Related Commercial Resources

CHAPTER 20

FANS

A FAN is an air pump that creates a pressure difference and causes airflow. The impeller does work on the air, imparting to it both static and kinetic energy, which vary in proportion, depending on the fan type.

Fan efficiency ratings are based on ideal conditions; some fans are rated at more than 90% total efficiency. However, actual connections often make it impossible to achieve ideal efficiencies in the field.

TYPES OF FANS

Fans are generally classified as centrifugal or axial flow according to the direction of airflow through the impeller. Figure 1 shows the general configuration of a centrifugal fan. The components of an axial-flow fan are shown in Figure 2. Table 1 compares typical characteristics of some of the most common fan types.

Two modified versions of the centrifugal fan are used but are not listed in Table 1 as separate fan types. Unhoused centrifugal fan impellers are used as circulators in some industrial applications (e.g., heat-treating ovens) and are identified as plug fans. In this case, there is no duct connection to the fan because it simply circulates the air within the oven. In some HVAC installations, the unhoused fan impeller is located in a plenum chamber with the fan inlet connected to an inlet duct from the system. Outlet ducts are connected to the plenum chamber. This fan arrangement is identified as a plenum fan.

PRINCIPLES OF OPERATION

All fans produce pressure by altering the airflow's velocity vector. A fan produces pressure and/or airflow because the rotating blades of the impeller impart kinetic energy to the air by changing its velocity. Velocity change is in the tangential and radial velocity components for centrifugal fans, and in the axial and tangential velocity components for axial-flow fans.

Centrifugal fan impellers produce pressure from the (1) centrifugal force created by rotating the air column contained between the blades and (2) kinetic energy imparted to the air by its velocity leaving the impeller. This velocity is a combination of rotational velocity of the impeller and airspeed relative to the impeller. When the blades are inclined forward, these two velocities are cumulative; when backward, oppositional. Backward-curved blade fans are generally more efficient than forward-curved blade fans.

Axial-flow fan impellers produce pressure principally by the change in air velocity as it passes through the impeller blades, with none being produced by centrifugal force. These fans are divided into three types: propeller, tubeaxial, and vaneaxial. Propeller fans, customarily used at or near free air delivery, usually have a small hub-to-tip-ratio impeller mounted in an orifice plate or inlet ring. Tubeaxial fans usually have reduced tip clearance and operate at higher tip speeds, giving them a higher total pressure capability than the propeller fan. Vaneaxial fans are essentially tubeaxial fans with guide vanes and reduced running blade tip clearance, which give improved pressure, efficiency, and noise characteristics.

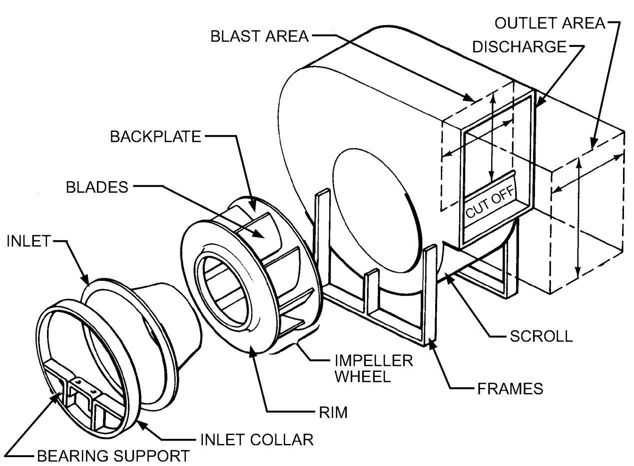
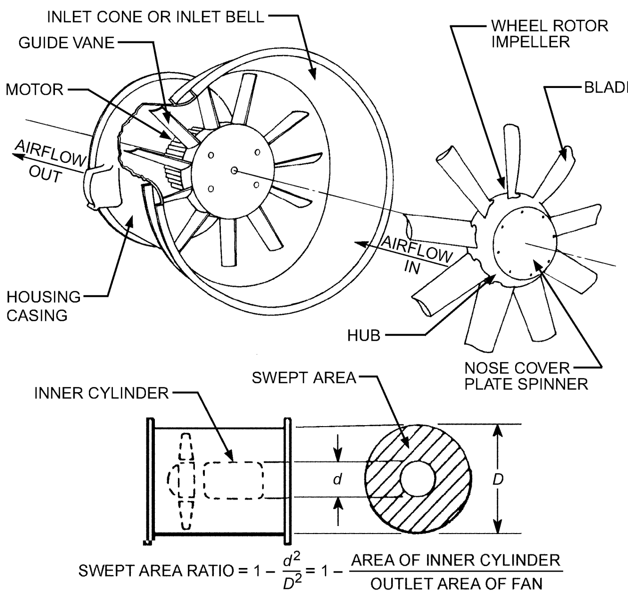


Fig. 1 Centrifugal Fan Components



Note: The swept area ratio in axial fans is equivalent to the blast area ratio in centrifugal fans.

Fig. 2 Axial Fan Components

The preparation of this chapter is assigned to TC 5.1, Fans.

Table 1 Types of Fans

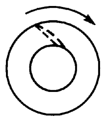
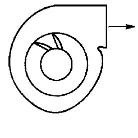
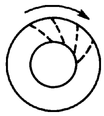
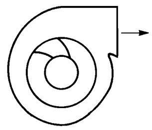
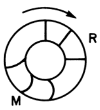
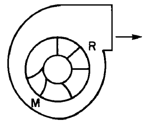
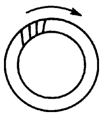
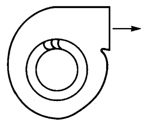
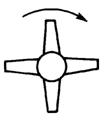
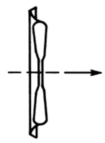
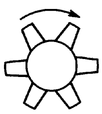
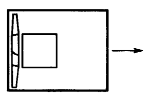
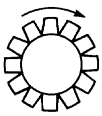
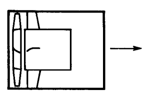
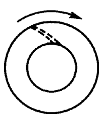
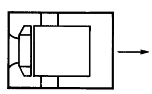
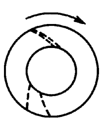
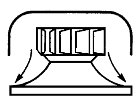
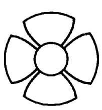
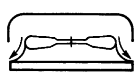
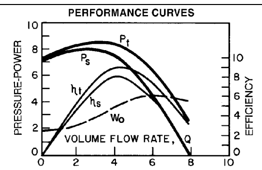
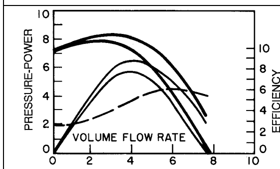
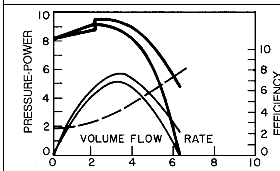
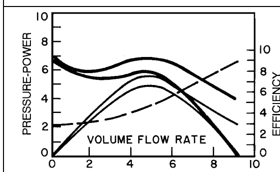
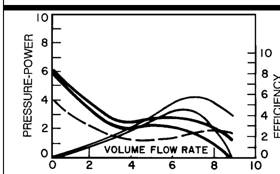
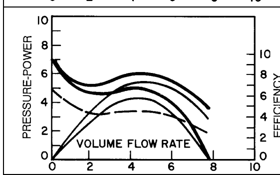
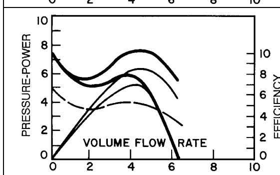
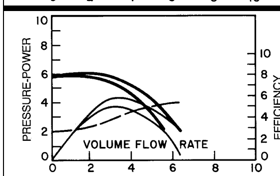
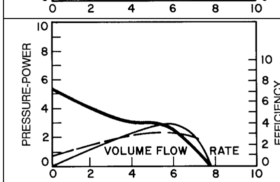
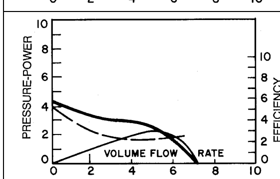
	TYPE		IMPELLER DESIGN		HOUSING DESIGN
CENTRIFUGAL FANS	AIRFOIL		Highest efficiency of all centrifugal fan designs. Ten to 16 blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. Air leaves impeller at velocity less than tip speed. For given duty, has highest speed of centrifugal fan designs.		Scroll design for efficient conversion of velocity pressure to static pressure. Maximum efficiency requires close clearance and alignment between wheel and inlet.
CENTRIFUGAL FANS	BACKWARD-INCLINED BACKWARD-CURVED		Efficiency only slightly less than airfoil fan. Ten to 16 single-thickness blades curved or inclined away from direction of rotation. Efficient for same reasons as airfoil fan.		Uses same housing configuration as airfoil design.
CENTRIFUGAL FANS	RADIAL		Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. Curve may have a break to left of peak pressure and fan should not be operated in this area. Power rises continually to free delivery.		Scroll. Usually narrowest of all centrifugal designs. Because wheel design is less efficient, housing dimensions are not as critical as for airfoil and backward-inclined fans.
CENTRIFUGAL FANS	FORWARD-CURVED		Flatter pressure curve and lower efficiency than the airfoil, backward-curved, and backward-inclined. Do not rate fan in the pressure curve dip to the left of peak pressure. Power rises continually toward free delivery. Motor selection must take this into account.		Scroll similar to and often identical to other centrifugal fan designs. Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.
AXIAL FANS	PROPELLER		Low efficiency. Limited to low-pressure applications. Usually low-cost impellers have two or more blades of single thickness attached to relatively small hub. Primary energy transfer by velocity pressure.		Simple circular ring, orifice plate, or venturi. Optimum design is close to blade tips and forms smooth airfoil into wheel.
AXIAL FANS	TUBEAXIAL		Somewhat more efficient and capable of developing more useful static pressure than propeller fan. Usually has 4 to 8 blades with airfoil or single-thickness cross section. Hub is usually less than half the fan tip diameter.		Cylindrical tube with close clearance to blade tips.
AXIAL FANS	VANEAXIAL		Good blade design gives medium- to high-pressure capability at good efficiency. Most efficient have airfoil blades. Blades may have fixed, adjustable, or controllable pitch. Hub is usually greater than half fan tip diameter.		Cylindrical tube with close clearance to blade tips. Guide vanes upstream or downstream from impeller increase pressure capability and efficiency.
SPECIAL DESIGNS	TUBULAR CENTRIFUGAL		Performance similar to backward-curved fan except capacity and pressure are lower. Lower efficiency than backward-curved fan. Performance curve may have a dip to the left of peak pressure.		Cylindrical tube similar to vaneaxial fan, except clearance to wheel is not as close. Air discharges radially from wheel and turns 90° to flow through guide vanes.
SPECIAL DESIGNS	POWER ROOF VENTILATORS — CENTRIFUGAL		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. Centrifugal units are slightly quieter than axial units.		Normal housing not used, because air discharges from impeller in full circle. Usually does not include configuration to recover velocity pressure component.
SPECIAL DESIGNS	POWER ROOF VENTILATORS — AXIAL		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units.		Essentially a propeller fan mounted in a supporting structure. Hood protects fan from weather and acts as safety guard. Air discharges from annular space at bottom of weather hood.

Table 1 Types of Fans (*Concluded*)

PERFORMANCE CURVES*	PERFORMANCE CHARACTERISTICS	APPLICATIONS
	Highest efficiencies occur at 50 to 60% of wide-open volume. This volume also has good pressure characteristics. Power reaches maximum near peak efficiency and becomes lower, or self-limiting, toward free delivery.	General heating, ventilating, and air-conditioning applications. Usually only applied to large systems, which may be low-, medium-, or high-pressure applications. Applied to large, clean-air industrial operations for significant energy savings.
	Similar to airfoil fan, except peak efficiency slightly lower.	Same heating, ventilating, and air-conditioning applications as airfoil fan. Used in some industrial applications where environment may corrode or erode airfoil blade.
	Higher pressure characteristics than airfoil and backward-curved fans. Pressure may drop suddenly at left of peak pressure, but this usually causes no problems. Power rises continually to free delivery.	Primarily for materials handling in industrial plants. Also for some high-pressure industrial requirements. Rugged wheel is simple to repair in the field. Wheel sometimes coated with special material. Not common for HVAC applications.
	Pressure curve less steep than that of backward-curved fans. Curve dips to left of peak pressure. Highest efficiency to right of peak pressure at 40 to 50% of wide-open volume. Rate fan to right of peak pressure. Account for power curve, which rises continually toward free delivery, when selecting motor.	Primarily for low-pressure HVAC applications, such as residential furnaces, central station units, and packaged air conditioners.
	High flow rate, but very low-pressure capabilities. Maximum efficiency reached near free delivery. Discharge pattern circular and airstream swirls.	For low-pressure, high-volume air moving applications, such as air circulation in a space or ventilation through a wall without ductwork. Used for makeup air applications.
	High flow rate, medium-pressure capabilities. Performance curve dips to left of peak pressure. Avoid operating fan in this region. Discharge pattern circular and airstream rotates or swirls.	Low- and medium-pressure ducted HVAC applications where air distribution downstream is not critical. Used in some industrial applications, such as drying ovens, paint spray booths, and fume exhausts.
	High-pressure characteristics with medium-volume flow capabilities. Performance curve dips to left of peak pressure because of aerodynamic stall. Avoid operating fan in this region. Guide vanes correct circular motion imparted by wheel and improve pressure characteristics and efficiency of fan.	General HVAC systems in low-, medium-, and high-pressure applications where straight-through flow and compact installation are required. Has good downstream air distribution. Used in industrial applications in place of tubeaxial fans. More compact than centrifugal fans for same duty.
	Performance similar to backward-curved fan, except capacity and pressure is lower. Lower efficiency than backward-curved fan because air turns 90°. Performance curve of some designs is similar to axial flow fan and dips to left of peak pressure.	Primarily for low-pressure, return air systems in HVAC applications. Has straight-through flow.
	Usually operated without ductwork; therefore, operates at very low pressure and high volume. Only static pressure and static efficiency are shown for this fan.	Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity flow exhaust systems. Centrifugal units are somewhat quieter than axial flow units.
	Usually operated without ductwork; therefore, operates at very low pressure and high volume. Only static pressure and static efficiency are shown for this fan.	Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity-flow exhaust systems.

*These performance curves reflect general characteristics of various fans as commonly applied. They are not intended to provide complete selection criteria, because other parameters, such as diameter and speed, are not defined.

Table 1 includes typical performance curves for various types of fans. These performance curves show the general characteristics of various fans as they are normally used; they do not reflect fan characteristics reduced to common denominators such as constant speed or constant propeller diameter, because fans are not selected on the basis of these constants. The efficiencies and power characteristics shown are general indications for each type of fan. A specific fan (size, speed) must be selected by evaluating actual characteristics.

TESTING AND RATING

ANSI/ASHRAE *Standard* 51 (ANSI/AMCA *Standard* 210) specifies the procedures and test setups to be used in testing fans and other air-moving devices. Figure 3 diagrams one of the most common procedures for developing characteristics of a fan tested from **shutoff** conditions to nearly **free delivery** conditions. At shutoff, the duct is completely blocked off; at free delivery, the outlet resistance is reduced to zero. Between these two conditions, various airflow restrictions are placed on the end of the duct to simulate various operating conditions on the fan. Sufficient points are obtained to define the curve between shutoff and free air delivery conditions. Pitot-static tube traverses of the test duct are performed with the fan operating at constant rotational speed. The point of rating may be any point on the fan performance curve. For each case, the specific point on the curve must be defined by referring to the airflow rate and corresponding total or static pressure. Other test setups described in ANSI/ASHRAE *Standard* 51 should produce the same performance curve.

Fans designed for use with duct systems are tested with a length of duct between the fan and measuring station. This length of duct smooths the air discharged from the fan and provides stable, uniform airflow conditions at the plane of measurement. Measured pressures are corrected back to fan outlet conditions. Fans designed for use without ducts, including almost all propeller fans and power roof ventilators, are tested without ductwork.

Not all sizes are tested for rating. Test information may be used to calculate performance of larger fans that are geometrically similar, but such information should not be extrapolated to smaller fans. For performance of one fan to be determined from the known performance of another, the two fans must be dynamically similar. Strict dynamic similarity requires that the important nondimensional parameters (those that affect aerodynamic characteristics, such as Mach number, Reynolds number, surface roughness, and gap size) vary in only insignificant ways. (For more specific information, consult the manufacturer's application manual or engineering data.)

FAN LAWS

The fan laws (see Table 2) relate performance variables for any dynamically similar series of fans. The variables are fan size D, rotational speed N, gas density ρ, volume airflow rate Q, pressure P_{tf} or P_{sf}, power W, and mechanical efficiency η_t. **Fan Law 1** shows the effect of changing size, speed, or density on volume airflow rate, pressure, and power level. **Fan Law 2** shows the effect of changing size, pressure, or density on volume airflow rate, speed, and power. **Fan Law 3** shows the effect of changing size, volume airflow rate, or density on speed, pressure, and power.

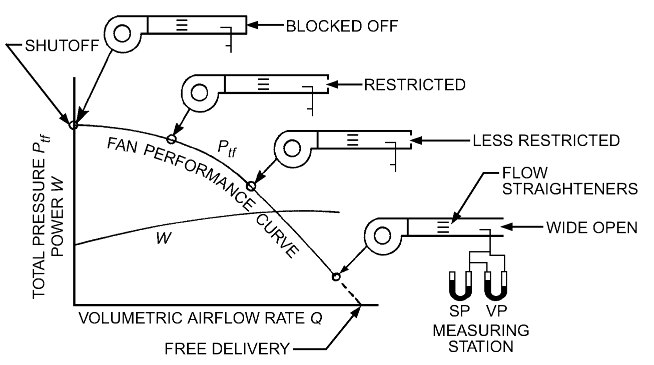


Fig. 3 Method of Obtaining Fan Performance Curves

The fan laws apply only to a series of aerodynamically similar fans at the same point of rating on the performance curve. They can be used to predict the performance of any fan when test data are available for any fan of the same series. Fan laws may also be used with a particular fan to determine the effect of speed change. However, caution should be exercised in these cases, because the laws apply only when all flow conditions are similar. Changing the speed of a given fan changes parameters that may invalidate the fan laws.

Unless otherwise identified, fan performance data are based on dry air at standard conditions: 14.696 psi and 70°F (0.075 lb/ft^3). In actual applications, the fan may be required to handle air or gas at some other density. The change in density may be caused by temperature, composition of the gas, or altitude. As indicated by the fan laws, fan performance is affected by gas density. With constant size and speed, power and pressure vary in accordance with the ratio of gas density to standard air density.

Figure 4 illustrates the application of the fan laws for a change in fan speed N for a specific-sized fan (i.e., $D_1 = D_2$). The computed P_t curve is derived from the base curve. For example, point E ($N_1 = 650$) is computed from point D ($N_2 = 600$) as follows:

At point D,

$$Q_2 = 6000 \text{ cfm and } P_{tf_2} = 1.13 \text{ in. of water}$$

Using Fan Law 1a at point E,

$$Q_1 = 6000(650/600) = 6500 \text{ cfm}$$

Using Fan Law 1b ($\rho_1 = \rho_2$),

$$P_{tf_1} = 1.13(650/600)^2 = 1.33 \text{ in. of water}$$

The total pressure curve P_{tf_1} at N = 650 rpm may be generated by computing additional points from data on the base curve, such as point G from point F.

If equivalent points of rating are joined, as shown by the dashed lines in Figure 4, they form parabolas, which are defined by the relationship expressed in Equation (2).

Each point on the base P_{tf} curve determines only one point on the computed curve. For example, point H cannot be calculated from either point D or point F. Point H is, however, related to some point between these two points on the base curve, and only that point can be used to locate point H. Furthermore, point D cannot be

Table 2 Fan Laws

Law No.	Dependent Variables				Independent Variables
1a	Q_1	=	Q_2	×	$(D_1/D_2)^3 \ (N_1/N_2)$
1b	P_1	=	P_2	×	$(D_1/D_2)^2 \ (N_1/N_2)^2 \ \rho_1/\rho_2$
1c	W_1	=	W_2	×	$(D_1/D_2)^5 \ (N_1/N_2)^3 \ \rho_1/\rho_2$
2a	Q_1	=	Q_2	×	$(D_1/D_2)^2 \ (P_1/P_2)^{1/2} \ (\rho_2/\rho_1)^{1/2}$
2b	N_1	=	N_2	×	$(D_2/D_1) \ (P_1/P_2)^{1/2} \ (\rho_2/\rho_1)^{1/2}$
2c	W_1	=	W_2	×	$(D_1/D_2)^2 \ (P_1/P_2)^{3/2} \ (\rho_2/\rho_1)^{1/2}$
3a	N_1	=	N_2	×	$(D_2/D_1)^3 \ (Q_1/Q_2)$
3b	P_1	=	P_2	×	$(D_2/D_1)^4 \ (Q_1/Q_2)^2 \ \rho_1/\rho_2$
3c	W_1	=	W_2	×	$(D_2/D_1)^4 \ (Q_1/Q_2)^3 \ \rho_1/\rho_2$

Notes:
1. Subscript 1 denotes fan under consideration. Subscript 2 denotes tested fan.
2. For all fans laws $(\eta_t)_1 = (\eta_t)_2$ and (Point of rating)$_1$ = (Point of rating)$_2$.
3. P equals either P_{tf} or P_{sf}.

used to calculate point F on the base curve. The entire base curve must be defined by test.

FAN AND SYSTEM PRESSURE RELATIONSHIPS

As previously stated, a fan impeller imparts static and kinetic energy to the air. This energy is represented in the increase in total pressure and can be converted to static or velocity pressure. These two quantities are interdependent: fan performance cannot be evaluated by considering one alone. Energy conversion, indicated by changes in velocity pressure to static pressure and vice versa, depends on the efficiency of conversion. Energy conversion occurs in the discharge duct connected to a fan being tested in accordance with ANSI/ASHRAE *Standard* 51, and the efficiency is reflected in the rating.

Fan total pressure rise P_{tf} is a true indication of the energy imparted to the airstream by the fan. System pressure loss (ΔP) is the sum of all individual total pressure losses imposed by the air distribution system duct elements on both the inlet and outlet sides of the fan. An energy loss in a duct system can be defined only as a total pressure loss. The measured static pressure loss in a duct element equals the total pressure loss only in the special case where air velocities are the same at both the entrance and exit of the duct element. By using total pressure for both fan selection and air distribution system design, the design engineer ensures proper design. These fundamental principles apply to both high- and low-velocity systems. (Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals* has further information.)

Fan static pressure rise P_{sf} is often used in low-velocity ventilating systems where the fan outlet area essentially equals the fan outlet duct area, and little energy conversion occurs. When fan performance data are given in terms of P_{sf}, the value of P_{tf} may be calculated from catalog data.

To specify the pressure performance of a fan, the relationship of P_{tf}, P_{sf}, and P_{vf} must be understood, especially when negative pressures are involved. Most importantly, P_{sf} is defined in ANSI/ASHRAE *Standard* 51 as $P_{sf} = P_{tf} - P_{vf}$. Except in special cases, P_{sf} is not necessarily the measured difference between static pressure on the inlet side and static pressure on the outlet side.

Figures 5 to 8 depict the relationships among these various pressures. Note that, as defined, $P_{tf} = P_{t2} - P_{t1}$. Figure 5 illustrates a fan with an outlet system but no connected inlet system. In this case, fan static pressure P_{sf} equals the static pressure rise across the fan. Figure 6 shows a fan with an inlet but no outlet system. Figure 7 shows a fan with both an inlet and an outlet system. In both cases,

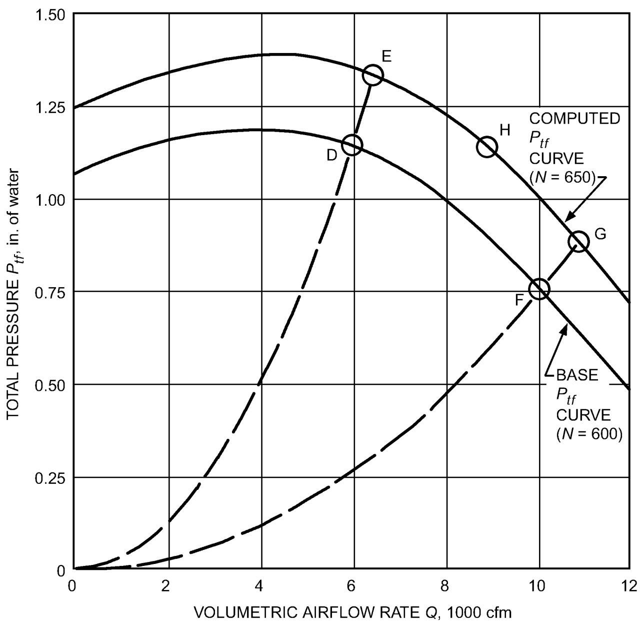


Fig. 4 Example Application of Fan Laws

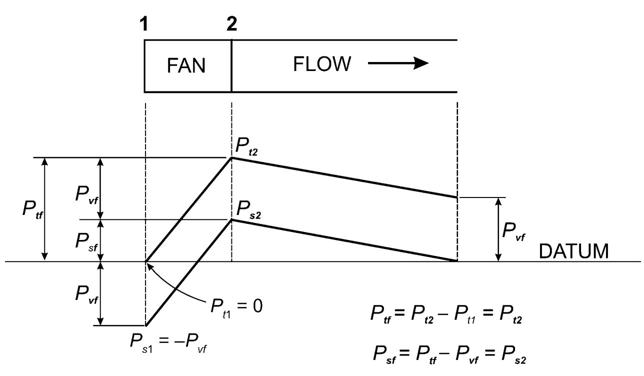


Fig. 5 Pressure Relationships of Fan with Outlet System Only

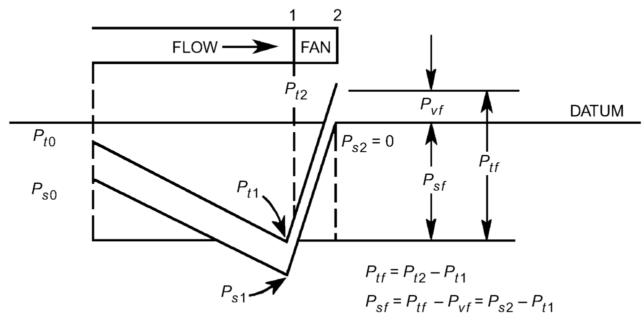


Fig. 6 Pressure Relationships of Fan with Inlet System Only

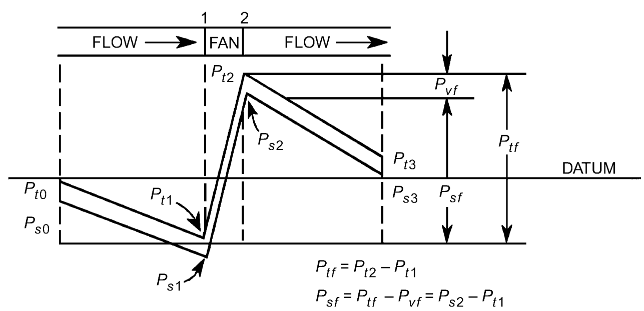


Fig. 7 Pressure Relationships of Fan with Equal-Sized Inlet and Outlet Systems

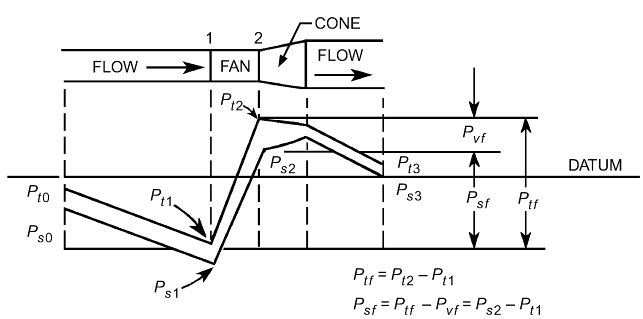


Fig. 8 Pressure Relationships of Fan with Diverging Cone Outlet

the measured difference in static pressure across the fan ($P_{s2} - P_{s1}$) is not equal to the fan static pressure (P_{sf}).

All the systems illustrated in Figures 5 to 7 have inlet or outlet ducts that match the fan connections in size. Usually the duct size is not identical to the fan outlet or inlet, so that a further complication is introduced. To illustrate the pressure relationships in this case, Figure 8 shows a diverging outlet cone, which is a common type of fan connection. In this case, the pressure relationships at the fan outlet do not match the pressure relationships in the airflow section. Furthermore, static pressure in the cone actually increases in the direction of airflow. The static pressure changes throughout the system, depending on velocity. The total pressure, which, as noted in the figure, decreases in the direction of airflow, more truly represents the loss introduced by the cone or by flow in the duct. Only the fan changes this trend. Total pressure, therefore, is a better indication of fan and duct system performance. In this normal fan situation, the static pressure across the fan ($P_{s2} - P_{s1}$) does not equal the fan static pressure P_{sf}.

TEMPERATURE RISE ACROSS FANS

In certain applications, it may be desirable to calculate the temperature rise across the fan. For low pressure rises (<10 in. of water), the temperature rise may be found by the following:

$$\Delta T = \frac{\Delta P C_p}{\rho c_p J \eta} \tag{1}$$

where

ΔT = temperature rise across fan, °F
ΔP = pressure rise across fan, in. of water
C_p = conversion factor = 5.193 $lb_f/ft^2 \cdot$in. of water
ρ = density = 0.075 lb_m/ft^3
c_p = specific heat = 0.24 $Btu/lb_m \cdot °F$
J = mechanical equivalent of heat = 778.2 $ft \cdot lb_f/Btu$
η = efficiency, decimal

If the motor is not in the airstream, the efficiency is the fan total efficiency. If the motor is in the airstream, the efficiency is the set efficiency (combined efficiencies of motor and fan).

DUCT SYSTEM CHARACTERISTICS

Figure 9 shows a simplified duct system with three 90° elbows. These elbows represent the resistance offered by the ductwork, heat exchangers, cabinets, dampers, grilles, and other system components. A given rate of airflow through a system requires a definite total pressure in the system. If the rate of airflow changes, the resulting total pressure required will vary, as shown in Equation (2), which is true for turbulent airflow systems. HVAC systems generally follow this law very closely.

$$(\Delta P_2/\Delta P_1) = (Q_2/Q_1)^2 \tag{2}$$

This chapter covers only turbulent flow (the flow regime in which most fans operate). In some systems, particularly constant- or variable-volume air conditioning, the air-handling devices and associated controls may produce effective system resistance curves that deviate widely from Equation (2), even though each element of the system may be described by this equation.

Equation (2) permits plotting a turbulent flow system's pressure loss (ΔP) curve from one known operating condition (see Figure 4). The fixed system must operate at some point on this system curve as the volume flow rate changes. As an example, in Figure 10, at point A of curve A, when the flow rate through a duct system such as that shown in Figure 9 is 10,000 cfm, the total pressure drop is 3 in. of water. If these values are substituted in Equation (2) for ΔP_1 and Q_1, other points of the system's ΔP curve (Figure 10) can be determined.

For 6000 cfm (Point D on Figure 10):

$$\Delta P_2 = 3(6000/10{,}000)^2 = 1.08 \text{ in. of water}$$

If a change is made within the system so that the total pressure at design flow rate is increased, the system will no longer operate on the previous ΔP curve, and a new curve will be defined.

For example, in Figure 11, an elbow added to the duct system shown in Figure 9 increases the total pressure of the system. If the total pressure at 10,000 cfm is increased by 1.00 in. of water, the system total pressure drop at this point is now 4.00 in. of water, as shown by point B in Figure 10.

If the system in Figure 9 is changed by removing one of the schematic elbows (Figure 12), the resulting system total pressure

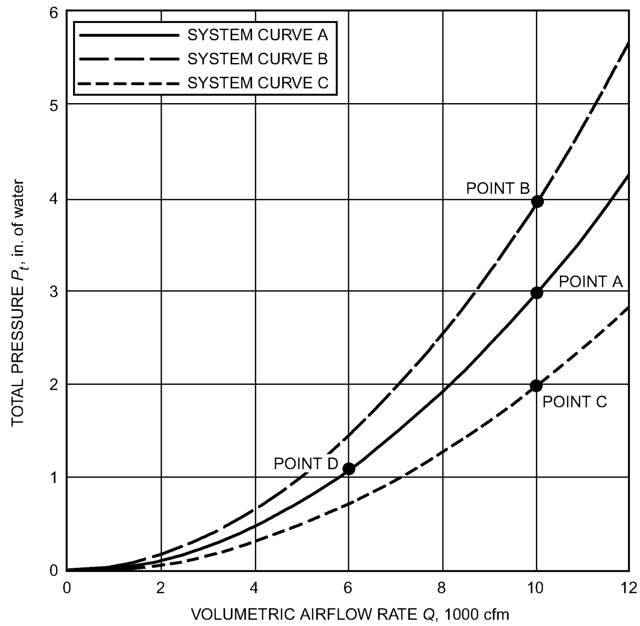


Fig. 10 Example System Total Pressure Loss (ΔP) Curves

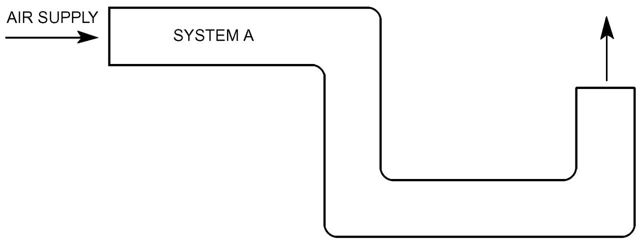


Fig. 9 Simple Duct System with Resistance to Flow Represented by Three 90° Elbows

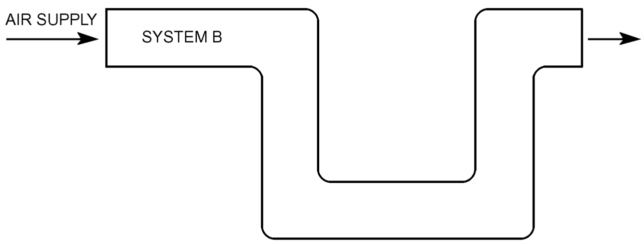


Fig. 11 Resistance Added to Duct System of Figure 9

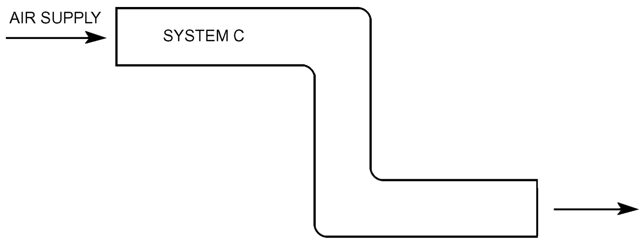


Fig. 12 Resistance Removed from Duct System of Figure 9

drops below the total pressure resistance, and the new ΔP curve is curve C of Figure 10. For curve C, a total pressure reduction of 1.00 in. of water has been assumed when 10,000 cfm flows through the system; thus, the point of operation is at 2.00 in. of water, as shown by point C.

These three ΔP curves all follow the relationship expressed in Equation (2). These curves result from changes in the system itself and do not change fan performance. During design, such system total pressure changes may occur because of alternative duct routing, differences in duct sizes, allowance for future duct extensions, or the design safety factor being applied to the system.

In an actual operating system, these three ΔP curves can represent three system characteristic lines caused by three different positions of a throttling control damper. Curve C is the most open position, and curve B is the most closed. A control damper forms a continuous series of these ΔP curves as it moves from wide open to completely closed and covers a much wider range of operation than is illustrated here. Such curves can also represent the clogging of turbulent flow filters in a system.

SYSTEM EFFECTS

Normally, a fan is tested with open inlets, and a section of straight duct is attached to the outlet. This setup results in uniform flow into the fan and efficient static pressure recovery on the fan outlet. If good inlet and outlet conditions are not provided in the actual installation, fan performance suffers. To select and apply the fan properly, these effects must be considered and the pressure requirements of the fan, as calculated by standard duct design procedures, must be increased.

These calculated system effect factors are only an approximation, however. Fans of different types, and even fans of the same type but supplied by different manufacturers, do not necessarily react to a system in the same way. Therefore, judgment based on experience must be applied to any design. Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals* gives information on calculating the system effect factors and lists loss coefficients for a variety of fittings. Clarke et al. (1978) and AMCA *Publication* 201 provide further information.

SELECTION

After the system pressure loss curve of the air distribution system has been defined, a fan can be selected to meet the system requirements (Graham 1966, 1972). Fan manufacturers present performance data in either graphic (curve) (Figure 13) or tabular form (multirating tables). Multirating tables usually provide only performance data within the recommended operating range. The optimum selection range or peak efficiency point is identified in various ways by different manufacturers.

Performance data as tabulated in the usual fan tables are based on arbitrary increments of flow rate and pressure. In these tables, adjacent data, either horizontally or vertically, represent different points of operation (i.e., different points of rating) on the fan performance curve. These points of rating depend solely on the fan's

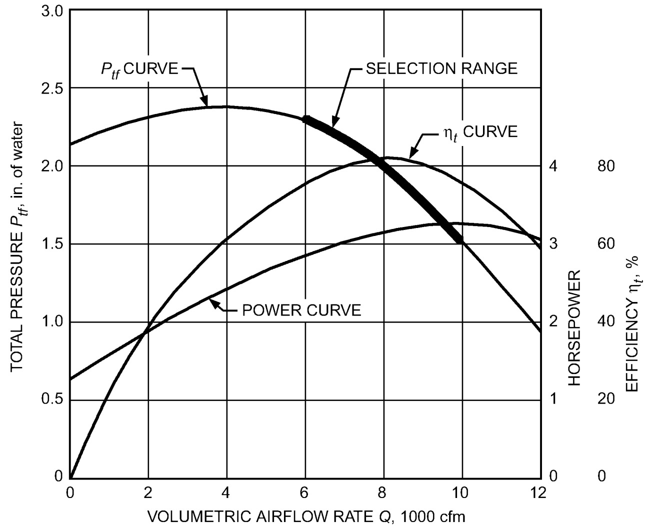


Fig. 13 Conventional Fan Performance Curve Used by Most Manufacturers

characteristics; they cannot be obtained from each other by the fan laws. However, points of operation listed in multirating tables are usually close together, so intermediate points may be interpolated arithmetically with adequate accuracy for fan selection.

Selecting a fan for a particular air distribution system requires that the fan pressure characteristics fit the system pressure characteristics. Thus, the total system must be evaluated and airflow requirements, resistances, and system effect factors at the fan inlet and outlet must be known (see Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*). Fan speed and power requirements are then calculated, using multirating tables or single or multispeed performance curves or graphs.

In using curves, it is necessary that the point of operation selected (Figure 14) represent a desirable point on the fan curve, so that maximum efficiency and resistance to stall and pulsation can be attained. In systems where more than one point of operation is encountered during operation, it is necessary to look at the range of performance and evaluate how the selected fan reacts within this complete range. This analysis is particularly necessary for variable-volume systems, where not only the fan undergoes a change in performance, but the entire system deviates from the relationships defined in Equation (2). In these cases, it is necessary to look at actual losses in the system at performance extremes.

PARALLEL FAN OPERATION

The combined performance curve for two fans operating in parallel may be plotted by using the appropriate pressure for the ordinates and the sum of the volumes for the abscissas. When two fans having a pressure reduction to the left of the peak pressure point are operated in parallel, a fluctuating load condition may result if one of the fans operates to the left of the peak static point on its performance curve.

The P_t curves of a single fan and of two identical fans operating in parallel are shown in Figure 15. Curve A-A shows the pressure characteristics of a single fan. Curve C-C is the combined performance of the two fans. The unique figure-8 shape is a plot of all possible combinations of volume airflow at each pressure value for the individual fans. All points to the right of CD are the result of each fan operating at the right of its peak point of rating. Stable performance results for all systems with less obstruction to airflow than is shown on the ΔP curve D-D. At points of operation to the left of CD,

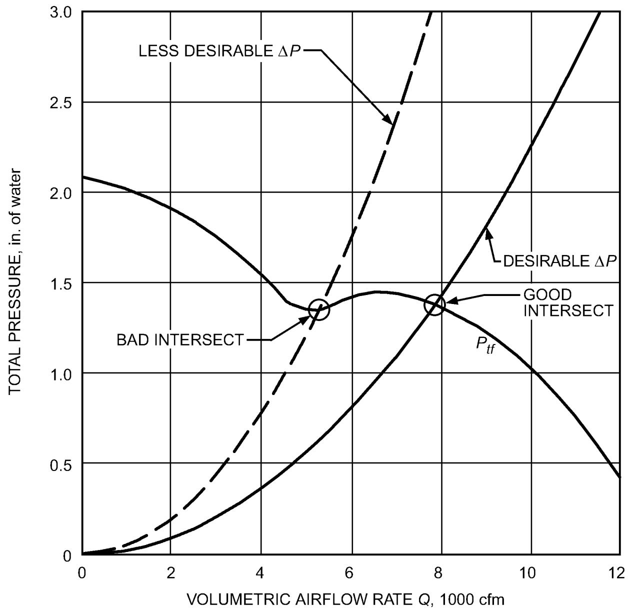


Fig. 14 Desirable Combination of P_{tf} and ΔP Curves

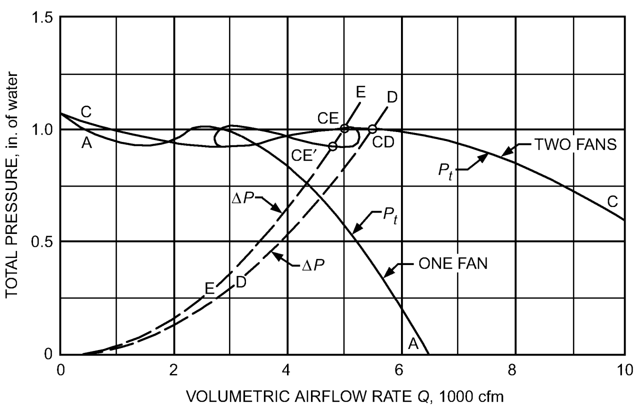


Fig. 15 Two Forward-Curved Centrifugal Fans in Parallel Operation

system requirements can be satisfied with each fan operating at a different point of rating. For example, consider ΔP curve E-E, which requires a pressure of 1.00 in. of water and a volume of 5000 cfm. The requirements of this system can be satisfied with each fan delivering 2500 cfm at 1.00 in. of water pressure, Point CE. The system can also be satisfied at Point CE′ with one fan operating at 1400 cfm at 0.9 in. of water, while the second fan delivers 3400 cfm at the same 0.9 in. of water.

Note that system curve E-E passes through the combined performance curve at two points. Under such conditions, unstable operation can result. Under conditions of CE′, one fan is underloaded and operating at poor efficiency. The other fan delivers most of the system requirements and uses substantially more power than the underloaded fan. This imbalance may reverse and shift the load from one fan to the other.

NOISE

Fan noise is a function of the fan design, volume airflow rate Q, total pressure P_t, and efficiency η_t. After a decision has been made regarding the proper type of fan for a given application (keeping in mind the system effects), the best size selection of that fan must be based on efficiency, because the most efficient operating range for a specific line of fans is normally the quietest. Low outlet velocity does not necessarily ensure quiet operation, so selections made on this basis alone are not appropriate. Also, noise comparisons of different types of fans, or fans offered by different manufacturers, made on the basis of rotational or tip speed are not valid. The only valid basis for comparison are the actual sound power levels generated by the different types of fans when they are all producing the required volume airflow rate and total pressure. Sound power level data should be obtained from the fan manufacturer for the specific fan being considered.

The data are reported by fan manufacturers as sound power levels in eight octave bands. These levels are determined by using a reverberant room for the test facility and comparing the sound generated by the fan to the sound generated by a reference source of known sound power. The measuring technique is described in AMCA *Standard* 300, Reverberant Room Method for Sound Testing of Fans. ANSI/ASHRAE *Standard* 68 (AMCA *Standard* 330), Laboratory Method of Testing to Determine the Sound in a Duct, describes an alternative test to determine the sound power a duct fan radiates into a supply and/or return duct terminated by an anechoic chamber. These standards do not fully evaluate the pure tones generated by some fans; these tones can be quite objectionable when they are radiated into occupied spaces. On critical installations, special allowance should be made by providing extra sound attenuation in the octave band containing the tone.

Discussions of sound and sound control may be found in Chapter 7 of the 2005 *ASHRAE Handbook—Fundamentals* and Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

VIBRATION

All rotating elastic structures, including fans, have certain operating speeds (known as **critical speeds**) at which resonances tend to cause objectionable vibrations. These critical speeds correspond to the various natural frequencies of the fan structure. Vibrations may be induced in a fan by unbalance, motor torque pulsations, and aerodynamic forces. Balancing fans with a frequency corresponding to the operating speed is desirable, but it is impossible to eliminate these forces completely.

Critical speeds are not solely the property of the shaft. Bearings, supports, foundations, and soil conditions all contribute to the elastic properties of any given system. Characteristics of the supporting system should be specified when a critical speed is calculated. Computer programs are available for calculating critical speeds, and manufacturers should be consulted. In axial-flow fans, the dynamic properties of the blades are of particular concern (see ANSI/ASHRAE *Standard* 87.2- 2002, In-Situ Method of Testing Propeller Fans for Reliability).

Vibration Isolation

During fan operation, some net force is transmitted to the supporting structure, making the supporting structure part of the vibrating system. Although many fans can be installed without vibration isolation, the system must be carefully designed and good balance maintained. Vibration isolation is required whenever vibrations in the supporting structure are annoying or destructive. More information on vibration and applying vibration isolation may be found in Chapter 7 of the 2005 *ASHRAE Handbook—Fundamentals* and Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

ARRANGEMENT AND INSTALLATION

Direction of rotation is determined from the drive side of the fan. On single-inlet centrifugal fans, the drive side is usually considered the side opposite the fan inlet. AMCA has published standard nomenclature to define positions.

Fan Isolation

In air-conditioning systems, ducts should be connected to fan outlets and inlets with unpainted canvas or other flexible material. Access should be provided in the connections for periodic removal of any accumulations tending to unbalance the rotor. When operating against high resistance or when low noise levels are required, it is preferable to locate the fan in a room removed from occupied areas or in a room acoustically treated to prevent sound transmission. The lighter building construction common today makes it desirable to mount fans and driving motors on resilient bases designed to prevent vibration transmission through floors to the building structure. Conduits, pipes, and other rigid members should not be attached to fans. Noise that results from obstructions, abrupt turns, grilles, and other items not connected with the fan may be present. Treatments for such problems, as well as the design of sound and vibration absorbers, are discussed in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

CONTROL

In many heating and ventilating systems, the volume of air handled by the fan varies. The proper method for varying airflow for any particular case is influenced by two basic considerations: (1) the frequency with which changes must be made and (2) balancing reduced power consumption against increases in first cost.

To control airflow, the characteristic of either the system or the fan must be changed. The system characteristic curve may be altered by installing dampers or orifice plates. This technique reduces airflow by increasing the system pressure required and, therefore, increases power consumption. Figure 10 shows three different system curves, A, B, and C, such as would be obtained by changing the damper setting or orifice diameter. Dampers are usually the lowest-first-cost method of achieving airflow control; they can be used even in cases where essentially continuous control is needed.

Changing the fan characteristic (P_t curve) for control can reduce power consumption. From this standpoint, the most desirable control method is to vary the fan's rotational speed to produce the desired performance. If change is infrequent, belt-driven units may be adjusted by changing the pulley on the drive motor of the fan. Variable-speed motors or variable-speed drives, whether electrical or hydraulic, may be used when frequent or essentially continuous variations are desired. When speed control is used, the revised P_t curve can be calculated with the fan laws.

Inlet vane control is frequently used. Figure 16 illustrates the change in fan performance with inlet vane control. Curves A, B, C, D, and E are the pressure and power curves for various vane settings between wide open (A) and nearly closed (E).

Tubeaxial and vaneaxial fans are made with adjustable-pitch blades to permit balancing the fan against the system or to make infrequent adjustments. Vaneaxial fans are also produced with controllable-pitch blades (i.e., pitch that can be varied while the fan is in operation) for frequent or continuous adjustment. Varying pitch angle retains high efficiencies over a wide range of conditions. The performance shown in Figure 17 is from a typical vaneaxial fan with variable-pitch blades. From the standpoint of noise, variable speed is somewhat better than variable blade pitch; however, both of these control methods give high operating efficiency control and generate appreciably less noise than inlet vane or damper control.

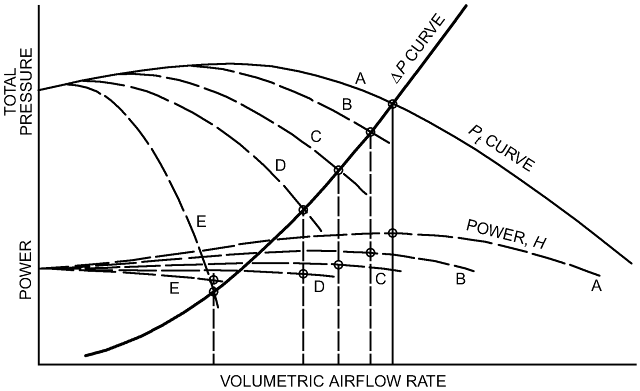


Fig. 16 Effect of Inlet Vane Control on Backward-Curved Centrifugal Fan Performance

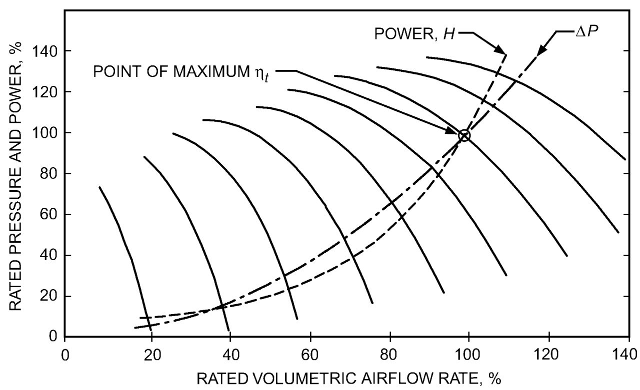


Fig. 17 Effect of Blade Pitch on Controllable-Pitch Vaneaxial Fan Performance

SYMBOLS

A = fan outlet area, ft^2
C_p = constant in Equation (1)
c_p = specific heat in Equation (1), $Btu/lb_m \cdot °F$
D = fan size or impeller diameter
J = mechanical equivalent of heat, $ft \cdot lb_f/Btu$
N = rotational speed, revolutions per minute
Q = volume airflow rate moved by fan at fan inlet conditions, cfm
P_{tf} = fan total pressure: fan total pressure at outlet minus fan total pressure at inlet, in. of water
P_{vf} = fan velocity pressure: pressure corresponding to average velocity determined from volume airflow rate and fan outlet area, in. of water
P_{sf} = fan static pressure: fan total pressure diminished by fan velocity pressure, in. of water. Fan inlet velocity head is assumed equal to zero for fan rating purposes.
P_{sx} = static pressure at given point, in. of water
P_{vx} = velocity pressure at given point, in. of water
P_{tx} = total pressure at given point, in. of water
ΔP = pressure change, in. of water
ΔT = temperature change, °F
V = fan inlet or outlet velocity, fpm
W_o = power output of fan: based on fan volume flow rate and fan total pressure, horsepower
W_i = power input to fan: measured by power delivered to fan shaft, horsepower
η_t = mechanical efficiency of fan (or fan total efficiency): ratio of power output to power input ($\eta_t = W_o/W_i$)
η_s = static efficiency of fan: mechanical efficiency multiplied by ratio of static pressure to fan total pressure, $\eta_s = (P_s/P_t)\eta_t$
ρ = gas (air) density, lb/ft^3

REFERENCES

AMCA. 1997. Laboratory method of testing to determine the sound power in a duct. ANSI/AMCA *Standard* 330-97 (ANSI/ASHRAE *Standard* 68-1997). Air Movement and Control Association, Arlington Heights, IL.

AMCA. 2002. Fans and systems. *Publication* 201-02. Air Movement and Control Association, Arlington Heights, IL.

AMCA. 1996. Reverberant room method for sound testing of fans. *Standard* 300-96. Air Movement and Control Association, Arlington Heights, IL.

AMCA. 1999. Laboratory methods of testing fans for aerodynamic performance rating. ANSI/AMCA *Standard* 210-99 (ANSI/ASHRAE *Standard* 51-1999). Air Movement and Control Association, Arlington Heights, IL.

ASHRAE. 1997. Laboratory method of testing in-duct sound power measurement procedure forms. ANSI/ASHRAE *Standard* 68-1997 (ANSI/AMCA *Standard* 330-86).

ASHRAE. 1999. Laboratory methods of testing fans for aerodynamic performance rating. ANSI/ASHRAE *Standard* 51-1999 (ANSI/AMCA *Standard* 210-99).

ASHRAE. 2002. In-situ method of testing propeller fans for reliability. ANSI/ASHRAE *Standard* 87.2-2002.

ASHRAE. 2001. Methods of testing propeller fan vibration—Diagnostic test methods. ANSI/ASHRAE *Standard* 87.3-2001.

Clarke, M.S., J.T. Barnhart, F.J. Bubsey, and E. Neitzel. 1978. The effects of system connections on fan performance. *ASHRAE Transactions* 84(2):227.

Graham, J.B. 1966. Fan selection by computer techniques. *Heating, Piping and Air Conditioning* (April):168.

Graham, J.B. 1972. Methods of selecting and rating fans. ASHRAE Symposium *Bulletin* SF-70-8, Fan Application—Testing and Selection.

BIBLIOGRAPHY

AMCA. 2003. Drive arrangements for centrifugal fans. *Standard* 99-2404-03. Air Movement and Control Association, Arlington Heights, IL.

AMCA. 2003. Designation for rotation and discharge of centrifugal fans. *Standard* 99-2406-03. Air Movement and Control Association, Arlington Heights, IL.

AMCA. 2003. Motor positions for belt or chain drive centrifugal fans. *Standard* 99-2407-03. Air Movement and Control Association, Arlington Heights, IL.

AMCA. 2003. Drive arrangements for tubular centrifugal fans. *Standard* 99-2410-86-03. Air Movement and Control Association, Arlington Heights, IL.

Howden Buffalo. 1999. *Fan engineering*, 9th ed. R. Jorgensen, ed. Howden Buffalo, Camden, NJ.

Related Commercial Resources