

CHAPTER 21

FANS

<i>Types of Fans</i>	21.1	<i>System Effects</i>	21.8
<i>Principles of Operation</i>	21.1	<i>Selection</i>	21.9
<i>Testing and Rating</i>	21.4	<i>Parallel Fan Operation</i>	21.10
<i>Field Testing of Fans for</i>		<i>Series Fan Operation</i>	21.10
<i>Air Performance</i>	21.5	<i>Noise</i>	21.11
<i>Fan Laws</i>	21.5	<i>Vibration</i>	21.11
<i>Fan and System Pressure Relationships</i>	21.6	<i>Arrangement and Installation</i>	21.12
<i>Temperature Rise Across Fans</i>	21.7	<i>Fan Control</i>	21.12
<i>Duct System Characteristics</i>	21.7	<i>Symbols</i>	21.13

AFAN uses a power-driven rotating impeller to move air. The impeller does work on the air, imparting to it both static and kinetic energy, which vary in proportion, depending on the fan type.

1. TYPES OF FANS

Fans are generally classified as centrifugal, axial, mixed, or cross 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.

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.

2. 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.

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

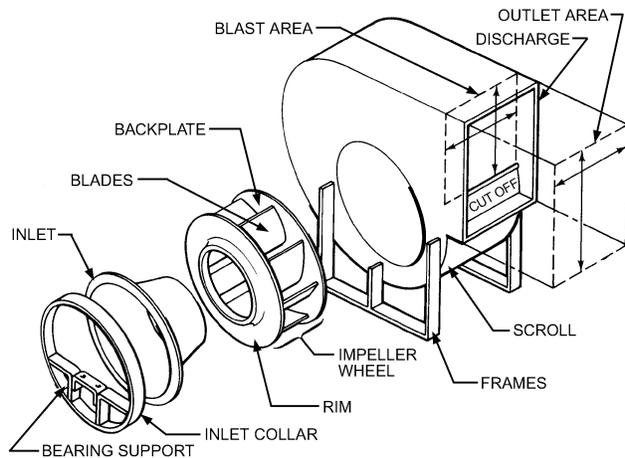
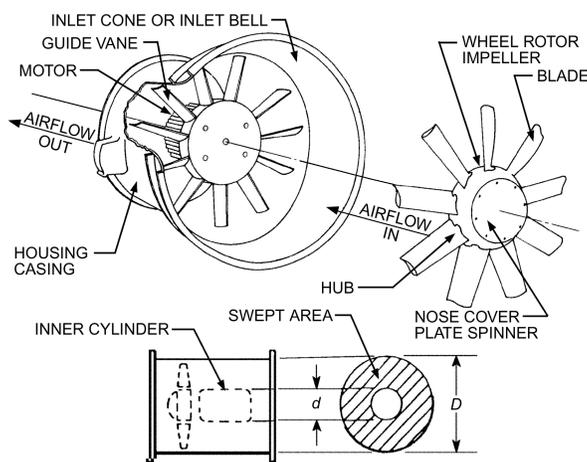


Fig. 1 Centrifugal Fan Components

Source: Adapted from AMCA Publication 201-02, Fans and Systems, with written permission from Air Movement and Control Association International, Inc.



$$\text{SWEPT AREA RATIO} = 1 - \frac{d^2}{D^2} = 1 - \frac{\text{AREA OF INNER CYLINDER}}{\text{OUTLET AREA OF FAN}}$$

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

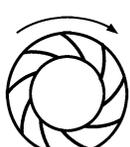
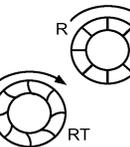
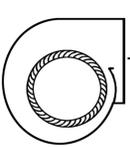
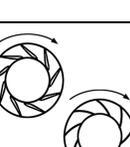
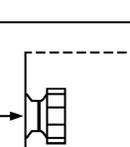
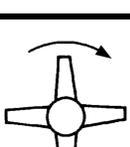
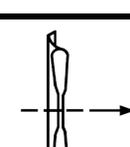
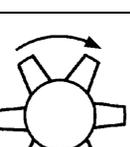
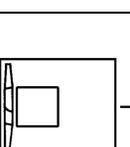
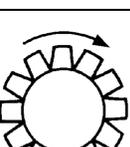
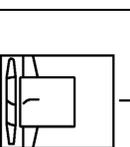
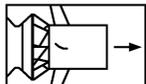
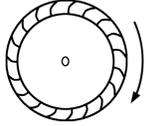
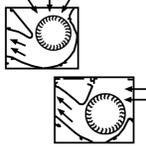
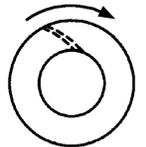
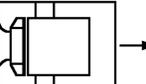
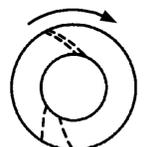
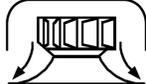
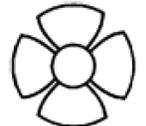
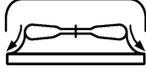
Type	Impeller Design	Housing Design
Centrifugal Fans	 <p>Blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. Air leaves impeller at velocity less than tip speed. For given duty, has highest speed of centrifugal fan designs.</p>	 <p>Scroll design for efficient conversion of velocity pressure to static pressure. Maximum efficiency requires close clearance and alignment between wheel and inlet.</p>
	 <p>Single-thickness blades curved or inclined away from direction of rotation. Efficient for same reasons as airfoil fan.</p>	 <p>Uses same housing configuration as airfoil design.</p>
	 <p>Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. Curve may have a break to left of peak pressure.</p>	 <p>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.</p>
	 <p>Flatter pressure curve and lower peak efficiency than the airfoil, backward-curved, and backward-inclined.</p>	 <p>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.</p>
	 <p>Plenum and plug fans typically use airfoil, backward inclined, or backward curved impellers in a single inlet configuration. Relative benefits of each impeller are the same as those described for scroll housed fans.</p>	 <p>Plenum and plug fans are unique in that they operate with no housing. The equivalent of a housing, or plenum chamber (dashed line), depends on the application. The components of the drive system for the plug fan are located outside the airstream.</p>
Axial Fans	 <p>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.</p>	 <p>Simple circular ring, orifice plate, or venturi. Optimum design is close to blade tips and forms smooth airfoil into wheel.</p>
	 <p>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.</p>	 <p>Cylindrical tube with close clearance to blade tips.</p>
	 <p>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.</p>	 <p>Cylindrical tube with close clearance to blade tips. Guide vanes upstream or downstream from impeller increase pressure capability and efficiency.</p>

Table 1 Types of Fans (Continued)

Performance Curves*	Performance Characteristics	Applications
	<p>Highest efficiency of all centrifugal fan designs and peak efficiencies occur at 50 to 60% of wide-open volume.</p> <p>Fan has a non-overloading characteristic, which means power reaches maximum near peak efficiency and becomes lower, or self-limiting, toward free delivery.</p>	<p>General heating, ventilating, and air-conditioning applications.</p> <p>Usually only applied to large systems, which may be low-, medium-, or high-pressure applications.</p> <p>Applied to large, clean-air industrial operations for significant energy savings.</p>
	<p>Similar to airfoil fan, except peak efficiency slightly lower.</p> <p>Curved blades are slightly more efficient than straight blades.</p>	<p>Same heating, ventilating, and air-conditioning applications as airfoil fan.</p> <p>Used in some industrial applications where environment may corrode or erode airfoil blade.</p>
	<p>Higher pressure characteristics than airfoil and backward-curved fans.</p> <p>Pressure may drop suddenly at left of peak pressure, but this usually causes no problems.</p> <p>Power rises continually to free delivery, which is an overloading characteristic.</p> <p>Curved blades are slightly more efficient than straight blades.</p>	<p>Primarily for materials handling in industrial plants.</p> <p>Also for some high-pressure industrial requirements.</p> <p>Rugged wheel is simple to repair in the field. Wheel sometimes coated with special material.</p> <p>Not common for HVAC applications.</p>
	<p>Pressure curve less steep than that of backward-curved fans. Curve dips to left of peak pressure.</p> <p>Highest efficiency occurs at 40 to 50% of wide-open volume.</p> <p>Operate fan to right of peak pressure. Use caution when selecting left of peak pressure, because instability is possible.</p> <p>Power rises continually to free delivery which is an overloading characteristic.</p>	<p>Primarily for low-pressure HVAC applications, such as residential furnaces, central station units, and packaged air conditioners.</p>
	<p>Plenum and plug fans are similar to comparable housed airfoil/backward-curved fans but are generally less efficient because of inefficient conversion of kinetic energy in discharge airstream.</p> <p>They are more susceptible to performance degradation caused by poor installation.</p>	<p>Plenum and plug fans are used in a variety of HVAC applications such as air handlers, especially where direct-drive arrangements are desirable.</p> <p>Other advantages of these fans are discharge configuration flexibility and potential for smaller-footprint units.</p>
	<p>High flow rate, but very low pressure capabilities.</p> <p>Maximum efficiency reached near free delivery.</p> <p>Discharge pattern circular and airstream swirls.</p>	<p>For low-pressure, high-volume air-moving applications, such as air circulation in a space or ventilation through a wall without ductwork.</p> <p>Used for makeup air applications.</p>
	<p>High flow rate, medium pressure capabilities.</p> <p>Pressure curve dips to left of peak pressure. Avoid operating fan in this region.</p> <p>Discharge pattern circular and airstream rotates or swirls.</p>	<p>Low- and medium-pressure ducted HVAC applications where air distribution downstream is not critical.</p> <p>Used in some industrial applications, such as drying ovens, paint spray booths, and fume exhausts.</p>
	<p>High-pressure characteristics with medium-volume flow capabilities.</p> <p>Pressure curve dips to left of peak pressure. Avoid operating fan in this region.</p> <p>Guide vanes correct circular motion imparted by impeller and improve pressure characteristics and efficiency of fan.</p>	<p>General HVAC systems in low-, medium-, and high-pressure applications where straight-through flow and compact installation are required.</p> <p>Has good downstream air distribution.</p> <p>Used in industrial applications in place of tubeaxial fans.</p> <p>More compact than centrifugal fans for same duty.</p>

Table 1 Types of Fans (Concluded)

Type	Impeller Design	Housing Design
Mixed-Flow Mixed-Flow	 Combination of axial and centrifugal characteristics. Ideally suited in applications in which the air has to flow in or out axially. Higher pressure characteristic than axial fans.	 The majority of mixed-flow fans are in a tubular housing and include outlet turning vanes. Can operate without housing or in a pipe and duct.
Cross-flow Cross-flow (Tangential)	 Impeller with forward-curved blades. During rotation the flow of air passes through part of the rotor blades into the rotor. This creates an area of turbulence which, working with the guide system, deflects the airflow through another section of the rotor into the discharge duct of the fan casing. Lowest efficiency of any type of fan.	 Special designed housing for 90° or straight through airflow.
Other Designs Power Roof Ventilators	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.
	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.
	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. Hood protects fan from weather and acts as safety guard.	 Essentially a propeller fan mounted in a supporting structure. Air discharges from annular space at bottom of weather hood.

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.

3. 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. The most common type of test uses multiple nozzle inlet or outlet chambers. Figure 3 illustrates a pitot traverse procedure for developing characteristics of a fan. Fan performance is determined from free delivery conditions to shutoff conditions. At shutoff, the fan is completely blocked off; at free delivery, outlet resistance is reduced to zero. Between these two conditions, an auxiliary fan and various airflow restrictions are used to simulate various operating conditions on the fan. Sufficient points are obtained to define the curve between shutoff and free air delivery conditions. 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 a similar performance curve, except for fans that produce a significant amount of swirl.

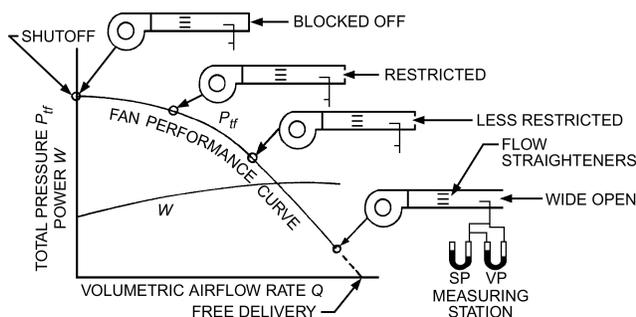
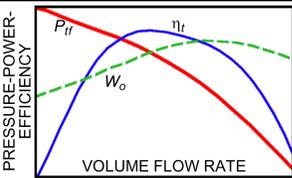
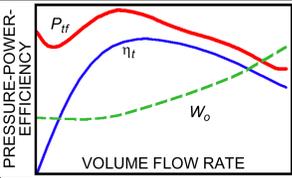
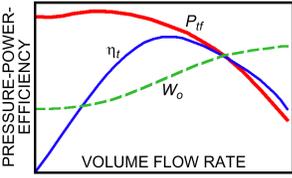
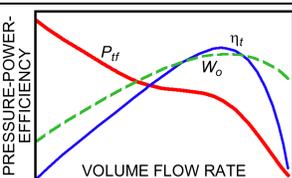
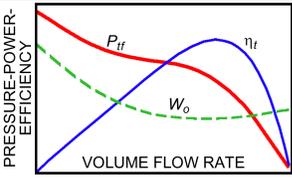


Fig. 3 Method of Obtaining Fan Performance Curves

Fans designed for use with duct systems are tested with a length of duct between the fan and measuring station. The length of duct evens out the air velocity profile discharged from the fan outlet to provide stable, uniform airflow conditions at the plane of measurement. Pressure losses of the ductwork and flow straightener between the fan outlet and the plane of measurement are added to the measured pressure at the plane of measurement to determine the actual fan performance. Fans designed for use without ducts, including almost all propeller fans and power roof ventilators, are tested without ductwork.

Table 1 Types of Fans (Concluded)

Performance Curves*	Performance Characteristics	Applications
	Characteristic pressure curve between axial fans and centrifugal fans. Higher pressure than axial fans and higher volume flow than centrifugal fans.	Similar HVAC applications to centrifugal fans or in applications where an axial fan cannot generate sufficient pressure rise.
	Similar to forward-curved fans. Power rises continually to free delivery, which is an overloading characteristic. Unlike all other fans, performance curves include motor characteristics. Lowest efficiency of any fan type.	Low-pressure HVAC systems such as fan heaters, fireplace inserts, electronic cooling, and air curtains.
	Performance similar to backward-curved fan, except capacity and pressure are 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.	Centrifugal units are somewhat quieter than axial flow units. 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.
	Usually operated without ductwork; therefore, operates at very low pressure and high volume.	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.

Not all fan 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. Test information can also be used to calculate the performance at other speeds by applying fan laws. 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, engineering data, or Howden Buffalo (1999).]

4. FIELD TESTING OF FANS FOR AIR PERFORMANCE

The aerodynamic performance of a fan as installed almost always differs from the performance determined by laboratory testing. Performance differences primarily derive from system configuration differences between the field and laboratory setup, such as added elbows, obstructions in the path of the airflow, and sudden changes of duct cross-sectional area in the field installation.

Because of the performance differences, it is sometimes necessary to determine the in situ performance of a fan. Typical reasons fan field testing that may be required include the following:

- A general fan system evaluation to be used as the basis for modifying or adjusting fan drive components or the system to which the fan is attached
- A fan acceptance test (FAT) per specification in a sales agreement to verify quoted fan performance
- A proof of performance test in response to a complaint to demonstrate fan performance

In North America, most general fan system evaluations in the field can be accomplished using the guidelines outlined in AMCA *Publication 203*. When field testing is required and more accurate results are needed to meet a stringent contract, AMCA *Standard 803* or ASME *Code PTC 11* is often used. In Europe and other areas of the world, ISO *Standard 5802* is sometimes used. Because finding a suitable flow measurement plane can often be difficult in the field, it is highly recommended to include provisions for a calibrated flow measuring station near the fan inlet and/or outlet or pressure ports on the fan as part of system design.

5. 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

Table 2 Fan Laws

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

Notes:
 1. Subscript 1 denotes fan under consideration. Subscript 2 denotes tested fan.
 2. For all fan laws $(\eta_1)_1 = (\eta_1)_2$ and $(\text{Point of rating})_1 = (\text{Point of rating})_2$.
 3. P equals either P_{sf} , P_{tf} , or P_{sf} .
 4. See Howden Buffalo (1999) for other considerations (e.g., compressibility effects are typically ignored for fan total pressure rises of less than 10 in. of water).

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.

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/ft³). 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_{tf} 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,

$$Q_2 = 6000 \text{ cfm and } P_{tf2} = 1.13 \text{ in. of water}$$

Using Fan Law 1a at point E,

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

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

$$P_{tf1} = 1.13(650/600)^2 = 1.33 \text{ in. of water}$$

The total pressure curve P_{tf1} 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 used to calculate point F on the base curve. The entire base curve must be defined by test.

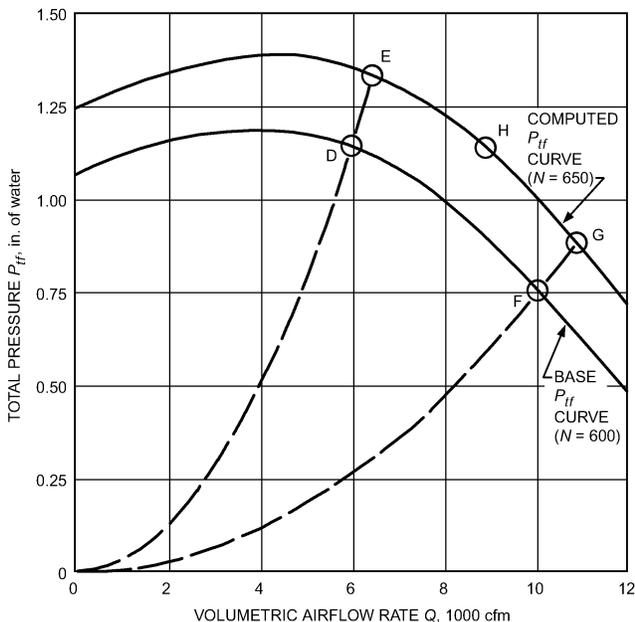


Fig. 4 Example Application of Fan Laws

6. 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 21 of the 2013 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. 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, 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 shown 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 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} .

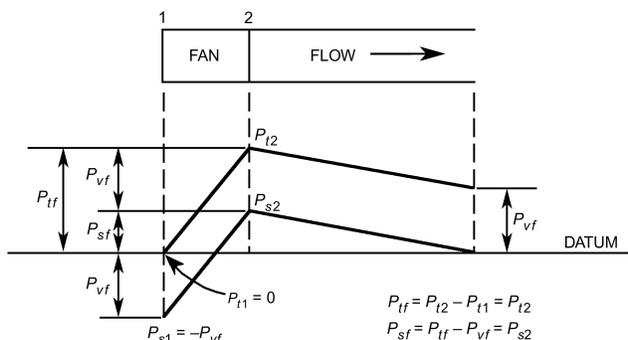


Fig. 5 Pressure Relationships of Fan with Outlet System Only

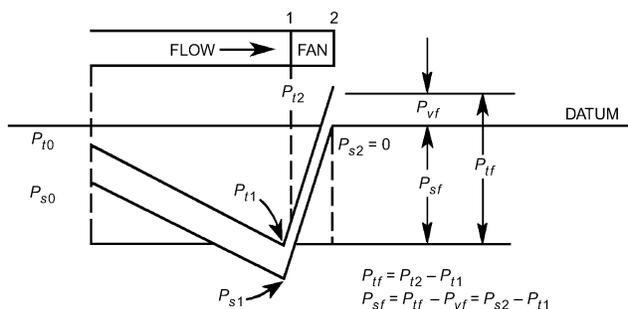


Fig. 6 Pressure Relationships of Fan with Inlet System Only

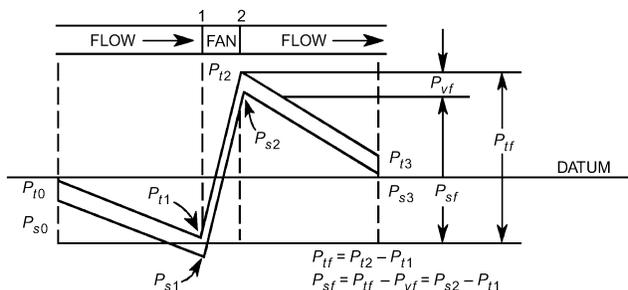


Fig. 7 Pressure Relationships of Fan with Equal-Sized Inlet and Outlet Systems

7. 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), estimate the temperature rise 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/ft²·in. of water
- ρ = density, lb_m/ft³
- c_p = specific heat = 0.24 Btu/lb_m·°F
- J = mechanical equivalent of heat = 778.2 ft·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).

8. 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.

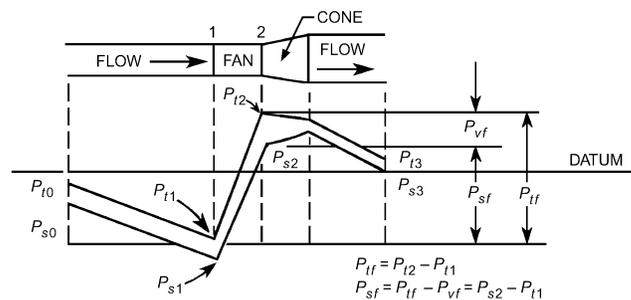


Fig. 8 Pressure Relationships of Fan with Diverging Cone Outlet

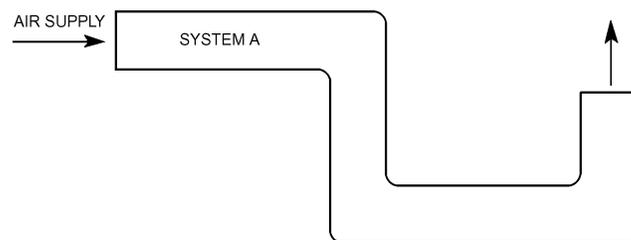


Fig. 9 Simple Duct System with Resistance to Flow Represented by Three 90° Elbows

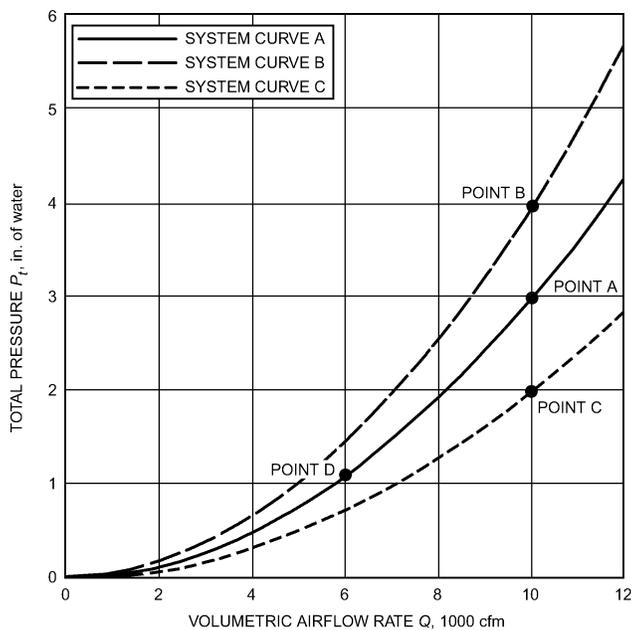


Fig. 10 Example System Total Pressure Loss (ΔP) Curves

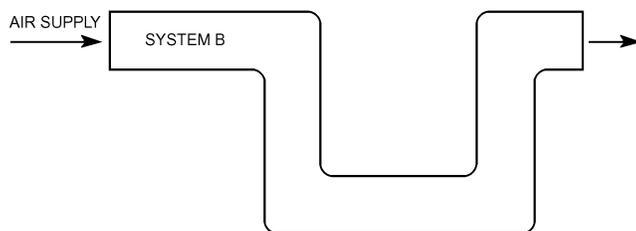


Fig. 11 Resistance Added to Duct System of Figure 9

Equation (2) allows 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 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.

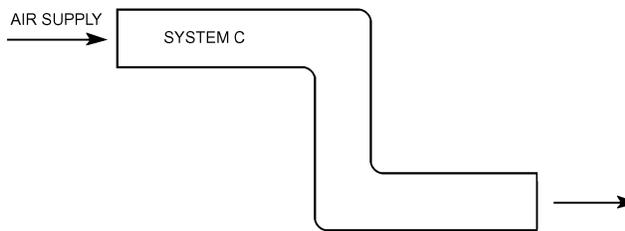


Fig. 12 Resistance Removed from Duct System of Figure 9

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 shown here. Such curves can also represent the clogging of turbulent flow filters in a system.

9. SYSTEM EFFECTS

A fan is normally tested under standardized laboratory conditions (e.g., wide-open inlet and a long, straight duct attached to the outlet), which result in uniform flow into the fan and efficient static pressure recovery on the fan outlet. When a fan is installed in an application, laboratory conditions are usually not preserved. Typically, inlet and outlet conditions in installations are not the same as in the testing environment, because the fan discharges to a plenum, the discharge/intake side of the fan is too close to a transition (change in area and direction of flow) or wall, or any other interference in the flow field. This difference affects the actual fan performance tested.

The adverse influences of system connections on fan performance are commonly called system effects. To select and apply the fan properly, the system effects must be considered and the pressure requirements of the fan, as calculated by standard duct design procedures, must be increased accordingly. More importantly, because of the huge potential pressure (and thus energy) losses associated with system effects, great care should be taken in actual air system design and installation to eliminate or minimize system effects.

The magnitudes of system effects are typically called **system effect factors**, and they are values (usually in terms of pressure) suggested to compensate for the system effects. Chapter 21 of the 2013 *ASHRAE Handbook—Fundamentals* and ASHRAE’s (2011) *Duct Fitting Database* provide information on calculating the system effect factors for certain duct fittings. AMCA *Publication 201* provides further information on determining system effect factors for various conditions. 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 air system design.

To provide improved knowledge and additional test data on system effects, ASHRAE completed a series of research projects to study the inlet system effects on both air and sound for different types of fans. Conclusions from these research projects include the following:

- Substantial system effects will be induced if the distance between the fan inlet and the cabinet wall is less than 0.5 times of the

impeller diameter; hence, it should be avoided as much as possible in actual system design and installation.

- System effects on sound caused by poor inlet conditions are very pronounced but also very hard to quantify.

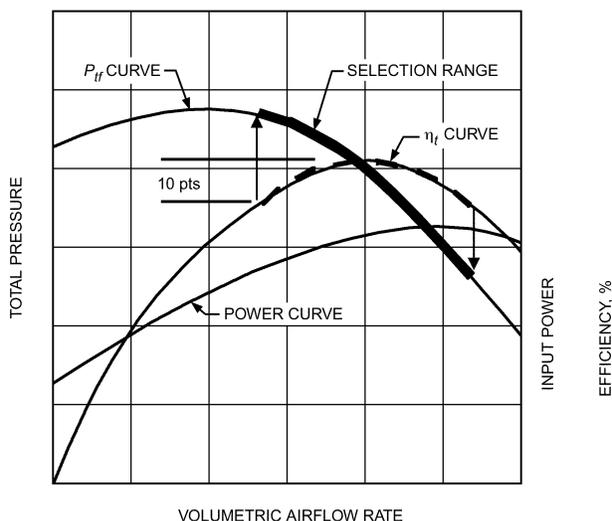
For more details, please see Darvennes et al. (2008), Stevens and Schubert (2010), and Swim (1997). ASHRAE research projects RP-1216 and RP-1420 (Guedel et al. 2011, 2014) also provide further information on inlet and discharge installation effects on airfoil centrifugal fans.

10. 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 graphics (curves) (Figure 13), tabular form (multirating tables), or selection software. 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.

One of the most important selection criteria for fans is energy use. In recent years, fan energy consumption has been addressed by several energy codes and regulations. Take care to ensure that fans selected for a particular purpose meet applicable codes and standards. For example, ASHRAE *Standard* 90.1-2013 requires fans in certain applications to meet a minimum **fan efficiency grade (FEG)**. The FEG is a metric used to characterize fan energy efficiency and is defined in ANSI/AMCA *Standard* 205-12 and ISO *Standard* 12759:2010. Fans that do not meet the minimum FEG requirement cannot be considered for selection. The FEG is usually reported with manufacturers' performance data or electronic selection tools. If not, consult the fan manufacturer. Fans that comply with a minimum FEG requirement must be selected so that the difference between the peak total efficiency and the total efficiency at the selection point is within a prescribed value (e.g., 10 points, as shown in Figure 13).

Codes and regulation may also apply to the extended fan system, which includes a motor and other mechanical or electric components. In Europe, for example, European Directive EU No.327/2011: *Ecodesign-Fans* requires the extended fan system to meet a minimum efficiency metric that is similar to **fan motor efficiency grade (FMEG)**. The FMEG is a wire-to-air efficiency metric and is defined in ISO 12759:2010 *Fans-Efficiency Classification for Fans*.



Curve shows performance of a fixed fan size running at a fixed speed.

Fig. 13 Fan Performance Curve

Determination of FMEG is based on overall fan system peak efficiency, electrical input power, efficiency category, and fan type.

The FEG and FMEG are separate and distinct metrics for describing fan efficiency. They must only be used to establish compliance with applicable codes or regulations, and not as a means to make fan selections or to quantify actual fan performance in a given application.

Performance data as tabulated in typical manufacturers' fan performance 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 characteristics; they cannot be obtained from each other by the fan laws. However, points of operation listed in fan performance 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 total pressure characteristic fit the system total pressure characteristic. The system total pressure characteristic must also account for any transition needed between the fan inlet/outlet and the system. Because fan performance can change depending on fan installation (e.g., ducted or unducted inlet or outlet), it is important that the system total pressure characteristic and fan performance characteristic refer to the same installation. 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 21 of the 2013 *ASHRAE Handbook—Fundamentals* and Chapter 19 of this volume). Fan speed and power requirements are then calculated, using multirating tables or single or multispeed performance curves or graphs.

Energy usage is a function of fan total pressure. Fan performance and system resistance must be specified as total pressure versus flow for proper selection. For many years, manufacturers often have presented fan performance as static pressure versus flow, leading many engineers to select fans by matching fan static pressure to the sum of system static pressure requirements. Although this practice is long standing, it is incorrect. To ensure the best chance to minimize energy consumption, total pressure must be used. Because many catalogs and selection programs use static pressure, it is necessary to calculate the total pressure: this can be done by adding the fan velocity pressure, which can easily be calculated from the flow and the outlet area, to the static pressure.

Fan manufacturers provide catalogs or electronic tools for making fan selections. Fan performance data (e.g., airflow rate, pressure rise, shaft power) are typically presented over a recommended range of fan operating conditions. When the fan is selected within this recommended range, airflow over the aerodynamic surfaces, such as the inlet cone and impeller blades, smoothly follows the surfaces in a way that results in good aerodynamic efficiency. However, when the fan operates outside of this recommended range, airflow may be unable to follow these surfaces and it breaks away, or separates, resulting in a number of undesirable effects. This condition, generally known as **stall**, is characterized by extensive regions of separated flow, increased noise, and highly unsteady behavior in the key flow variables (airflow rate, pressure rise, and shaft power). The latter effect is known as **surge**. Fans operating under these conditions are subject to mechanical damage because of the large unsteady forces involved.

Different fan types exhibit stall and surge to varying degrees. Care must be taken when selecting a fan to ensure that operation does not extend beyond the recommended boundaries provided by the fan manufacturer. Fan manufacturers provide a recommended selection range for each fan to avoid stall and flow pulsations. For further information about stall, refer to Eurovent (2007).

In using curves, the point of operation selected (Figure 14) must represent a desirable point on the fan curve, to attain maximum efficiency and resistance to stall and pulsation. In systems where more than one point of operation is encountered during operation, 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.

Special attention must be given to selecting the proper motor and also related drive components, such as belts and variable-speed drives, when used. The motor and other component ratings must be sufficient to allow operation at all anticipated points of operation. Fans with overloading power characteristics and fans operating at elevated temperatures present areas of concern. Variable-volume systems may require special consideration if operated before system balancing.

11. PARALLEL FAN OPERATION

When two identical fans with **stable** performance characteristics throughout the range of their individual performance curves operate in parallel, ideally their combined performance curve can be created by doubling the airflow of a single fan at any pressure

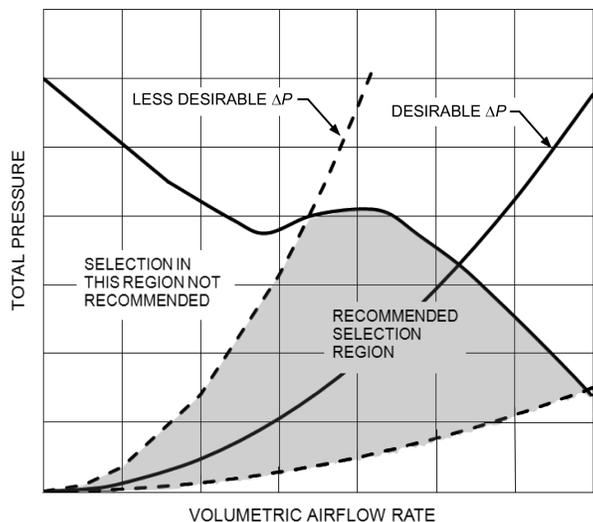
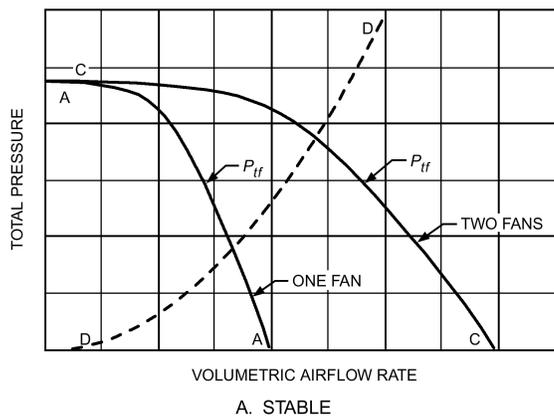
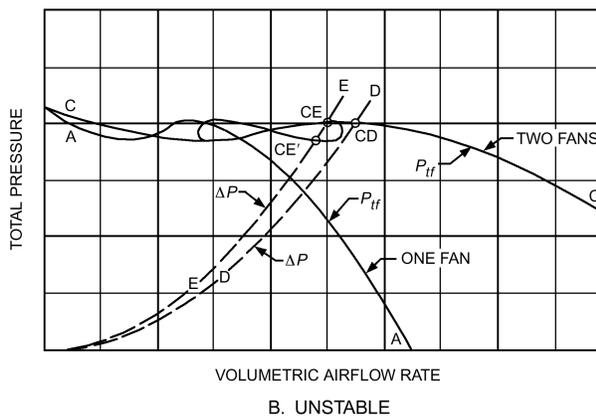


Fig. 14 Desirable Combination of P_{tf} and ΔP Curves



A. STABLE



B. UNSTABLE

Fig. 15 Two (A) Stable and (B) Unstable Fans in Parallel Operation

level. Examples of such fans include backward-inclined centrifugal fans and axial-flow fans operated with a low pitch blade angle. Figure 15A illustrates a combined performance curve of two fans with stable operating characteristics.

When two fans with **unstable** performance characteristics (i.e., a pressure reduction to the left of the peak pressure point) operate in parallel, an unstable flow condition may occur if the fans operate too close to the peak pressure point of the combined performance curve. Figure 15B illustrates the combined performance curve of two fans operating in parallel.

Curve A-A of Figure 15B represents the pressure characteristic of a single fan. Curve C-C is the combined performance of two fans operating in parallel. All operating points to the right of point CD, the peak pressure point, are the sum of two times the airflow values for a single fan at the same pressure. The points are along the stable operating range of each fan. All systems with a resistance curve intersecting the combined performance curve to the right of point CD, such as resistance curve D-D, have stable flow characteristics.

Points to the left of point CD are the sum of all airflow values possible at the same pressure level of the fans. The points are along the unstable operating range of each fan. The double-loop shape (∞) seen in Figure 15B to the left of the peak pressure of two-fan operation is formed by summing the multiple airflows possible at a single pressure level. Systems with a resistance curve intersecting the performance curve in this region, to the left of point CD (e.g., resistance curve E-E), have unstable flow characteristics. Unstable flow is shown by the intersection of the system resistance curve at multiple points on the combined fan performance curve, points CE and CE'. The fans oscillate between these two points with random changes in airflow, noise, and vibration levels.

Avoid operating at unstable conditions. Always select fans to operate in their stable range, well to the right of the peak pressure point subject to the guidance in the section on Selection. If possible, plot the combined fan performance curve, including the double loop, and select the fan operating point along the stable portion of the performance curve to the right of the double loop.

When three or more identical fans are installed in parallel, the resulting airflow is the individual fan performance multiplied by the number of fans in the system, for properly selected and installed fans. The designer must ensure that the operating point of each fan is well to the right of the peak to avoid oscillation of any one of the fans. Also, the fan configuration plays a major part in how well the system performs.

12. SERIES FAN OPERATION

Two identical fans operating in series theoretically double the pressure rise without changing airflow (Figure 16). Actual performance

of the two fans operating in series will be less than the theoretical, because losses occur in the transition between the two fans and the second fan operates less efficiently. Lack of sufficient spacing and/or straightening of flow between the fans further reduces performance. At fan pressures below 30 in. of water, air compressibility can be neglected. Above that limit, calculation of the operating point, fan material selection, and fan design should consider temperature rise caused by friction and compressible gases.

13. NOISE

Fan noise is a function of the fan design, volume airflow rate Q , total pressure P_{tf} , and efficiency η_t . After the proper type of fan for a given application has been determined (keeping in mind the system effects), the best size selection of that fan is commonly 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. Obtain octave-band sound power level data from the fan manufacturer for the specific fan being considered.

Sound power levels L_w can be determined using several different methods, such as a reverberant room comparing the sound generated by the fan to the sound generated by a reference source of known sound power; using an anechoic room; or using sound intensity measurements. The reverberant room measuring technique is described in ANSI/AMCA Standard 300; the enveloping surface method, which uses an anechoic room, is described in ISO Standard 13347-3; and the sound intensity method is described in ANSI/

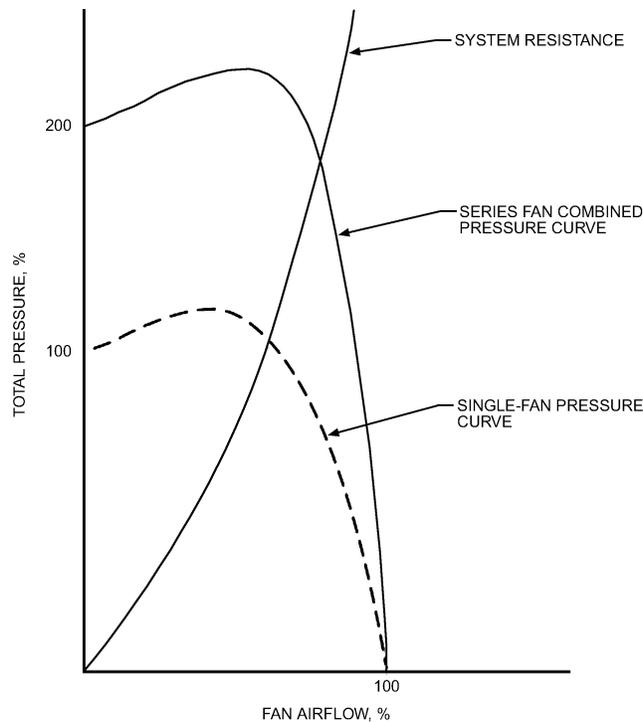


Fig. 16 Theoretical Characteristic Curve of Two Fans Operating in Series

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AMCA Standard 320. ANSI/ASHRAE Standard 68 (AMCA Standard 330) 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 radiate into occupied spaces. On critical installations, make special allowance by providing extra sound attenuation in the octave band containing the tone.

Discussions of sound and sound control may be found in Chapter 8 of the 2013 ASHRAE Handbook—Fundamentals and Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications.

14. VIBRATION

Fan vibration is the structural response of a fan to excitations caused by impeller imbalance, unsteady aerodynamic forces, and drive torque pulsations. The magnitude and extent of the structural response, or vibration level, is determined by the stiffness of the fan components, drive alignment, and bearing properties, among other factors.

Excessive vibration levels can lead to premature failure of the fan, produce high noise levels, and transmit undesirable forces into the support structure. Although fan vibration can never be entirely eliminated, acceptable levels can be achieved through proper fan design, manufacture, and application.

Acceptable vibration levels have been established from a long history of practical fan experience. ANSI/AMCA Standard 204 defines recommended fan balance and vibration levels based on the fan application and fan drive power. The fan balance and vibration (BV) category, shown in Table 3, determines acceptable levels for fan balance quality and vibration levels.

Fan balance quality, or balance grade (e.g., G 6.3), is used to establish the maximum residual imbalance specification for a given fan speed and size. Balance grades for each of the BV categories are shown in Table 4.

Acceptable vibration levels, as measured in in/s, are shown in Table 5. These levels are generally obtained using an accelerometer placed in the vicinity of the fan or motor. A newly commissioned fan would be expected to meet the start-up values given in the table. If

Table 3 Fan Application Categories for Balance and Vibration

Application	Examples	Driver Power Limits, hp	Fan Balance and Vibration (BV) Application Category
Residential	Ceiling fans, attic fans, window AC	≤0.2	BV-1
		>0.2	BV-2
HVAC and agricultural	Building ventilation and air conditioning; commercial systems	≤5.0	BV-2
		>5.0	BV-3
Industrial processes, power generation, etc.	Baghouse, scrubber, mine, conveying, boilers, combustion air, pollution control, wind tunnels	≤400	BV-3
		>400	BV-4
Transportation and marine	Locomotives, trucks, automobiles	≤20	BV-3
Transit/tunnel	Subway emergency ventilation, tunnel fans, garage ventilation	≤100	BV-3
		>100	BV-4
Petrochemical process	Hazardous gases, process fans	All	BV-4
		≤50	BV-3
Computer chip manufacture	Cleanroom	>50	BV-4
		All	BV-5

Source: Reprinted from AMCA Standard 204-05, Balance Quality and Vibration Levels for Fans, with written permission from Air Movement and Control Association International, Inc.

Table 4 BV Categories and Balance Quality Grades

Fan Application Category	Balance Quality Grade for Rigid Rotors/Impeller
BV-1*	G 16
BV-2	G 16
BV-3	G 6.3
BV-4	G 2.5
BV-5	G 1.0

*Note: In category BV-1, there may be some extremely small fan rotors weighing less than 0.5 lb. In such cases, residual unbalance may be difficult to determine accurately. The fabrication process must ensure reasonably equal weight distribution about the axis of rotation.

Source: Reprinted from AMCA Standard 204-05, Balance Quality and Vibration Levels for Fans, with written permission from Air Movement and Control Association International, Inc.

Table 5 Seismic Vibration Velocity Limits for In Situ Operation

Condition	Fan Application Category	Rigidly Mounted, in/s	Flexibly Mounted, in/s
Start-up	BV-1	0.55	0.60
	BV-2	0.30	0.50
	BV-3	0.25	0.35
	BV-4	0.16	0.25
	BV-5	0.10	0.16
Alarm	BV-1	0.60	0.75
	BV-2	0.50	0.75
	BV-3	0.25	0.65
	BV-4	0.40	0.40
	BV-5	0.20	0.30
Shutdown	BV-1	*	*
	BV-2	*	*
	BV-3	0.50	0.70
	BV-4	0.40	0.60
	BV-5	0.30	0.40

Source: Reprinted from AMCA Standard 204-05, Balance Quality and Vibration Levels for Fans, with written permission from Air Movement and Control Association International, Inc.

Values shown are peak velocity, in/s, filter out.

*Shutdown levels for fans in fan application grades BV-1 and BV-2 must be established based on historical data.

alarm levels are reached during operation, appropriate steps should be taken to identify, contain, and correct the source of the vibration. The fan should be taken out of service if vibration levels exceed the shutdown values. Operation at these levels can lead to premature failure of the fan.

Acceptable vibration performance at the fan design conditions can be achieved by following these recommendations. However, with the widespread use of variable-speed control, fans may still produce excessive vibration levels at certain *critical* speeds that correspond to natural frequencies of the fan and/or fan support structure. These critical speeds should be avoided, or eliminated through proper design of the fan components.

The vibration level of a fan changes with fan operating condition and time. Therefore, it is important to monitor, or periodically check, the fan vibration level to ensure safe and reliable operation.

Vibration Isolation

During fan operation, vibration is transmitted to the support structure and building. This can lead to objectionable noise in occupied spaces, or to undesirable vibration in other components of the HVAC system (e.g., the ductwork). Vibration isolation (e.g., coil springs, rubber-in-shear) can be used to attenuate the vibration reaching the building. More information on vibration and application of vibration isolation can be found in Chapter 8 of the 2013

ASHRAE Handbook—Fundamentals and Chapter 48 of the 2015 ASHRAE Handbook—HVAC Applications.

15. 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. The AMCA Standard 99 series defines standard nomenclature for fan arrangements.

For a duct connected to a fan outlet or inlet, a flexible connection should be used to minimize vibration transmission. Provide access to the fan wheel 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 away from occupied areas or acoustically treated to prevent sound transmission. The lighter-mass 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 48 of the 2015 ASHRAE Handbook—HVAC Applications.

16. FAN CONTROL

Many fan applications require the air volume to vary in response to ventilation requirements. Control strategy selection is based on several considerations, including frequency of airflow changes, effects on fan energy consumption, and first cost of control device(s).

Airflow control can be achieved by changing the system characteristic or the fan characteristic. Methods that change the system characteristic have low first cost, but generally consume more energy compared to methods that affect the fan characteristic.

The system characteristic can be altered by installing dampers or orifice plates. This approach reduces airflow by increasing the system restriction. In general, the resulting input power is higher than that of a comparable fan correctly selected for the new airflow operating point. Dampers are usually the lowest-first-cost method of achieving airflow control and are sometimes used in cases where continuous control is needed, although other, more energy-efficient means are available.

Inlet vanes are a form of damper control that offer better energy efficiency. Inlet vanes are typically a series of triangular vanes located in the inlet of a fan. The vanes are controlled through a linkage mechanism, similar to a damper, with one important difference: the vanes are oriented such that incoming flow is turned to provide a preswirl into the fan impeller. In many cases, this can improve fan performance and offset the energy penalty of the vanes. Figure 17 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).

Changing the fan characteristic (P_f curve) for airflow control can reduce power consumption. One option is to vary the fan's rotational speed to produce the desired performance. If the change is infrequent, the speed of belt-driven fans may be adjusted by changing the drive pulley combination. More frequent changes using belt drive can be accomplished with adjustable sheaves that are manually, electrically, or hydraulically actuated. An alternative method for speed control, and the only option for direct-driven fans, is to vary the motor speed with a variable-speed control, such as a variable-frequency drive (VFD) or similar electronic device. Electronic motor speed control is often the best option from an energy consumption standpoint. When using variable-speed controls, the fan characteristic can be calculated with the fan laws.

An alternative to speed control is to change the fan geometry (especially blade setting angles) to optimize aerodynamic efficiency. This approach is more common in axial fans than with centrifugal fans.

Tubeaxial and vaneaxial fans are available with adjustable-pitch blades to allow balancing the airflow system, or to make infrequent adjustments. Vaneaxial fans can also be 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 18 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 control methods give high operating efficiency and generate less noise than inlet vanes or dampers.

Table 6 summarizes control strategies and their relative energy consumption and first cost.

Table 6 Summary of Control Strategies

Control Type	Control Method	Energy Consumption	First Cost
System characteristic	Orifice plates	High	Very low
	Dampers	High	Low
	Inlet vanes	Moderate	Medium
Fan characteristic	Variable sheaves	Moderate	High
	Motor speed (belt)	Moderate	High
	Motor speed (direct)	Low	High
	Variable pitch (axial)	Low	Very High

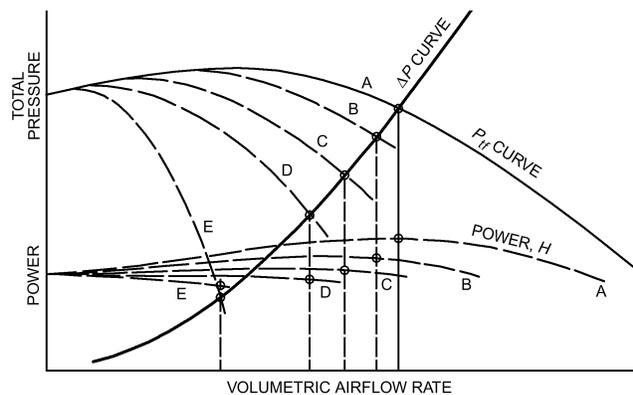


Fig. 17 Effect of Inlet Vane Control on Backward-Curved Centrifugal Fan Performance

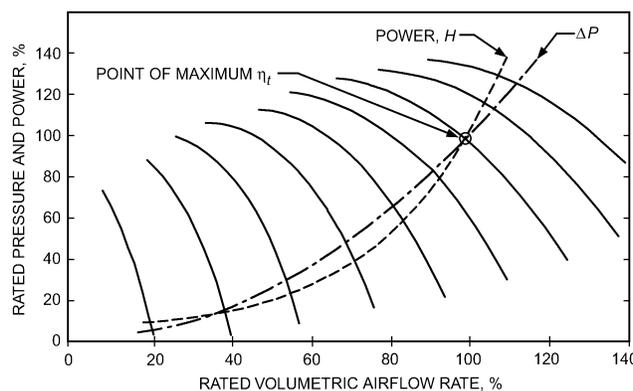


Fig. 18 Effect of Controlled Blade Pitch on Vaneaxial Fan Performance

Further information on fan control as it relates to specific HVAC distribution systems may be found in Chapter 45 of this volume and in Chapter 47 of the 2015 ASHRAE Handbook—HVAC Applications.

17. SYMBOLS

- A = fan outlet area, ft²
- C_p = constant in Equation (1)
- c_p = specific heat in Equation (1), Btu/lb_m·°F
- D = fan size or impeller diameter
- d = area of inner cylinder
- J = mechanical equivalent of heat, ft·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 static pressure is also the difference between static pressure at outlet and total pressure at inlet.
- 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, $\eta_s = (P_{sf}/P_{tf})\eta_t$
- ρ = gas (air) density, lb/ft³

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