressures. Inaccurate clearances allow leakage of compressed air back to the suction side of the wheel. The pressure curve rises somewhat from shutoff with increase in flow until a maximum pressure is reached. The maximum efficiency is reached at 50–65% of wide-open volume. The power curve reaches a maximum near the point of peak efficiency and then tends to drop off slightly with increased flow resulting in a nonoverloading design from the standpoint of motor sizing.

1.2.3. Straight Radial Blades

The straight radial blade design is the simplest of all centrifugal fans and also the least efficient (Fig. 3**c**). However, the wheel can be designed with great mechanical strength and is easily repaired. It is useful for two different applications. For a given speed, it tends to develop a higher static pressure than other wheel designs and thus is attractive for high speed, high pressure fans compressing air to 108–120 kPa (1–3 psig). Such designs often have pressure performance curves that are fairly flat to 70% of the wide-open flow. This is desirable for applications where a constant output pressure is needed, such as for primary combustion air. Another use is in material handling. The blades can be coated with abrasion-resistant coatings or equipped with replaceable liners (see Coatings). When large objects must occasionally be handled, such as a loose bag from a bag-filter house, the rims of the wheel may be omitted entirely and the blades supported with stiff struts from the hub. The shape of the power curve leads to overloading characteristics.

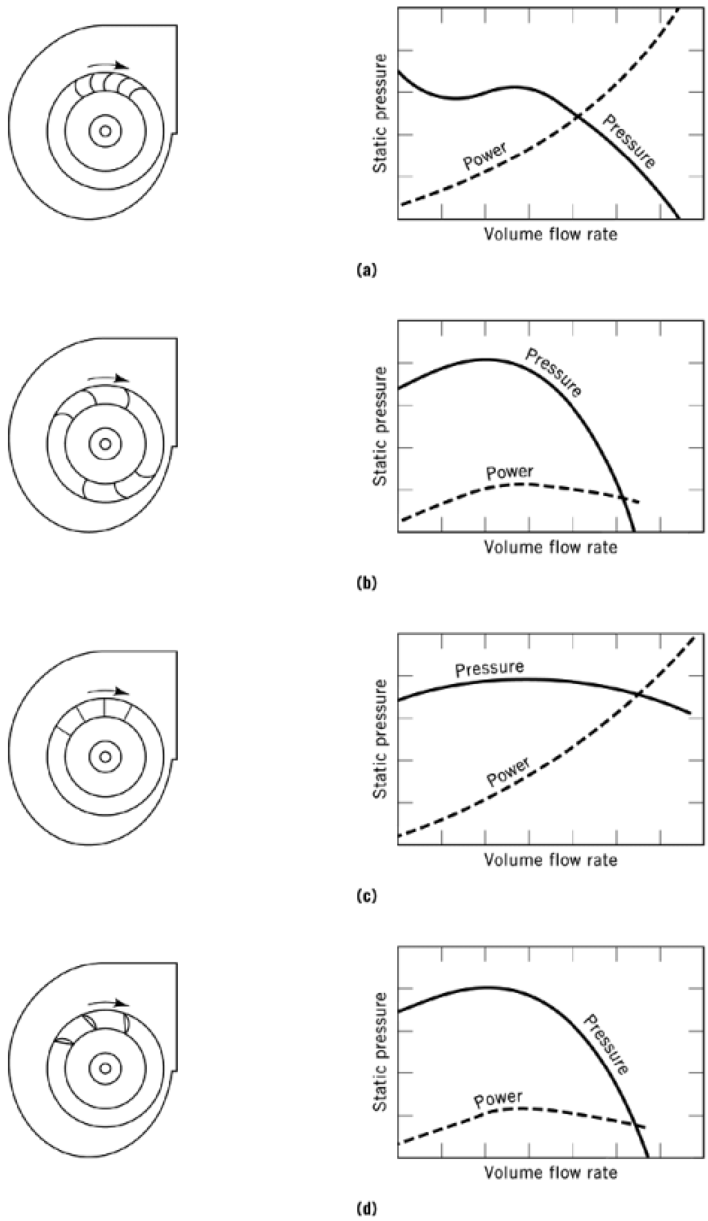

Fig. 3. Shape of fan blades and typical performance curves: (**a**) forward-curved blades; (**b**) backward-inclined blades, at left, straight backward blades at top, curbed backward blades at bottom; (**c**) straight radial blades; (**d**) airfoil blades.

1.2.4. Airfoil Design

The airfoil design is similar to the backward-curved blade (Fig. 3**d**), except that it is designed for maximum mechanical efficiency. Each blade is composed of two pieces, with the upper surface contoured to reduce air friction and provide for most efficient compression of the air. For a given performance, this fan has the highest rotational speed of any of the wheel designs. The scroll is usually designed for the most efficient conversion of velocity head to static pressure. Performance characteristics are generally similar to a backward-curved blade but power requirements are somewhat less. Such fans are generally more expensive to construct and are used only in larger sizes with higher pressures or large flow volumes where reduced operating cost justifies the increased initial expense.

2. Fan Laws and Their Applications

Manufacturers' performance ratings are generally based on atmospheric pressure at sea level, 20°C, and 50% rh. Changes in temperature, gas density, and fan speed affect the performance. Fan laws predict these effects, are invaluable in predicting fan performance at various operating points, and are accurate over an extremely wide range of speeds. Some authorities (4) list as many as 10 fan laws relating variables such as size, speed, capacity, gas density, discharge pressure, power, efficiency, and sound level. For most fan users, the following four laws are adequate.

When fan speed is changed: (*1*) the capacity or flow rate varies directly with the speed ratio; (*2*) discharge pressure varies directly with the square of the speed; (*3*) power varies directly with the cube of the speed (at constant inlet density with no change in temperature, absolute pressure, or composition); and (*4*) discharge pressure and power requirements at a constant capacity and fan speed vary directly with gas density, p. These laws apply to varying the operating conditions of the same fan, and can be used to predict the performance of a given fan if sped up or slowed down. Other laws (4) describe the effects of varying the diameter of a fan or the solidity ratio. For example, when considering performance of different diameter fans, airflow capability is a function of diameter squared, although not necessarily at the same power requirement.

2.1. Fan Selection

A fan is selected according to its location in the air-flow system, system performance and control characteristics, cost, efficiency, control stability, flexibility, and noise level (see Insulation, acoustic). Location in the flow system is very important. A fan operating on the highest density inlet gas available is smaller and less expensive, has lower operating costs, and requires less maintenance. This is partly evidenced by the fourth fan law, which states that the fan pressure varies directly with the density. Because a system has a required pressure drop at a given flow, a fan operating on a low density gas has to be operated at a higher speed than if it were operating on a more dense gas. At this higher speed the low density fan requires more power, and bearing life is shorter. Frequently, a fixed mass of air must be moved through the system rather than a fixed volume. Under this condition, the capacity of a low density fan has to be greater than that of a high density fan. Thus the benefits of using a forced-draft fan located near the air introduction point of the system become evident. Inlet density is reasonably high and most of the system pressure drop occurs on the discharge side of the fan. Placement of an exhaust fan at the end of the system with most of the system pressure drop on the suction side ensures that the fan handles a lower density gas. Similarly, if throttling flow control is used, or if the air is to be heated, it is desirable to place the throttling damper or the air heater on the fan discharge.

System flow resistance as a function of flow rate is needed to select the proper fan size. For calculation of system pressure drop see References (5–8). The resistance pressure curve for a typical system (Fig. 4**a**) shows that the pressure required to force air through the system increases with the flow rate. The pressure-volume curve of a proposed centrifugal fan has a different shape. This fan curve must be drawn for the anticipated fan inlet density expected at its location in the system. The point of intersection of these two curves locates the flow rate and pressure rise at which the fan and system operate. This intersection represents a desirable operating combination for fan and system. The system curve intersects the fan curve in the middle of its maximum efficiency range and also at a point where the fan pressure produced varies smoothly but distinctly in a constant trend with flow rate which is desirable for flow control.

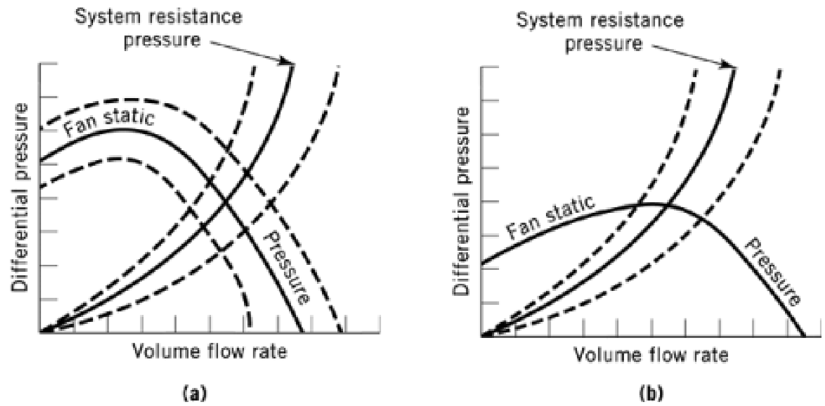

Fig. 4. Selection of fan size where the solid line represents a typical setting and the dashed lines the operating extremes. (**a**) Desirable sizing. The system resistance curve intersects the fan curve near its maximum efficiency. Changes in system resistance from a flow-control element also intersect the fan curve at desirable points for good flow control. The dashed curves also intersect system resistance curves at desirable locations. (**b**) A fan essentially too large for the system. The intersection of the system curve near the peak of the fan curve results in poor system flow control and perhaps surging.

For air-flow control, the system may contain a control valve or damper that automatically or manually modulates system pressure drop. The dotted curves in Figure 4**a** on each side of the system resistance curve might represent operating extremes of the system resistance as the control valve is varied from maximum to minimum opening. These curves also intersect the fan curve at desirable operating portions of its range both for efficiency and flow control.

If a much larger fan as in Figure 4**b** had been considered so that the system resistance curve intercepted the fan curve close to its pressure peak, flow control would be much poorer. Fan pressure rise changes very little over the anticipated flow control range so that larger changes in flow volume accompany small changes in system pressure drop. In addition, fan pressure decreases on both a flow rise and decrease. This is a situation likely to cause surging and out-of-phase hunting between the fan and an automatic control system. Higher flow rates may be required for future expansion. Lower flow rates may also be desirable seasonally. These flow changes might best be achieved through changes in speed. The dashed lines in Figure 4**a** on each side of the fan curve represent higher and lower wheel speeds for which this fan is suitable.

The wisest fan choice is frequently not the cheapest fan. A small fan operates well on its curve but may not have adequate capacity for maximum flow control, future needs, or process upset conditions. It may be so lightly constructed that it is operating near its peak speed with no provision for speed increases in the future, if needed. As fan size is increased, efficiency generally improves and wheel speed is lower. These factors decrease operating cost and provide reserve capacity for the future. However, it is also possible to oversize a fan and impair its performance.

Noise level has to be considered in fan selection. Most manufacturers provide tables of operating ranges of quietest operation. There is no set fan discharge velocity that is applicable to all fans to ensure quiet operation. Fans do not operate as quietly when throttled back as when allowed to handle substantial quantities of air. Figure 5 illustrates the range of quiet operation of a specific airfoil fan as a function of outlet velocity and discharge pressure. Outlet velocity and hence fan capacity must be allowed to increase with static pressure to stay in the quiet region. Table 1 lists typical fan outlet velocities for quiet operation. Industrial process fans having backward-inclined blades should usually be selected with discharge velocities somewhat higher than those for quiet operation to achieve best all around performance and to provide pressure reserve.

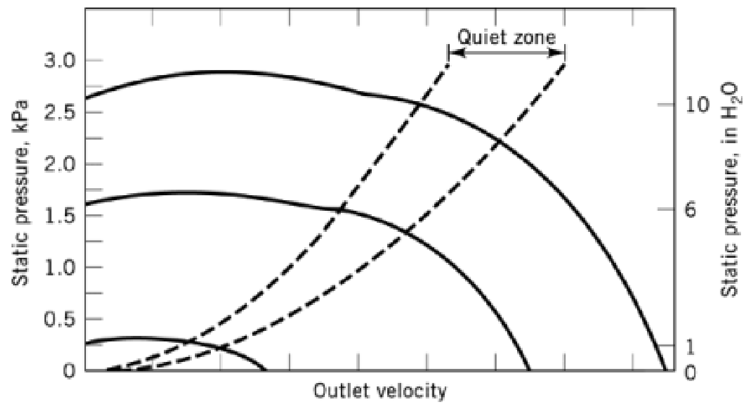

Fig. 5. Static pressure vs outlet velocity for a specific airfoil fan, where the dashed lines define the quiet operating range of an airfoil fan.

Table 1. Fan Outlet Velocities for Quiet Operation[a]

Static pressure		Forward-curved fan		Flow-nozzle airfoil fans[b]	
kPa	in. of water	m/s	ft/min	m/s	ft/min
0.25	1	8.1–10.4	1600–2050	4.3–7.4	850–1450
0.50	2	11.2–14.4	2200–2840	6.4–10.2	1250–2000
0.75	3			7.6–12.7	1500–2500
1.0	4			8.6–14.5	1700–2850
1.2	5			9.4–16.3	1850–3200
1.5	6			10.7–17.8	2100–3500
1.7	7			11.7–19.3	2300–3800
2.0	8			12.7–20.3	2500–4000
2.2	9			13.5–21.8	2650–4300
2.5	10			14.2–22.9	2800–4500
2.7	11			14.7–24.4	2900–4800
3.0	12			15.2–25.4	3000–5000

[a]Recomputed from data of New York Blower Co.
[b]Somewhat higher outlet velocities should normally be used for industrial processes using backward-inclined blade fans.

2.1.1. Duct Connections

Performance curves are measured under ideal laboratory conditions. However, to obtain the same performance curve from a fan in a field installation, the system must approach the characteristics of the test conditions at least in that part of the system close to the fan. Both inlet and outlet duct connections can influence fan performance significantly. These connections can actually change the shape of the fan curve. Therefore, no single correction factor can account for the performance change over its entire range. Although poor outlet connections affect performance, improper inlet connections generally hurt performance more, reducing it the most near free-delivery conditions and the least at peak pressure.

Poor performance can result from fan inlet eccentric or spinning flow, and discharge ductwork that does not permit development of full fan pressure. Sometimes inlet restrictions starve a fan and limit performance. To obtain rated performance, the air must enter the fan uniformly over the inlet area without rotation or unusual turbulence. This allows all portions of the fan wheel to do equal work. If more air is distributed to one side of the wheel, such as with an elbow on the inlet, the work performed by the lightly loaded portions of the

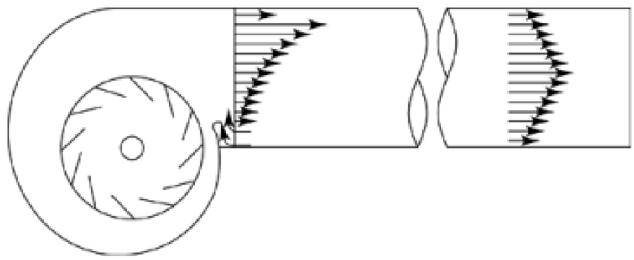

Fig. 6. Illustration of variation of velocity of air at the outlet of a centrifugal fan and the function filled by several diameters of straight discharge duct in converting velocity head to static head and establishing normal turbulent flow distribution. Bends or obstructions at the discharge outlet cause turbulence and prevent conversion of velocity head to static pressure.

wheel is reduced and capacity is decreased by 5–10%. The use of an inlet box duct on a fan can reduce capacity by as much as 25% unless there are turning vanes in the duct. Use of the vanes reduces the capacity loss to around 5%.

Spinning or vortex flow of air entering a fan can have as much effect on performance characteristics as the installation of inlet vanes to provide for reduced flow. If the air spins in the direction of wheel rotation, the bite of the blades on the air is reduced and both air flow and pressure are reduced. If the spin opposes wheel rotation, the wheel must overcome the momentum of the air: power requirements increase and efficiency is reduced. Spiral flow in a duct can be set up by a series of bends and elbows forming a corkscrew path, cyclones, and tangential inlets. A full diameter inlet duct that is straight for 10 diameters is desirable. Where such inlet connections are not possible, corrective devices should be provided in the ductwork. Spiral flow can be eliminated with the use of eggcrate straightening vanes. Turning vanes in the ductwork can largely eliminate problems of eccentric flow.

The velocity of air discharging from a fan is not uniform across the discharge outlet but tends to be higher toward the outside of the scroll as shown in Figure 6. The discharge duct evens the velocity distribution into the standard turbulent-flow distribution some distance downstream and converts part of the discharge velocity to static pressure. If a fan is operated without an outlet duct (discharging into a large plenum or the atmosphere), it loses 1–1.5 velocity heads. Such a loss must be added to the calculated system resistance. The addition of a straight discharge duct for several diameters and of the same size as the fan outlet can obviate this loss. An expanding outlet can increase static pressure beyond the curve performance if an efficient expander (included angle no more than 17°) is used. An elbow placed directly on a fan discharge destroys most of the velocity pressure leaving the fan. Suggestions for improving fan duct connections and effect factors to be applied to fan performance curves are given in various publications (7–12).

2.1.2. Flow Control

In many applications, it is desirable to be able to change the quantity of air being handled through the system. The need to change the flow may be frequent, such as every few minutes or every hour, or less frequent such as daily, weekly, or even seasonally. The choice of control method can be influenced by the frequency with which the flow must be changed. In order to control flow, either the system characteristics or the fan characteristics must be changed. Generally, flow control affects the energy input to the fan. Low cost control devices often result in reduced fan efficiency and increased power consumption. Thus if flow reduction is to occur for a long time with powerful fans, more energy-efficient control devices should be considered.

The simplest and cheapest control device is a damper, butterfly valve, or an orifice placed in the duct to throttle the flow and change the system resistance characteristics. As the flow is throttled more, system resistance is increased as illustrated in Figure 4**a.** To produce a higher discharge pressure, flow through the

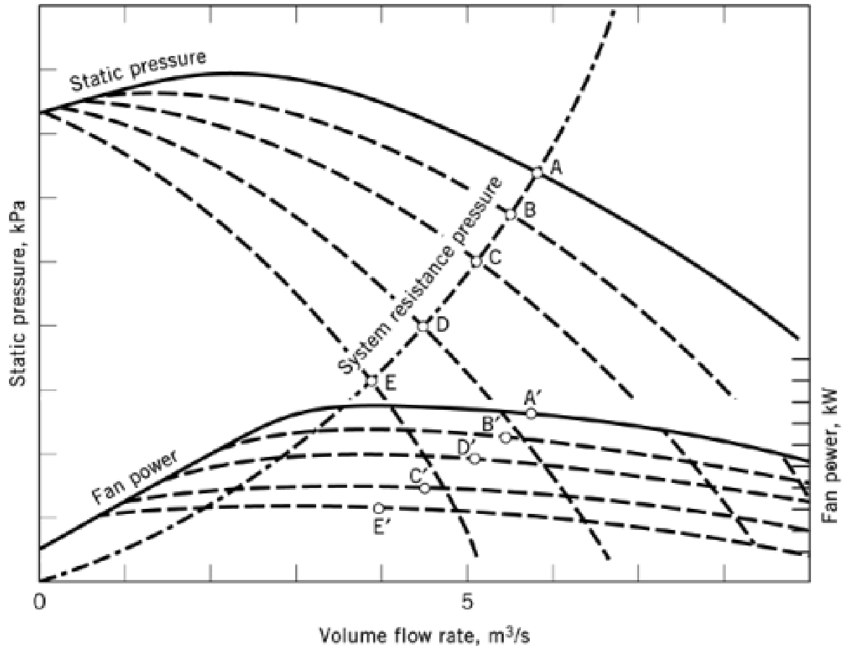

Fig. 7. Control of fan performance with inlet vane control. Solid lines marked A and A′ show normal performance without vanes (vanes wide open). As vanes are progressively closed, static and power curves are modified as indicated by dashed lines. Intersection (___) of the system resistance curve with these reduced pressure curves at points B, C, D, and E shows how imparting more spin to the inlet air reduces flow. Projecting points A to E vertically downward to the corresponding power curve locates fan power points A′ through E′. Power savings achieved over throttling control can be estimated by projecting points B through E vertically downward to the A′ power curve and comparing the value with that from the proper reduced power curve. To convert kPa to in. H_2O, divide by 0.249; to convert m^3/s to ft^3/min, multiply by 2119.

fan has to decrease. The power input is reduced but the energy expended in pressure drop across the throttling element is wasted. Placement of a throttling device on the discharge side of a fan is often preferable because the density of the air entering the fan is not reduced.

Changing centrifugal fan characteristics usually results in greater energy savings than changing system characteristics. If fan pressure can be reduced together with flow, the most desirable method of energy conservation is to change the speed, because that leaves the efficiency unchanged. If fan capacity is to be changed only infrequently, speeds of belt-driven fans can be adjusted easily with sheave changes. Where frequent speed changes are required, variable-speed motors and drives (electric or hydraulic) are the best but the most expensive. Multispeed motors and motors having step speed control can be used when infinitely variable control is not needed. The effects of speed control on a fan can be predicted from the fan laws. An alternative to speed change for axial-flow fans is blade-pitch control.

Inlet-vane control can be used to change the shape of the fan performance curve through imparting spin to the air entering the fan. As more spin is imparted, less energy can be transferred to the air from the blades and static pressure output is reduced. Figure 7 illustrates how the performance is reduced as more and more spin is imparted to the inlet air. Each setting of the inlet vanes has a separate power curve. The intersection of the system curve with the various fan pressure curves is shown, as is the equivalent power. The power required using inlet-vane control is usually intermediate between that required using throttling control and speed control.

2.1.3. Motor and Drive

The preferred prime mover for a fan is usually an electric motor. For fans of low to moderate power, V-belt drives are frequently employed. This permits selection of fans that can be operated over a wide range of speeds rather than being limited to motor synchronous speeds. Furthermore, change of speed is less expensive with V-belt drives. However, fans requiring powerful motors, 37–75 kW (50–100 hp) and higher, are generally directly connected to the motor and driven at synchronous speed.

When selecting the motor, power requirements, effect of temperature changes on load, and motor starting current and torque have to be considered. Calculation of system air-flow resistance is subject to some error and cannot always be predicted precisely. Therefore, the fan power predicted by the intersection of the fan and system curves may not be precise. If the system resistance is higher, it may be necessary to speed up the fan, which makes it draw more power. If the resistance is less than anticipated, the flow increases (unless dampered) also resulting in higher power consumption. A general rule is to size the motor for the power required for a system pressure drop both 25% greater and less than that predicted. Air temperature can also affect power requirements. A fan normally operated on a hot gas may have to be started when the system is cold. Under such conditions, the inlet gas density is much higher. The fan develops more head and a greater mass of air is delivered. Unless the system flow can be throttled back until normal operating temperatures are reached, the motor has to be sized for the cold-starting conditions based on density ratios, often two or three times normal running power.

In starting a fan, the air power increases gradually with speed which is a desirable starting load. However, in large heavy fans considerable torque is required to overcome the fan wheel inertia (referred to as WR^2, where W is the mass of the wheel and shaft and R is the radius of gyration). Figure 8 illustrates typical fan wheel and motor torques as a function of system speed during the starting process. Fan torque is that required for overcoming wheel inertia and for running power for the speed attained. Motor torque at every point on the starting curve must be greater than fan torque. The vertical difference shown in Figure 8 is the torque available for acceleration. If the motor is started across the line and the length of time required to reach full load is too long (usually 10 s is desirable), the motor may become overheated and overload controls shut off the power. Thus the fan cannot be started. On long power lines, the inrush of starting current may also drop line voltage sufficiently so that fan and motor are too slow in coming up to speed. Alternatively, reduced-voltage starters can be used, which permit extended starting periods without motor overheating, or special motors with winding that can be bypassed during starting can be used. In calculating the starting time of a fan (13), in addition to the WR^2 of the fan wheel, the flywheel effect of large drive sheaves and the motor rotor itself must also be included.

2.1.4. Other Selection Problems

Additional considerations can arise when fans must handle solids or gases of low density, or must be operated in parallel or series. A complicated flow system involving several fans in parallel, all of which are in series with a common exhaust fan, can lead to surging and vibration unless selected carefully. Maximum tip speed, bearing types, single- and double-inlet fans, and wheel and shaft natural frequency and rigidity must also be considered.

2.1.5. Low Density Gases

A fan may have to operate on low density gas because of temperature, altitude, gas composition (high water vapor content of the gas can be a cause of low density), reduced process pressure, or a combination of such causes. To develop a required pressure, the fan has to operate at a considerably higher speed than it would at atmospheric pressure, and hence it must operate much closer to top wheel speed. Bearing life is shorter, and the fan tends to vibrate more or can be overstressed more easily by a slight wheel unbalance. Abrasion of the blades from dust particles is more severe. Therefore, a sturdier fan is needed for low density gas service.

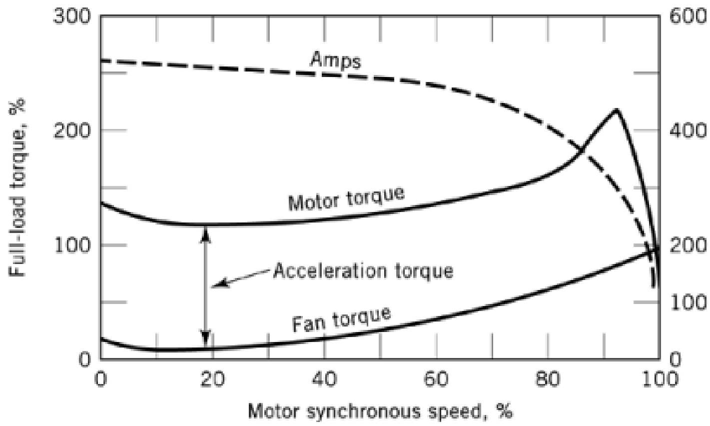

Fig. 8. Plot of fan and motor starting-torque curves.

Near top speed, a fan may operate at a speed that is near or above the natural frequency of the wheel and shaft. Under such conditions, the fan can vibrate badly even when the wheel is clean and properly balanced. Whereas manufacturers often do not check the natural frequency of the wheel and shaft in standard designs, many have suitable computer programs for such calculations. Frequency calculations should be made on large high speed fans. The first critical wheel and shaft speed of a fan that is subject to wheel deposits or out-of-balance wear should be about 25–50% above the normal operating speed.

2.1.6. Mechanical Considerations

The mechanical design of a fan and the various forces that fan parts must withstand are discussed in Reference 14. The forces result from a combination of fluid, inertial, and vibrational effects.

Tangential forces from air compression act on the blades and are transmitted through the fan hub to the shaft in the form of a resisting torque. Axial thrust may be developed on the fan wheel and shaft because of pressure differences about the wheel and the directional change in momentum of the air at the wheel inlet. The net unbalanced axial thrust must be taken by a thrust bearing that transmits it through the bearing supports to the fan foundation. At maximum fan efficiency, the radial fluid forces acting on the wheel are nearly balanced, but the volute can be correctly designed for only one rating condition. Therefore, as fan operation departs from maximum efficiency, unbalanced radial thrust increases which must be carried by the bearings to the foundation. Centrifugal forces also act on the wheel. If the center of the wheel and shaft rotation does not coincide exactly with the center of the rotating mass, a flexural force produces bending of the shaft, apparent as vibration. In a rotating elastic system, dangerous vibrations are likely to occur at critical speeds. The application of repeated external forces such as flow surging or wheel unbalance excites the elastic structure and causes it to vibrate. If the excitation frequency is close to the natural frequency, resonance can occur with large amplitude vibrations. All of these forces must be carried by the bearings. Thus it is common to use heavier components as fans are called on to operate at higher speeds or higher pressure differentials. Many fan designs are available in different construction strengths designated Class I, II, III, and IV. The higher numbers denote a fan capable of operation at a higher speed and higher pressure rise. The required class of the fan needed must be considered in its selection.

Bearings used on fans may be either sleeve or antifriction type and must be designed to withstand loads resulting from dead weight, unbalance, and rotor thrust and be able to operate at the intended maximum speed without excessive heating (see Bearing materials). When natural convection from the bearings is inadequate, some other cooling method must be provided. Lubricating oil may be circulated through an external cooler,

Table 2. Fan-Bearing Vibrational Displacement[a]

	Bearing displacement,[b] μm			
Wheel speed, rpm	Smooth	Fair	Rough	Very rough
600	50	100	200	380–500
900	38	70	150	200–250
1200	25	50	115	150–200
1800	19	38	90	125–180
3600	10	18	65	100–125

[a]Ref. 15
[b]For qualitative degrees of unbalance.

or the pillow blocks may be cored with passages for forced circulation of air or water. Fans operated at high temperatures increase the bearing cooling problem caused by heat conduction along the shaft. A small external fan wheel on the shaft, called a heat slinger, is frequently provided, or forced-circulation water cooling is used. In addition to the bearings of fans operating on hot, low density gas at high pressure rise, special attention is needed to ensure high rigidity of the wheel and shaft. Fan wheels should be balanced both statically and dynamically, eg, in the field with chalk and weights (15). Elaborate electronic test instruments are also available. An unbalanced condition causes a vibrational displacement of the bearings which is frequently checked. Table 2 lists typical displacements of fans operating at various speeds and various degrees of unbalance.

Small volume fans, usually designed with an air inlet on only one side of the wheel and casing, are known as single-inlet fans. With an enclosed wheel, the fan hub is fastened to a solid backplate that supports the blades. Larger capacity fans can be either single- or double-inlet fans. A double-inlet fan has an inlet on both sides of the wheel and casing. The hub is usually fastened to a common backplate midway between the two inlets. A double-inlet fan is generally more efficient or runs at a slower speed than a large single-inlet fan for the same capacity because the air is better distributed over the width of the wheel. Finally, air volumes are reached with large fans such that only double-inlet designs are feasible.

Vibration in a fan may be caused by mechanical problems or by the flowing air, eg, surging, poor fan-curve operating position, poor design of fan-duct connections resulting in poor air distribution, etc. A double-inlet fan is expected to have little axial unbalance because the symmetrical design of the air flow between the two halves of the wheel tends to result in a balancing of opposing forces. Such fans are frequently supplied with bearings suitable for only small thrust loads. Poor inlet ductwork arrangements can result in excessive thrust if unequal air flows are provided to opposite sides of the wheel. An unsteady air-flow unbalance that alternates between inlets can set up an alternating thrust pattern which can be very damaging to bearings designed for low thrust load. Mechanical vibration and elastic deformation problems and diagnostic techniques for structural inadequacies in fan design are discussed in Reference 16.

3. Axial Flow Fans

Axial flow fans, in which the air flow is parallel to the fan axis, are the workhorse fans in many petrochemical and utility industry applications. These are the first choice of air mover whenever large volumes of air at low (most commonly up to 500 Pa (2.0 in. H_2O)) pressures are needed. Axial flow fans range in size from 25 mm diameter (cooling computers) up to 12.3 m in diameter (cooling condenser water in power plant cooling towers). These fans are used in air-cooled heat exchangers for process cooling in many chemical plants in sizes of 1.8–4.3 m (see Heat-exchange technology). Axial fans from 0.6 to 9 m diameter are used in heating, ventilation, and air conditioning (HVAC) applications in homes and office buildings around the world. Most commonly, axial flow fans are used in short ducts called fan rings or cylinders, discharging into the atmosphere. Most large fans in

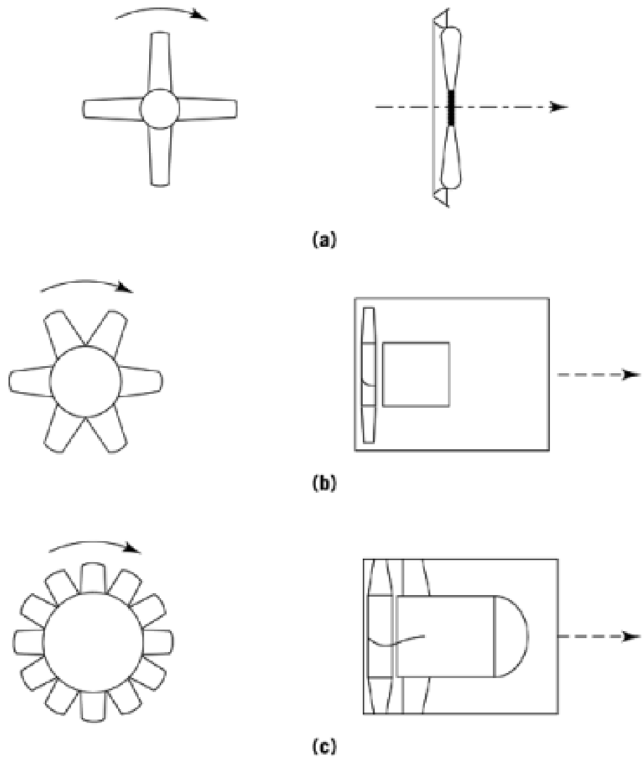

Fig. 9. Types of axial-flow fans where the dashed arrows denote the direction of air flow: (**a**) propeller fans; (**b**) tube-axial fans; (**c**) vane-axial fans (17).

cooling towers have velocity recovery stacks that capture the wasted velocity pressure energy and convert it back into useful work.

Axial fans are classified as propeller, tube-axial, and vane-axial (Fig. 9). The choice of fan required is determined by the resistance (static pressure) the fan must work against as well as the volume flow required. Propeller fans usually discharge into a plenum or directly into the atmosphere. Tube-axial fans are usually mounted in ducts as in an air conditioning system. Vane-axial fans are also mounted in ducts but feature a stationary guide vane on the discharge side that straightens the air flow to improve efficiency. Tube-axial fans can work at static pressures up to 623 Pa (2.5 in. H_2O); vane-axial fans can work up to 2000 Pa (8 in. H_2O).

3.1. Design Elements

Ideal conditions are obtained in the design of an axial-flow fan when energy transfer from the blade to the gas is uniform along the length of the blade, resulting in uniform pressure generation, minimum losses, and maximum efficiency and stability. Because the blade linear velocity varies with position from tip to hub, attainment of a uniform pressure rise along the blade at different radii requires variation of the blade angle from hub to tip. The choice of blade section is dictated by the required aerodynamic characteristics and varies in practice from cast or molded precise airfoil profiles to formed materials to single-thickness plate materials. Hub size is increased for higher pressure designs where it is impractical to generate equal pressures nearer the center of the wheel. Low pressure designs have hubs ranging from 1/3 to 1/2 wheel diameter whereas hubs in higher pressure designs may occupy 75–85% of the tip diameter. The number of blades must also be increased

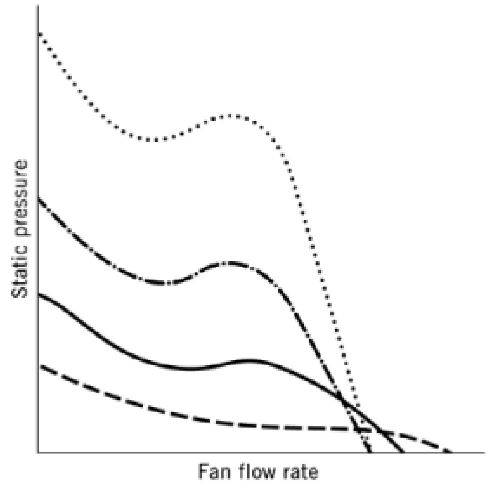

Fig. 10. Performance characteristics for axial flow fans: (–), propeller fan; (—), tube-axial fan; and (_._), single-stage and (.....), two-stage vane-axial fans.

as pressure rise is increased; 3 to 5 may be used with lower pressure designs, as many as 24 with higher pressures. Close clearance between blade tips and fan housing is a stringent requirement to prevent backflow losses at the housing wall. High pressure designs require clearances of less than 0.79 mm. The cylindrical housing of a vane-axial fan may be cast or rolled. To attain the close clearance at the blade tips, either very careful forming or machining is required. Inlet and outlet connections are carefully designed to minimize turbulence and connecting inlet and outlet ducts should be straight for at least 2–3 diameters to avoid undue effect on fan performance. Performance curves are shown in Figure 10.

3.1.1. Propeller Fans

Propeller fans may have from 2 to 6 blades mounted on a central shaft and revolving within a narrow mounting ring, either driven by belt drive or directly connected. The form of the blade in commercial units varies from a basic airfoil to simple flat or curved plates of many shapes. The wheel hub is small in diameter compared to the wheel. The blades may even be mounted to a spider frame or tube without any hub. The housing surrounding the blades can range from a simple plate or flat ring to a streamlined or curved bell–mouth orifice.

Some type of close-clearance shroud at the blade tips is desirable to prevent air recirculation from the discharge side of the blades back to the suction side. A propeller fan with no shroud has fairly low efficiency because of air recirculation. A curved orifice-like ring greatly reduces air recirculation and improves efficiency. An angle-shaped ring essentially eliminates recirculation, and optimum efficiency is achieved with an angle-like ring with streamlined edges. Power requirements for most propeller fans increase as flow decreases and static pressure increases.

3.1.2. Tube-Axial Fans

The tube-axial fan is a refinement of the propeller fan in both wheel design and mechanical strength, having improved capacity, pressure level, and efficiency. Designs are often capable of operating over a greater range of speeds. The cheapest fans may have an open-type propeller wheel with the motor enclosed in a tube if directly connected. Belt-drive models are also available. In more refined types, the blades are shorter and of airfoil cross section mounted on a large diameter hub which may approach 50% of the wheel diameter. The hub and motor tube are normally of the same diameter and reduce the back flow of higher pressure air, which might recycle through less effective central portions of the wheel if a smaller hub were utilized. The performance curve (Fig. 10) may have a dip to the left of the pressure peak which would constitute an unstable region for fan operation and which should be avoided. Commercial models are available having static pressures up to 750 Pa (3 in. of H_2O). The general range of application is for pressures of 125–375 Pa permitting use of appreciable ductwork. Maximum efficiencies are in the range of 65–75%. Principal applications are in industrial processes and ventilation requiring moderate static pressures and the need for simplicity of fan installation in a straight duct.

3.1.3. Vane-Axial Fans

The vane-axial fan resulted from the development of the propeller fan using refined aerodynamic principles and precise manufacturing procedures and control. Where such principles and techniques are applied, excellent capacity, pressure, efficiency, and sound emission levels are attained. Some units have mechanical efficiencies above 90%. High efficiency vane-axial fans are more efficient than comparable centrifugal fans, and have been used for energy conservation in Europe for some time. U.S. industry has also shown interest in large vane-axial fans for energy conservation in applications such as electric power boiler service (18, 19).

The vane-axial fan wheel has short, stubby airfoil blades mounted on a hub which may be as large as 75% of the wheel diameter. The air leaving the axial-flow wheel has an appreciable rotational component which can be converted to static pressure in a suitably designed set of stationary straightening vanes. The straightening vanes are shaped to pick up the air leaving the wheel blades without shock. Although straightening vanes of airfoil cross section are theoretically desirable, vanes formed of pressed heavy sheet metal are less expensive. The motor is enclosed in a housing having the same diameter as the hub and has either a rounded cap or a bullet-shaped tail to reduce eddy losses. The straightening vanes surround the motor housing and can serve as structural supports for the housing. Generally, the number of guide vanes exceeds the number of propeller vanes by one, with the numbers selected so that there is no common divisor for the number of hub vanes and guide vanes. This minimizes flow pulsation and noise. Single-stage fans can develop pressures to 1.5 kPa (6 in. of water) with some designs going as high as 2.25 kPa (9 in. of water). Standard designs are available either belt-driven or directly connected to motors with speeds as high as 3450 rpm. In addition, two-stage units have been developed that produce considerably higher pressures but have received little industrial use. Performance curves show a dip to the left of the pressure peak (Fig. 10). Whereas vane-axial-flow fans can be designed that do not have such dips, those that do have dips should be operated to the right of the pressure peak. The principal advantage of the vane-axial fan is compactness and convenience of use in inline ducts, plus its better efficiency when carefully designed. The higher manufacturing precision required generally eliminates any cost savings that might result from its smaller size.

3.2. Capacity Control

3.2.1. Variable Air Flow Fans

Variable air flow fans are needed in the process industry for steam or vapor condensing or other temperature critical duties. These also produce significant power savings. Variable air flow is accomplished by (*1*) variable speed motors (most commonly variable frequency drives (VFDs); (*2*) variable pitch fan hubs; (*3*) two-speed motors; (*4*) selectively turning off fans in multiple fan installations; or (*5*) variable exit louvers or dampers. Of these methods, VFDs and variable pitch fans are the most efficient. Variable louvers, which throttle the airflow, are the least efficient. The various means of controlling air flow are summarized in Table 3.

Variable frequency drives are based on the principle that motor speed is a direct function of the frequency of the alternating current. In other words, a frequency of 60 Hz produces 100% speed; 30 Hz frequency produces 50% speed. The development of these drives is receiving much attention and both costs and the size of the controllers are steadily decreasing. Many options for control are programmable via keyboards mounted on the control boxes. One advantage of VFDs is that often several types of soft start options, ie, variable ramp times,

Table 3. Variable Air Flow Devises

Device	Air flow control	Cost		Noise
		Initial	Operating	
variable speed	continuously variable	high	lowest	decreases with speed
variable pitch	continuously variable	medium	low	constant
two-speed motor	full/half	medium–low	medium–low	decreases at half-speed
outlet louvers	variable	medium	highest	constant

are available as well as digital readout of many functions, etc. The main advantage of the VFD is that once the operating point of the fan is selected, that efficiency is carried at all speeds and flows. Capacity is directly related to motor frequency and thus to speed. Another important benefit is that as the fan speed decreases, fan noise and vibration also decrease significantly.

Variable pitch fans or controllable pitch fans operate by means of specifically designed hubs which permit changing blade pitch while the fan is in operation. These are available in fans of 1.5 m to 6 m diameter for use in the petrochemical industry consuming from 3.7 to 56 kW. Large fans in the utility industry can consume up to 373 kW and work at up to 12.5 kPa (50 in. H_2O). These fans typically are pneumatically operated by 20–103 kPa (3–15 psi) or 4–20 mA signals, and are designed to fail to maximum air flow in the event of signal pressure loss. When fan speed is constant, fan noise remains almost constant, even if air flow is essentially zero. Typically the range of air flow is an essentially linear decrease from maximum to zero flow, the pitch being regulated by the controller. Typically, a 20 kPa (3 psig) signal relates to 100% (design duty) whereas a 103 kPa (15 psig) signal relates to essentially zero air flow.

A variable pitch fan has the unique capability to produce significant amounts of negative air flow when adjusted to do so. Negative flow, in the opposite direction from normal flow, is used in some refinery services to keep from over cooling some types of fluids in the tubes which have low pour points, and in cooling tower services to periodically deice the tower inlet louvers. The variable pitch fan can produce approximately 60% of its upward flow in the negative direction at maximum power. Another advantage of variable pitch fans is that the initial cost is generally 50% or less compared to VFD-type drives.

Two-speed motors are typically used on noncondensing services where the process is not sensitive to temperature but mostly seasonal or variable throughput of fluids in the air cooler requires some degree of air flow control. This is a simple, rather inexpensive means to control air flow when volume air flow is not critical. Typical motor ratings are 1800/900 rpm, although 1800/1200 rpm types are available.

Air control louvers or dampers, popular in the past for air flow control, are used primarily for only very low scale air flow control. Louvers are used in many winterized heat exchangers in extremely low ambient temperature locations to retain and recirculate warm air in completely enclosed heat exchangers, sometimes in very complicated control schemes. The use of louvers on the discharge side of a fan to control air flow is inefficient and creates mechanical problems in the louvers because of the turbulence. A fan is a constant volume device, thus use of louvers to control flow is equivalent to throttling flow through a valve. The fan keeps moving the original amount of air at the original power even when the net flow is reduced by the partially closed louver. This is very wasteful of energy. Louvers are controlled by pneumatic or electric actuators utilizing the 20–103 kPa (3–15 psig) or 4–20 mA signal. Large banks of louvers are linked together by mechanical linkages.

3.3. Fan Rating

Axial fans have the capability to do work, ie, static pressure capability, based on their diameter, tip speed, number of blades, and width of blades. A typical fan used in the petrochemical industry has four blades, operates near 61 m/s tip speed, and can operate against 248.8 Pa (1 in. H_2O). A typical performance curve

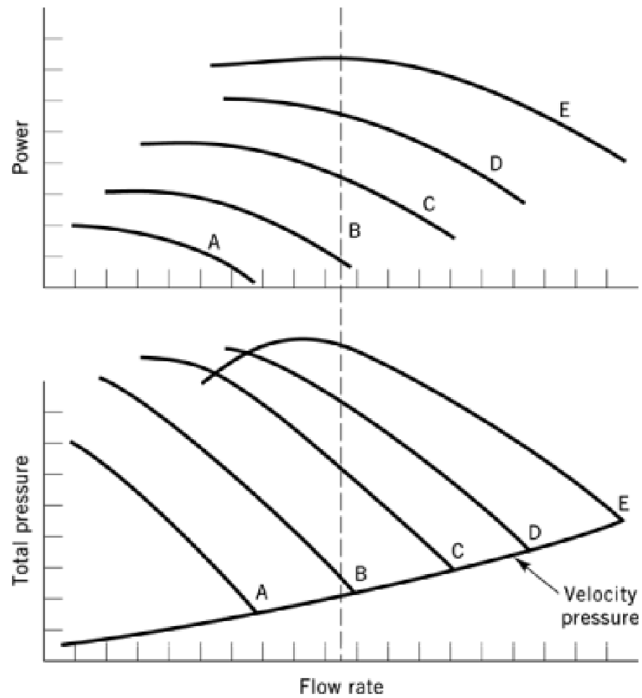

Fig. 11. Performance curve for an axial fan where A, B, C, D, and E represent fan pitch angles of 6°, 10°, 14°, 18°, and 22°, respectively.

is shown in Figure 11 where both total pressure and velocity pressure are shown, but not static pressure. However, total pressure minus velocity pressure equals static pressure. Velocity pressure is the work done just to collect the air in front of the fan inlet and propel it into the fan throat. No useful work is done but work is expended. This is called a parasitic loss and must be accounted for when determining power requirements. Some manufacturers' fan curves only show pressure capability in terms of static pressure vs flow rate, ignoring the velocity pressure requirement. This can lead to grossly underestimating power requirements.

3.3.1. Efficiency

Fan efficiency describes a fan's ability to do work and is calculated as total efficiency3.3.1 (Eff) using total pressure (TP) or static efficiency3.3.1 (Eff) using static pressure (SP). Total efficiencies for axial fans range from 55 to 80%; static efficiencies range from about 40 to 65%. When pressure is in pascals and flow rate in m^3/s,

$$\text{total Eff} = \frac{\text{TP} \times \text{flow rate} \times 10^{-3}}{\text{kW}}$$

$$\text{static Eff} = \frac{\text{SP} \times \text{flow rate} \times 10^{-3}}{\text{kW}}$$

For flow rate in ft^3/min and power in brake horsepower, a correction factor of 0.157 must be applied to each equation.

An axial fan is a constant volume device. That is, a fan at a certain pitch moves a constant volume of air or gas at a constant speed and resistance (static pressure). If the density changes, the static pressure and wattage

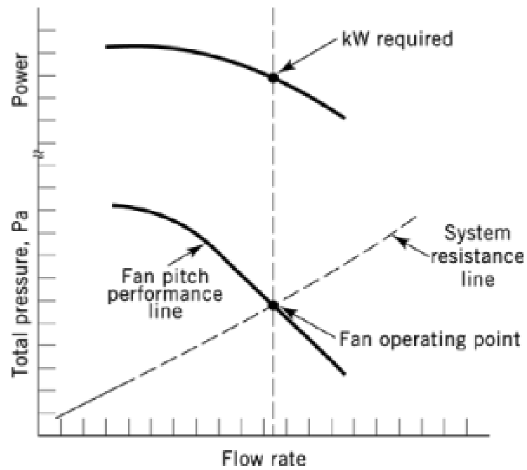

Fig. 12. Fan operating point.

change, but the volume remains constant, ie, if the density (temperature) decreases, the static pressure and kW go up, but air flow remains the same.

3.3.2. Performance Curves

Fan manufacturers furnish fan performance curves for each type fan available. These are typically based on 61 m/s (12,000 ft/min) tip speed and 1.20 kg/m^3 (0.075 lb/ft^3) density. To select a fan for a specific duty requires knowledge of the flow, static pressure resistance, and density of the actual operating conditions. Usually the fan diameter is known as well as some idea of operating speed; a 61 m/s tip speed can often be assumed.

3.4. Selection

The fan selection process consists of determining the exact operating point that coincides with the design static pressure, air flow, and density required by the system resistance line. A typical procedure would be (*1*) select a fan curve of the appropriate diameter, assuming four blades for small fans, eight blades for larger fans; (*2*) calculate the required operating point which relates tip speed and pressures to the tip speed and density of the fan curve; (*3* select a fan pitch which relates the flow rate and total pressure required; (*4*) read curve power requirement and pitch angle; and (*5*) relate power curve to actual rpm and density by using the fan laws. Most fan manufacturers gladly assist in fan selections. A fan operating point, the point where fan output exactly satisfies the system requirements, is shown in Figure 12.

3.5. Application Criteria

The design and construction of axial fans is dictated by size and function. Small fans are usually molded plastic having fixed blades. Most fans larger than one meter in diameter feature hollow fiber glass or extruded or cast aluminum blades that can be adjusted to the proper pitch when the fan is at rest, to provide the required air flow at the design speed. To perform properly, the output of the fan in terms of air flow and static pressure capability must match the system resistance at the design air density. This requirement dictates the fan diameter, rpm, number and types of blades, and blade pitch settings. If a fan has fixed pitch blades, the fan speed must be adjusted. Another increasingly important requirement is fan noise. To meet the maximum allowable noise, fan speed is normally limited. Often fan diameters are determined by space limitations as well as by volume flow requirements.

In petrochemical plants, fans are most commonly used in air-cooled heat exchangers that can be described as overgrown automobile radiators (see Heat-exchange technology). Process fluid in the finned tubes is cooled usually by two fans, either forced draft (fans below the bundle) or induced draft (fans above the bundles). Normally, one fan is a fixed pitch and one is variable pitch to control the process outlet temperature within a closely controlled set point. A temperature indicating controller (TIC) measures the outlet fluid temperature and controls the variable pitch fan to maintain the set point temperature to within a few degrees.

The utility industry utilizes fans typically from 6.7–10 m diameter in banks of 8 to 12 fans in wet cooling towers. These towers cool the water used to condense the steam from the turbines. Many towers may be needed in large plants requiring as many as 50 to 60 fans 12 m in diameter. These fans typically utilize velocity recovery stacks to recoup some of the velocity pressure losses and convert it to useful static pressure work.

4. Noise of Fans

Fan noise is demanding and receiving much attention because of environmental laws. The basic control document is the federal OSHA limitation of 90 dB(A) at an operator's work place for 8-h exposure. There are other limitations on entire plant noise at the boundary of new plants from local ordinances which are typically more severe than the OSHA limitation.

Two terms are important: the sound pressure level (SPL) and sound power level (PWL). SPL or Lp is expressed as decibels (dB). PWL or Lw is a number representing the energy level of the noisemaking device, also in dB. The terms dB(A) and octave bands refer to the sound spectrum as a whole, where the dB(A) is a weighted sum of a spectrum represented by a single number. The A weighting system represents the response of the human ear, ie, low frequency sounds are not distinguished as well as high frequencies. For a more demanding description of noise, the octave band center frequencies are used. The most frequently used octave bands are 32, 63, 125, 250, 500, 1k, 2k, 4k, and 8k Hz.

To utilize a noise limitation specification, several items must be known: (*1*) the specification of SPL in decibels expressed either in dB(A) or octave band levels; (*2*) the distance from the measurement point to the geometric center of the noisemaking device or array; and (*3*) the quantity of noisemaking devices (fans) in the array. Additionally, noise attenuates with distance by the relation $10\log_{10}(1/12.57R^2)$ when R is line of sight distance in meters from source to measurement point (the term is $-20\log R$ when R is in feet). Noise from multiple sources increases by the relation $10\log N$ where N is the number of identical sources in an array. For example, two identical fans produce 3 dB more noise than one fan ($10\log 2 = 3$). The terms SPL and PWL are related by $\text{SPL} = \text{PWL} + 10\log_{10}(1/12.57R^2) + 10\log N$. For example, if a unit having two fans must not exceed a SPL of 64 dB(A), and it is 10.7 m (35 ft) from the measurement point to the center of the noise source, to determine the fan operating conditions the PWL must be used. Solving for PWL and using R in m, $\text{PWL} = \text{SPL} - 10\log_{10}(1/12.57R^2) - 10\log N = 91.9\ \text{dB(A)}$. Therefore the fan designer must select a fan that meets the design airflow needs at the design static pressure but produces no more than the PWL of 91.9 dB(A).

The required PWL is used to determine the tip speed of the fan. The velocity of the blade tips equals the $\text{rpm} \times \text{dia} \times \pi$. The American Petroleum Institute (API) determines PWL through the equation

$$\text{PWL dB(A)} = 36 + 30\log U\text{tip} + 10\log\text{kW}$$

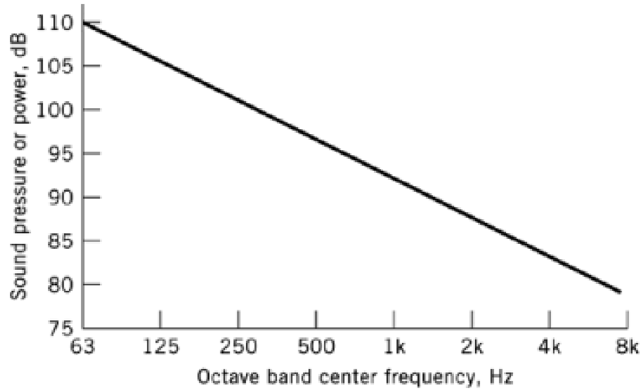

Fig. 13. Axial fan frequency profile.

where *U*tip is tip speed in m/s and the power absorbed by the fan is in kW. For tip speed in ft/min and power in brake horsepower (HP), the equation becomes

$$\text{PWL dB (A)} = 56 + 30\log\frac{\text{tip speed}}{1000} + 10\log\text{HP}$$

Those values also determine the rpm to operate the fan so that the PWL can be met.

A typical axial fan produces most noise at low frequencies, eg, 63, 125, and 250 Hz. A frequency spectrum for a 4.3 m diameter fan operating at 61 m/s is shown in Figure 13.

4.1. Effect of Vibration

All objects have a natural frequency of vibration when struck sharply and fan rings, blades, structure, etc are no exception. Vibration is usually sinusoidal and its frequency measured in Hertz. The travel or displacement of the vibration is measured in mils (1/1000 of an inch) in the United States but in micrometers elsewhere. Another measurement is velocity (mm/s) of movement.

A fan blade is continuously vibrating millions of cycles up and down in operation over a short period of time. Each time a blade tip moves past an obstruction it is loaded and then unloaded. If forced by virtue of tip speed and number of blades to vibrate at its natural frequency, the amplitude is greatly increased and internal stresses result. It is very important when selecting or rating a fan to avoid operation near the natural frequency. The most common method of checking for a resonance problem is by using the relation:

$$\text{BPF} = \frac{\text{no. blades} \times \text{rpm}}{60}\,\text{Hz}$$

when BPF is the blade passing frequency. This is compared to the blades first and second mode natural frequency to make sure there is sufficient safety margin. Conventional margins are 5–20% difference between resonant frequencies and BPF. Another term used is that of critical speed which is a way of checking that the first or second mode resonant frequency is not selected for fan operation.

$$\text{rpmc1} = \frac{\text{Fn1} \times 60}{\text{no. blades}}$$

where rpmc1 = critical speed and Fn1 = first mode critical frequency. Similarly, rpmc2 can be calculated to check for second mode resonance. Important points regarding vibration and fans are fan rotor unbalance occurs only at 1 times running speed; vibration at blade pass frequency is usually caused by aerodynamic problems or structure resonance; and vibration at motor speed usually indicates misalignment or unbalance of a drive shaft. Resolving fan vibration problems usually requires knowledge not only of amplitude but also frequency of vibration. Fan rotor unbalance can be corrected by a dynamic fan balance operation in-place. Vibration at blade pass frequency is harder to resolve because it often relates to a resonance in the structure or some component, and the only means to resolve this type of problem is to change fan speed which is often expensive, or to change the number of blades in the fan.

A design guide for air-cooled heat exchangers in refinery services specifies 0.15 mm (6 mils) maximum vibration on motors and principal structural components (20). Generally, large fans are slow-moving compared to large, high speed turbines so amplitudes can generally be larger without damage.

4.1.1. Fatigue from Vibrations

To avoid the possibility of fatigue caused by a resonance, more extensive use of fiber glass blades is being made rather than metallic blades such as aluminum. Fiber glass composite blades, which are often hollow, are not notch sensitive, ie, a small scratch or crack does not spell the disaster it would in a metal blade. Secondly, the lighter mass of a fiber glass blade means less kinetic energy to dissipate in the event of an accident, because the destructive energy in a fan wreck is directly proportional to the mass weight of the blades.

5. Uses

Fans and blowers are the most widely used mechanical devices for moving air and gases in both large and small volumes (21). Uses include ventilation, mechanical draft for combustion (including forced- and induced-draft fans and primary- and secondary-air fans), local exhaust for fume and dust containment at hoods and equipment enclosures, forced- and induced-draft cooling for spray towers, cooling towers and ponds, and air-cooled heat exchangers, and conveying of solids (see also Heat-exchange technology). Other applications include air or gas movement in dryers, gas-recirculation fans, air supply for air curtains and air-blast operations, and a great many miscellaneous process industry uses often involving hot and corrosive gases. The range of performance required by fans for these various applications is enormous. Most ventilating applications require pressures ranging from 25 to 1500 Pa (0.1–6 in. of water). Induced-draft fans must often handle gases of 150–425°C containing various levels of suspended erosive particles. Such fans are frequently equipped with replaceable wear pads of abrasion-resistant materials or are coated with wear-resistant surfaces.

Forced-draft fans generally operate on clean air and at pressures from a few hundred Pa to as high as 20 kPa (80 in. of water) for pressurized furnaces. Backward-inclined blading is used almost exclusively for high efficiency. Blades with airfoil contours give improved structural strength, higher efficiency, and lower sound levels in large fans. Conveying systems in which the solids pass through the fan almost always use low speed wheels of the radial-blade paddle-wheel-type of construction. In the area of hot and corrosive process gas handling, fan designs are adapted to the specific need of the process. Frequently stainless steel or other alloy construction is required. Where dilution of the gas with atmospheric air is objectionable, the fan shaft is equipped with a stuffing box, a rubber labyrinth seal, or even a purged rotary seal depending on the degree of contamination control required. Occasionally, fans must handle gases having sticky or tarry particulates where solids buildup can occur. Continuous or intermittent flushing of the fan with a liquid spray in the fan inlet is helpful. In addition to deliberate flushing, fans may be called on to handle gases containing mist or entrained liquid droplets. A large percentage of such mist may be collected and agglomerated in the fan, particularly if operated at a high tip speed. Liquid-handling fans must be equipped with oversize motors as the acceleration of the liquid within the fan can utilize considerable power. Particular attention must be paid to the corrosiveness of the wet–dry environment within the fan. The presence of chloride ions and high wheel stress can lead to stress corrosion cracking in stainless-steel wheels. Although elimination of the chlorides is the best solution, the use of much lower wheel-tip speeds and wheels that can be stress-relieved to remove residual fabrication stresses is often helpful. Fans with rubber and polymeric coatings are often useful in moist environments, but special considerations in fan design are necessary to assure thorough bonding of such coatings to the wheel. Buildup of liquid within the fan casing can also be a problem with liquid-handling fans. The use of bottom–horizontal discharge designs with a large discharge duct drain is generally more satisfactory than a small fan-housing drain.

The choice between axial-flow and centrifugal fans in certain applications is by no means clear-cut. Axial fans have an advantage when compactness is important and when straight through flow benefits the installation and frequently, they also have higher efficiency which is important in energy conservation. Advantages of the centrifugal fan are the better ability to cope with fluctuating operating conditions, conditions that could result in unstable fan operation; greater ability to vary fan performance through speed changes; better access to the fan motor and greater facility to provide sturdy structural support for the fan and motor; generally lower noise level; and natural adaptability to design situations where a 90° turn in gas direction is desirable.

Centrifugal blowers may be obtained from American Fan Co., Fairfield, Ohio; TLT-Babcock, Inc., Akron, Ohio; Buffalo Forge Co., Buffalo, New York; Zurn Industries, Clarage Fan Division, Birmingham, Alabama; Lamson Corp., Centrifugal Air Systems Division, Syracuse, New York; and Hartzell Fan, Inc., Piqua, Ohio. Axial fans are available from Hudson Products Corp., Houston, Texas; Buffalo Forge Co., Buffalo, New York; Hartzell Fan, Inc., Piqua, Ohio; Howden Sirocco Inc., Hyde Park, Massachusetts; Moore Co., Marceline, Missouri; and Aerovent, Inc., Piqua, Ohio.

BIBLIOGRAPHY

"Fans and Blowers" in *ECT* 3rd ed., Vol. 9, pp. 768–794, by B. B. Crocker, Monsanto Co.

Cited Publications

1. *ASHRAE 1992 Handbook - HVAC Systems and Equipment*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, Ga., 1992.
2. *Laboratory Methods of Testing Fans for Rating AMCA Standard 210-85*, Air Moving and Conditioning Association, Arlington Heights, Ill., 1990.
3. *Methods for Calculating Fan Sound Ratings from Laboratory Test Data Standard 301*, Air Moving and Conditioning Association, Arlington Heights, Ill., 1990.
4. R. Jorgensen, *Fan Enginering*, 8th ed., Buffalo Forge Co., Buffalo, N.Y., 1983.
5. J. L. Alden and J. M. Kane, *Design of Industrial Exhaust Systems for Dust and Fume Removal*, 4th ed., Industrial Press, New York, 1970.
6. *AMCA Fan Application Manual, AMCA Publication B200-3*, Air Moving and Conditioning Association, Arlington Heights, Ill., 1990.
7. J. W. Market, *Heat., Piping, Air Cond.* **42**, 100 (Oct. 1970).
8. D. G. Traver, in D. G. Traver, *Fan Application - Testing and Selection Symposium Papers, San Francisco, Calif., Jan. 1970*, ASHRAE, Atlanta, 1972.
9. *AMCA Fan Application Manual, AMCA Publication 201*, Air Moving and Conditioning Association, Arlington Heights, Ill., 1973, Sect. 1.
10. D. H. Cristie, *ASHRAE Trans.* **77**, 84 (1971).
11. L. S. Marks and E. A. Winzenburger, *Trans. ASME* **54**(21), 213 (1932).
12. H. F. Farquhar, "Outlet Ducts–Effect on Fan Performance" in Ref. 8, 7–10.

13. Ref. 4, 295–313.
14. Ref. 4, 271–283.
15. Ref. 4, 285–288.
16. J. W. Martz and R. R. Pfahler, *Hydrocarbon Process.* **54**, 57 (June 1975).
17. Ref. 1, p. 3.2.
18. C. E. Wagner, *Combustion* **47**, 20 (Mar. 1976).
19. C. C. Curley and P. Olesen, *Combustion* **48**, 23 (Sept. 1976).
20. *Air-Cooled Heat Exchangers for General Refinery Service*, API 661, American Petroleum Institute, Washington, D.C., Apr. 1992.
21. Ref. 4, Chapt. 14–22.

General References

22. T. Baumeister Jr., *Fans*, McGraw-Hill Book Co., Inc., New York, 1935.
23. W. C. Osborne, *Fans*, 2nd ed., Pergamon Press, New York, 1977.
24. J. K. Salisbury, ed., *Kent's Mechanical Engineers Handbook*, Power Vol., 12th ed., John Wiley & Sons, Inc., New York, 1950.
25. R. Pollak, *Chem. Eng.* **80** (Jan. 1973).
26. J. B. Graham, in J. B. Graham, *Fan Application - Testing and Selection Symposium Papers, San Francisco, Calif., Jan. 1970*, ASHRAE, New York, 1972.
27. J. Thompson, *Plant Eng.* **31** (May 1977).
28. R. C. Monroe, *Hydrocarbon Proc.*, (Dec. 1980).
29. R. C. Monroe, *Chem. Eng.*, (May and June 1985).
30. *American Petroleum Institute STD 661*, Washington, D.C., Apr. 1992.
31. *Noise Control*
32. L. L. Beranek, *Noise and Vibration Control*, Institute of Noise Control Engineering, Cambridge, Mass., 1988.
33. A. Thumann and R. Miller, *Fundamentals of Noise Control Engineering*, 2nd ed., The Fairmont Press, Inc., Lilburn, Ga., 1990.
34. W. Neise, *Sound Vibration* (Apr. 1976).
35. J. B. Graham, *Can. Min. J.*, (Oct. 1975).
36. *In-Duct Measurement of Centrifugal Fan Sound Power*, ASHRAE, Sept. 1976.

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