PDHonline Course M213 (4 PDH)

Characteristics and Selection Parameters of Fans and Blower Systems

Instructor: A. Bhatia, B.E.

2012

PDH Online | PDH Center

5272 Meadow Estates Drive
Fairfax, VA 22030-6658
Phone & Fax: 703-988-0088
www.PDHonline.org
www.PDHcenter.com

An Approved Continuing Education Provider

HVAC – Characteristics and Selection Parameters of Fans and Blower Systems

Course Content

Fans and blowers provide air for ventilation and industrial process requirements. They are differentiated by the method used to move the air, and by the system pressure they must operate against. As a general rule, fans typically operate at pressures up to about 55 in-wg (2psi) and blowers at between 2psi and 28psi, although custom-designed fans and blowers may operate well above these ranges. Air compressors are used for systems requiring more than 20psi.

There are two broad categories of fans: centrifugal and axial. The fundamental difference between the two is that the centrifugal fans discharge air perpendicular to the axis of the impeller rotation whereas an axial fan discharges air parallel to the axis of the impeller rotation.

Principle of Operation

Axial – Operate on the principle of deflection of airflow by the fan blade. The pressure is produced simply by the lift of the rotating blades as air moves over it. These types are characterized by the path of the airflow through the fan.

Centrifugal –Operating principle is a combination of deflection by the blades as well as the centrifugal force exerted on the air rotating with the wheel and thereby moving outward in a radial direction.

Axial and centrifugal fans have overlapping capabilities in terms of pressure, airflow, and efficiency; however, usually they are not interchangeable. For the same impeller diameter and speed, a centrifugal fan produces more static pressure than an axial flow fan.

Types of Fans

Centrifugal	Axial	Special Designs
Forward Curved (FC)	Propeller	Roof ventilators
Radial Fans (RF)	Tubeaxial	Inline Centrifugal
Backward Inclined (BI)	Vaneaxial	Plug / Plenum fans
Backward Curved (BC)		Blowers
Aerofoil Bladed (AF)		

CENTRIFUGAL FANS

Key Characteristics

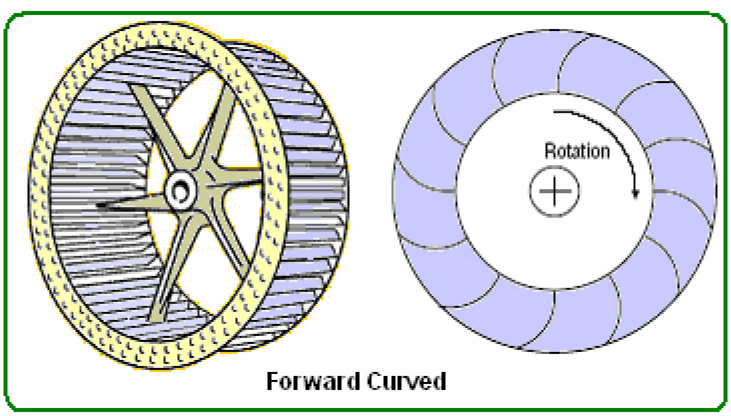
1) A centrifugal fan moves air and creates pressure partly by centrifugal action and partly by rotating velocity. The centrifugal forces are created by a rotational air column, which is enclosed between the blades of the fans.

2) Air flow can be varied to match air distribution system requirements by simple adjustments to the fan drive or control devices
3) Typically exhibit non-overloading characteristics except for forward curved fan
4) Commonly used in applications which require higher pressures
5) Air flow through a centrifugal fan cannot be reversed
6) Predominant choice for ducted applications because of their quite operation and availability in the low to medium pressure range of HVAC applications
7) Frequently used in "dirty" airstreams (high moisture and particulate content), in material handling applications, and in systems at higher temperatures
8) Can be constructed to accommodate harsh operating conditions
9) Fans for industrial ventilation typically operate against pressures up to 22 in-wg; heavy-duty fans can achieve 55 in-wg or more
10) The largest fans may have impeller diameters of 200 inches or more, and may handle more than 100,000 CFM
11) Most versatile; their benefits include:
    - Compact dimensions
    - Quiet in operation
    - Ability to overcome high system resistance
    - Available in a wide range of sizes and motor speeds
    - Can be tailored to suit specific applications

The four basic types of centrifugal fans distinguished by the angle of the impeller wheel blades.

Forward-curved fans

*S*ometimes called "squirrel cage" fans, have blades with leading edge curved toward the direction of rotation.



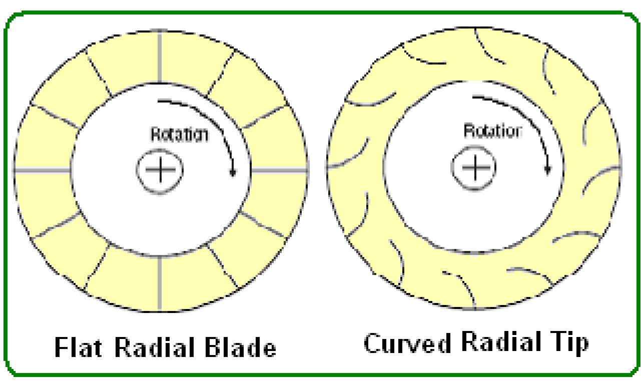
1) Usually contain 24 to 64 impeller blades
2) Air leaves the impeller at velocities greater than the impeller tip speed i.e. air tip velocity > wheel peripheral velocity
3) Best suited for moving large volumes at low static pressures for the same speed and wheel size

4) Operate at lower speed than other centrifugals, which make them quiet
5) Smaller size relative to other fan types
6) Lighter in construction and less expensive
7) Because these fans generate high airflow at relatively low speeds, these require a relatively accurate estimate of the system airflow and pressure demand. If, for some reason, system requirements are uncertain, then an improper guess at fan rotational speed can cause under-performance or excessive airflow and pressure.
8) Characterized by relatively low efficiency between 55 and 65 percent and is somewhat less than airfoil and backward-curved fans.
9) Well suited for low pressure heating, ventilating, and air conditioning applications such as domestic furnaces, central station units, and packaged air conditioning equipment from room type to roof top units.
10) Exhibit ‘overloading’ power curve characteristic i.e. the power increases steadily with airflow toward free delivery; consequently, careful driver selection is required to avoid overloading the fan motor.
11) Fan output is difficult to adjust accurately and these fans are not used where airflow must be closely controlled.
12) Usable only with clean air applications because blades easily accumulate dirt
13) Not constructed for high pressures or harsh service
14) Not recommended for fumes or dusts that would stick to the short curved blades because they would cause unbalance and would make cleaning difficult

---

Radial fans

Radial fans have blades, which extend straight from the shaft, and typically have 6 to 16 blades.



1) Simplest of all centrifugal fans and the least efficient. Radial fans have efficiencies of 50 - 65 percent
2) Operate at medium speed and can move air against higher pressures than other centrifugal fans. Pressures up to 55in-wg are common; custom-designed fans can achieve pressures over 100in-wg
3) Produce high sound levels with pronounced blade passage tones at high pressure applications
4) Usually have medium tip speed and noise factor and are used for buffing exhaust, woodworking exhaust, or for applications where a heavy dust load passes through the fan
5) Exhibit ‘overloading’ power characteristic i.e. horsepower rises with increasing air quantity in an almost directly proportional relation

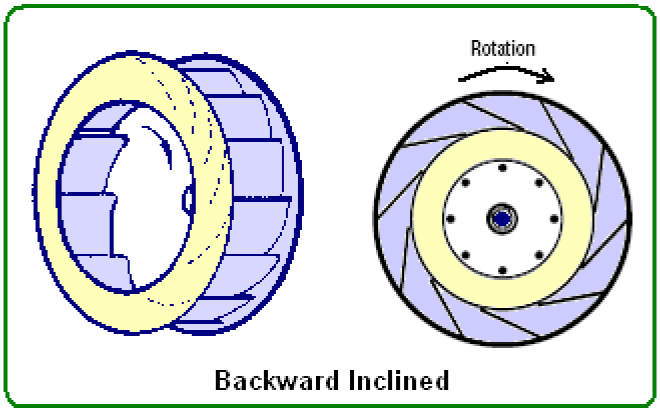
6) Less prone to clogging and minimize contaminant build-up. Especially suited for conveying "dirty" airstreams and in material handling applications

7) Because of their simple blade shape, these can be ruggedly constructed with expensive alloys that are strong and corrosion resistant, or they can be less expensively constructed with fiberglass-reinforced plastic or coated with a corrosion resistant material. These are widely used in corrosive applications and in high-temperature environments

---

Backward inclined & Backward Curved Fans

Backward curved fans have blades with leading edge curved or inclined away from the direction of rotation.



1) Backward curved fans have about 6 to 16 blades

2) Backward inclined blade impellers produce air velocity slower than blade tip speed, but require greater rotational speed than forward curved types

3) These fans run faster and require heavier, well-balanced construction. Larger shaft and bearing sizes are required for higher speeds. Because of this, proper wheel balance is more important.

4) Up to 75 - 80 percent efficient. The high operating efficiencies available from this fan type can provide low system life-cycle costs

5) Normally used for high capacity, high pressure applications where power savings may outweigh its higher first cost

6) A common application for backward-inclined fans is forced-draft service. In these applications, the fan is exposed to the relatively clean airstream on the upstream side of the process

7) Exhibit 'non-overloading' power characteristics. The motor brake horsepower increases with airflow for most of the performance curve but drops off at high airflow rates. Because of this characteristic, this fan type is often selected when system behavior at high airflow rates is uncertain

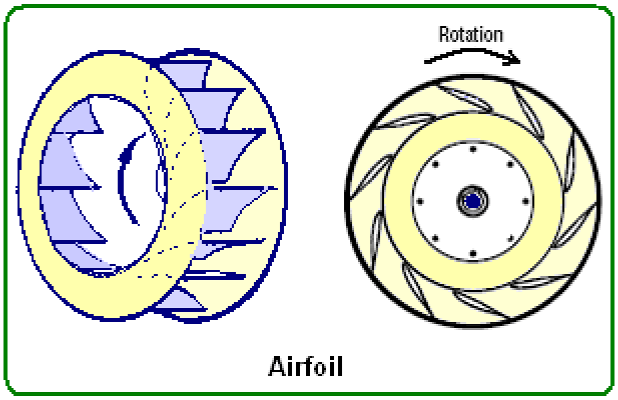
8) Backward inclined shape promotes low velocity air across the blades and is susceptible to contaminant build-up. Only recommended for clean air streams containing no condensable fumes or vapors.

9) Centrifugal fans of the backward-inclined blade design are of three general types: flat, curved and airfoil. Flat blade types are more robust while the curved-blade fans tend to be more efficient. Airfoil blades are the most efficient of all, capable of achieving efficiencies exceeding 85 percent.

10) At the cost of more maintenance and some increase in mechanical noise, the airfoil and backward inclined fan types generate lower sound levels overall.

Airfoil fans

A variation of the backward inclined fan constructed of aerodynamic hollow profile blade, which makes these quieter and more efficient.



1) Most efficient of all fan types and achieves the static efficiency is around 86-92 percent
2) Have lower rotating mass due to thin wall blade construction, which makes them aerodynamically efficient. However, this thin walled characteristic makes this fan type highly susceptible to erosion problems
3) For the given duty, the airfoil impeller design will provide for the highest speed of the centrifugal fan designs
4) Airfoil fans have the highest initial cost and are suitable for clean air applications only. Airfoil blade is not recommended in corrosive and/or erosive environment unless blades are suitable coated.

In summary,

Parameters	Backward Curve			Forward Curve
	BC	BI	AF	FC
Blades	6-16	6-16	6-16	24-64
Maximum Efficiency (%)	78	85	90	70
Speed	High	High	High	Low
Cost	Medium	Medium	High	Med-Low
Static Pressure	Very high	High	Very high (40in-wg)	Low (5 in- wg)
Power Curve	Non-overloading	Non-overloading	Non-overloading	Overloading
Housing	Scroll	Scroll	Scroll	Scroll

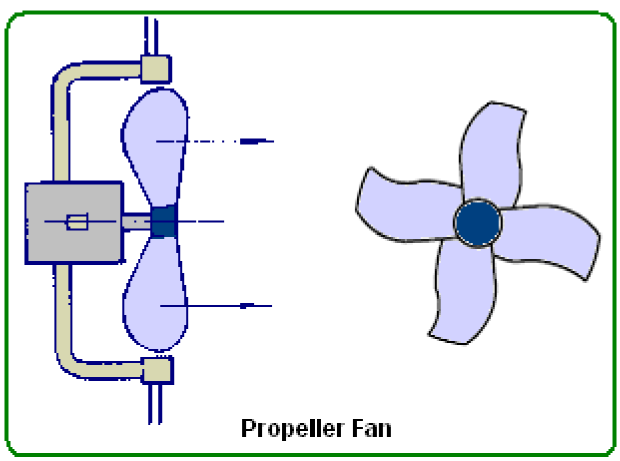
AXIAL FANS

Key Characteristics

1) Axial fans, as the name implies, move air parallel to the shaft, or axis, of the fan.
2) Axial-flow fans impart energy to the air by giving it a twisting motion. The air is pressurized by the aerodynamic lift generated by the fan blades, much like a propeller and an airplane wing.
3) Typically provide large air volumes at relatively low pressures.
4) Key advantages are compactness, low cost, and lightweight.
5) Axial fans are frequently used in exhaust applications where airborne particulate size is small, such as dust streams, smoke, and steam.
6) Less bulky than a centrifugal fan of comparable capacity and have the advantage of straight-through airflow
7) Typically designed to generate flow in one direction but they can operate in the reverse direction. This characteristic is useful when a space may require contaminated air to be exhausted or fresh air to be supplied.
8) Axial fans must rotate faster than comparable centrifugal fans to achieve the same airflow capacity. This characteristic makes them noisier than comparable centrifugal fans. However, high noise of axial fans tends to be dominated by high frequencies, which are easier to attenuate.
9) Axial fans have a severe stall region that makes them particularly unsuitable for systems with widely varying operating conditions
10) Axial fans have non-overloading power characteristic which enables the correct motor horsepower to be used for any particular fan or application without causing a burn-out due to overload
11) Are available in both direct drive and belt drive options; belt drives offer an advantage by removing the motor from the airstream subjected to high level of contaminants otherwise for harsh applications with direct drive, sealed motor should be used.

As a general rule, axial fans are preferred for high volume, low pressure, and non-ducted systems. There are three main types 1) Propeller, 2) Tube axial and 3) Vane axial.

Propeller fans



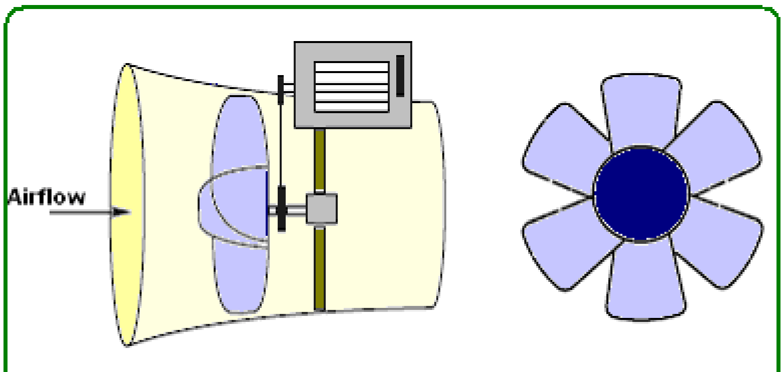
1) Have two or more blades that generate very high airflow volumes
2) Low static pressure up to ¾ in-wg
3) Can provide reversed air flow at reduced volumes and pressure by reversing the direction of rotation
4) Exhibit ‘overloading power’ characteristic i.e. the power required to drive the impeller continues to increase as the resistance to airflow increases

5) Very low efficiencies of approximately 50 percent or less
6) Light weight and inexpensive because of their simple construction
7) Noise levels of the propeller fan are only slightly higher than those of the tubeaxial and vaneaxial fans, but such of the noise is at low frequencies and therefore is difficult to attenuate.
8) Primary applications include low pressure, high volume air moving applications such as air circulation within a space or ventilation through a wall without attached ductwork.
9) Used for replacement air applications. Ideal for exhaust applications and are often used in rooftop ventilation applications. Also used for make up or replacement air applications

---

Tubeaxial fans

Tubeaxial fans have a wheel inside a cylindrical housing, with close clearance between blade and housing. Generally, the numbers of blades range from 4 to 8 with the hub normally less than 50 percent of fan tip diameter.

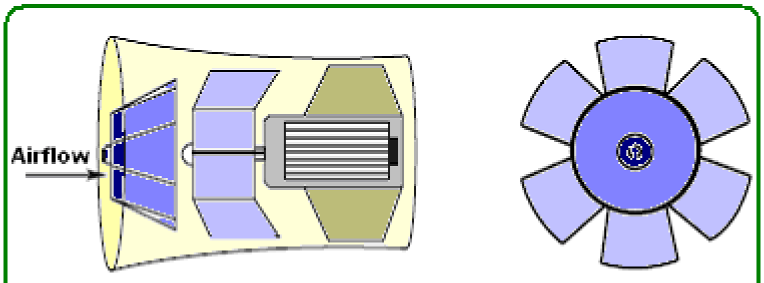


Tubeaxial Fan

1) Capable of developing a more useful static pressure range (to 10 – 15in-wg)
2) Capable of delivering operating efficiencies up to 75 percent
3) The downstream profile is uneven with a large rotation component. This airflow characteristic is accompanied by moderate airflow noise.
4) Suited for ducted applications where downstream velocity profiles is not very critical
5) Most good design blades are aerofoil shaped
6) Can be either connected directly to a motor or driven through a belt configuration. Because of the high operating speeds of 2-, 4-, and 6-pole motors, most tubeaxial fans use belt drives to achieve fan speeds below 1,100 revolutions per minute.
7) Adjustable blade pitch fans allow capacity control
8) Tubeaxial fan generates a somewhat higher noise level than the vaneaxial fan; its spectrum contains a very strong blade frequency component.
9) Used in medium-pressure, high airflow rate applications and are frequently used in exhaust applications because they create sufficient pressure to overcome duct losses and are relatively space efficient. Also used in some industrial applications such as drying ovens, paint spray booths, and fume exhaust systems.

Vaneaxial Fans

Vane-axial fans are similar to tube-axial fans with guide vanes on down streamside to improve flow profile.



Vaneaxial Fan

1) Typically have 5 to 20 aerofoil type blades with a large hub diameter.
2) Very close blade tip to housing clearance
3) Blades are fixed or adjustable pitch types and the hub is usually greater than 50 percent of the fan tip diameter
4) Very efficient. When equipped with airfoil blades and built with small clearances, they can achieve efficiencies up to 85 percent.
5) High-pressure capability up to 8 in-wg. Custom equipment is capable of 20 -40 in-wg
6) Typically, vaneaxial fans have performance curves that have unstable regions to the left of the peak pressure.
7) Typically used in medium- to high-pressure applications, such as induced draft service for a boiler exhaust.
8) Like tubeaxial fans, vaneaxial fans tend to have a low rotating mass, which allows them to achieve operating speed relatively quickly. This characteristic is useful in emergency ventilation applications where quick air removal or supply is required.
9) Vaneaxial fans generate somewhat higher noise levels than centrifugal ventilating fans of comparable output.
10) Adjustable pitch blades allow capacity control

In summary

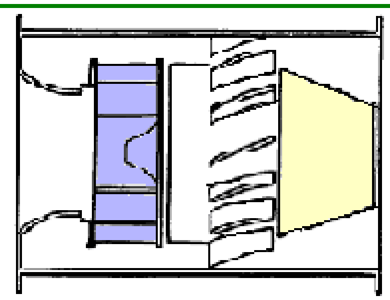
Parameters	Propellers	Tube Axial	Vane axial
Blades	2 to 8	4 to 8	5 to 20
Maximum Efficiency (%)	50	75	85
Speed	Medium	High	Very high
Cost	Low	Medium	High
Static Pressure	Low (up to ¾ in)	Medium	High (up to 8 in)

Parameters	Propellers	Tube Axial	Vane axial
Power Curve	Non-overloading	Non-overloading	Non- overloading
Housing	Annular ring	Cylindrical	Cylindrical with guide vanes on downstream side

SPECIAL FAN TYPES

Inline flow centrifugal fans

This type of fan has backward-curved blades and a special housing that permits a space-saving straight-line duct installation. The wheel is very similar to that of the airfoil. Space requirements are similar to a vane axial fan.



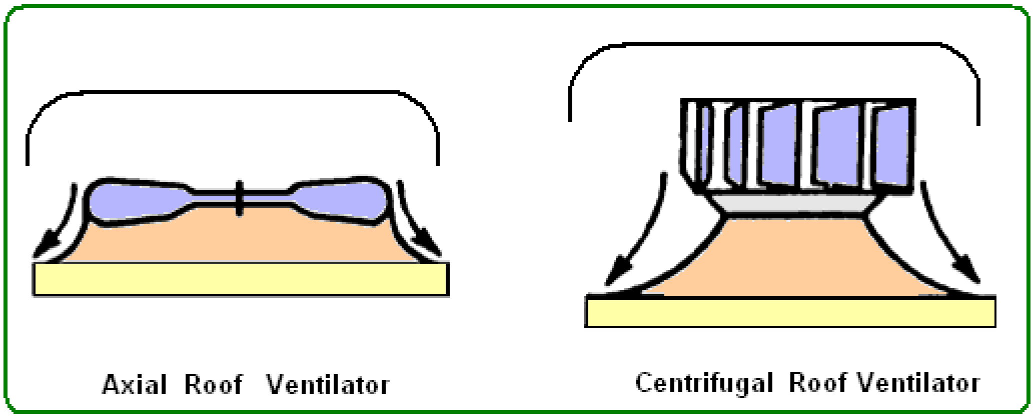
Inline Flow Centrifugal Fan

Key Characteristics

1) BC/BI impeller inside a tubular casing to produce flow inline with wheel axis and duct
2) Also known as tubular centrifugal fans
3) Absence of scroll housing makes these fans compact
4) Small size fans have impeller mounted directly on motor and placed inside a spun aluminum casing split in two halves
5) Typically used in low pressure return air or exhaust applications

Power roof ventilators

The objective of these ventilators is to produce a high-volume flow rate at low pressure. They can be of centrifugal fan type or axial fan type.



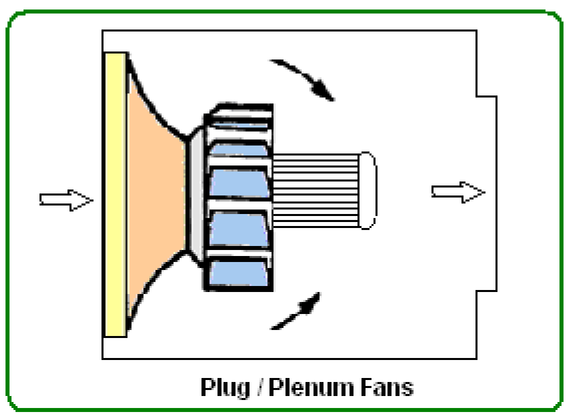
Key characteristics

1) Roof top mounted exhaust
2) BC/BI wheel or axial blade
3) Vertically mounted on motor shaft
4) Air discharged in a radial or axial direction
5) BC wheels quieter than axial blades
6) Now housing but a weather protection casing made of spun aluminum or sheet fabrication

---

Plug / Plenum Fans

Like inline centrifugal fans, plug / plenum fans also use a BC (or AF) impeller but without the tubular casing.



Key characteristics

1) Impeller / motor assemble placed inside a plenum which may contain coils and filters in case of an air-handling unit
2) Air enters through a well designed circular inlet cone and comes out of the impeller in a radial direction
3) Because of the absence of scroll housing, efficiencies are much lower than BC/BI fans
4) Lower initial cost makes them popular choice in clean room applications

---

Common Blower Types

Blowers can achieve much higher pressures than fans, as high as 20psi. They are also used to produce negative pressures for industrial vacuum systems. Blowers look more like centrifugal pumps than fans. The impeller is typically gear-driven and rotates as fast as 15,000 rpm. In multi-stage blowers air is accelerated as it passes through each impeller, although larger single-stage blowers can be more efficient because the air does not have to take as many turns.

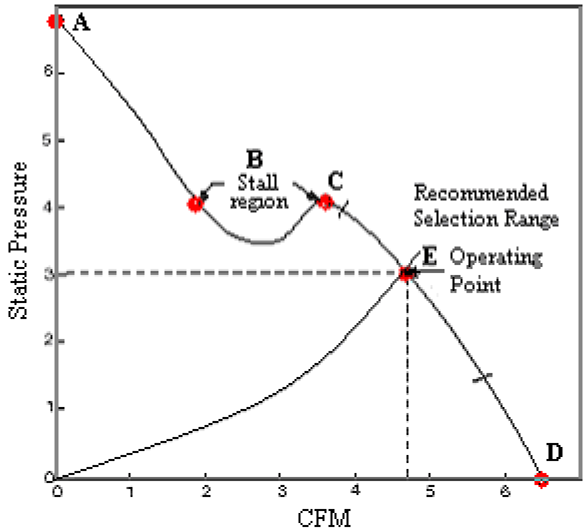
1) Centrifugal blowers typically operate against pressures of 5 to 10psi, but can achieve higher pressures. One characteristic is that airflow tends to drop drastically as system pressure increases, which can be a disadvantage in material conveying systems that depend on a steady air volume. Because of this, they are most often used in applications that are not prone to clogging.
2) Positive-displacement blowers have rotors, which "trap" air and push it through housing. A "helical screw blower" uses two screw-shaped rotors. Positive-displacement blowers provide a constant volume of air even if the system pressure varies. They are especially suitable for applications prone to clogging, since they can produce enough pressure - typically up to 18psi - to blow clogged materials free. They turn much slower than centrifugal blowers (e.g. 3,600 rpm), and are often belt driven to facilitate speed changes.

FAN LAWS & CHARACTERISTICS

The most commonly used fan characteristic is the relationship between pressure rise and volume flow rate for a constant impeller speed (RPM). The air movement has a pressure associated with it, which is termed as static pressure (SP) and the velocity pressure (VP). The static pressure (SP) is the useful working pressure available for overcoming the resistance of a ventilating system and is exerted in all directions at once weather in motion or not. SP may be +ve or –ve. Velocity pressure (VP) is the pressure due to the speed of the air and is always +ve and exerted in direction of flow. The total pressure (TP) produced by a fan is made up of the static pressure (SP) and the velocity pressure.

Fan pressure rise characteristics are normally expressed in either TP or SP, with static pressure being the unit most commonly used in the United States. The fan volume flow rate is commonly expressed in cubic feet per minute, or CFM. Therefore, the system pressure loss and volume flow rate requirements are typically expressed as a certain value of static pressure (SP) at some CFM.

The fan "pressure-volume" curve is generated by connecting the fan to a laboratory test chamber in accordance with the very specific test procedures outlined in the Air Movement and Control Association (AMCA) Standard 210. The starting point is to measure the airflow at zero pressure, sometimes referred to as free inlet & discharge or FID. The test rig is then adjusted to increase the pressure that the fan has to work against and to measure the airflow at each point. Data points are collected and plotted graphically for a constant rpm, from a "no flow" block off condition to a "full flow" or wide condition. Figure below represents such a curve that is typical for a vaneaxial fan, and is commonly referred to as a "static pressure" curve.



In the curve above:

1) Point A represents the point of zero airflow. It is frequently referred to as "block off," "shut off," "no flow," and "static no delivery."

2) Point B depicts the stall region of the static pressure curve. Operation in this area is discouraged because of erratic airflow that generates excessive noise and vibration.

3) Point C depicts what is referred to as the peak of the static pressure curve, and

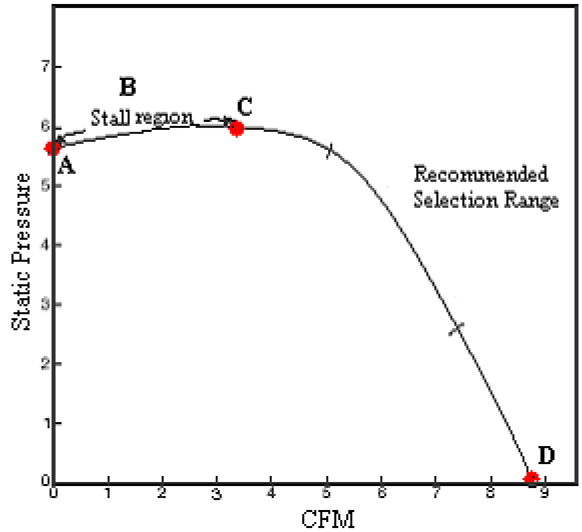
4) Point D is the point of maximum airflow. Point D is also referred to as "free delivery," "free air," "wide open performance," and "wide open volume."

Curve segment CD is often referred to as the right side of the fan curve. This is the stable portion of the fan curve and is where the fan is selected to operate. It then follows that curve segment AC is the left side of the fan curve and is considered to be the unstable portion of the curve.

The fan static pressure curve is the basis for all airflow and pressure calculations. For a given SP on the static pressure curve, there is a corresponding CFM at a given rpm. Point E on the curve represents one such point on the fan curve. Simply locate some unit of pressure on the left hand SP scale and project a horizontal line to the point of intersection on the static pressure curve. From the point of intersection, project a vertical line to the bottom CFM scale to establish the corresponding airflow at that particular speed. In this example, a SP of three units results in a CFM of 4.71 units.

Note that point E is also known as the ***"operating point"*** (point of operation or design point), which corresponds to the condition at which the fan pressure rise (SP) / volumetric flow rate (CFM) characteristic intersects at operating or design condition.

Figure above is a vaneaxial curve with a pronounced dip (stall region) that is also a typical curve shape for high angle propeller fans and forward curve centrifugal fans. In contrast, compare this curve to figure below, which represents a typical curve for a backward inclined centrifugal fan. This curve shape is also representative of radial blade centrifugal fans.

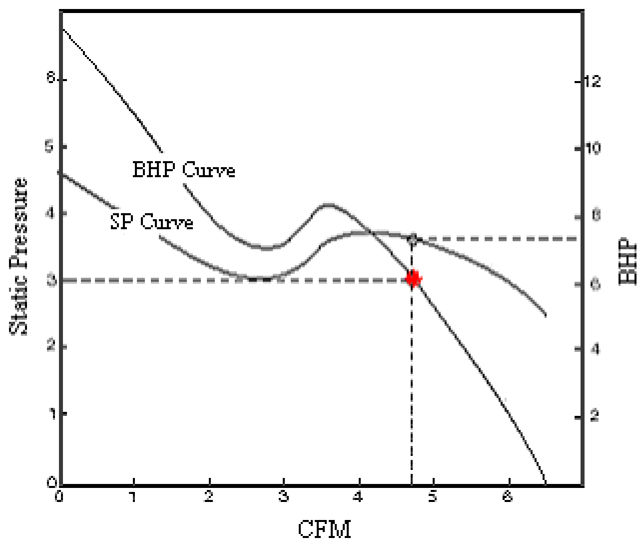


A typical curve for backward inclined centrifugal fan

Note the lack of a pronounced dip on this curve. Nevertheless, the area left of peak is also a stall region and selections in this area should be avoided.

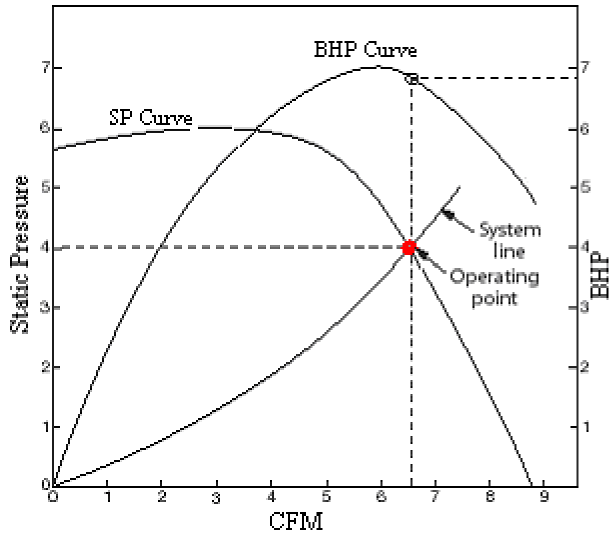
Brake Horse Power Curve

Having established a SP and airflow (CFM), an operating brake horsepower (BHP) can be established.



By adding the BHP curve to the static pressure curve, the fan performance curve is completed. To determine BHP simply extends vertically the CFM point until it intersects the BHP curve. Draw a horizontal line from this point of intersection to the right to the BHP scale to establish a BHP of 7.25 units, which corresponds to the previously established performance of 4.7 FM units at 3 SP units.

Similarly, we can add the BHP curve to the static pressure curve of the backward inclined centrifugal fan to complete that fan performance curve.



Even though the performance curves for the vaneaxial fan and the centrifugal fan have completely different shapes, the curves are read in the same way. Locate some unit of pressure on the left hand SP scale (4 units) and project a horizontal line to the point of intersection with the SP curve. Projecting downward from this point of intersection to the CFM scale, we establish airflow of 6.6units. Now project vertically upward to intersect the BHP curve. Project a horizontal line from this point to the BHP scale and read a BHP of 6.9 units.

Best Efficiency Point

Fan efficiency is the ratio of the power imparted to the airstream to the power delivered by the motor. The power of the airflow is the product of the pressure and the flow, corrected for unit's consistency. The equation for total efficiency is:

$$\text{Total Efficiency} = \frac{\text{Total Pressure x Airflow}}{\text{BHP x 6362}}$$

Where

Total pressure is in inches of water

Airflow is in cubic feet per minute (CFM)

BHP is brake horsepower

An important aspect of a fan performance curve is the best efficiency point (BEP), where a fan operates most cost-effectively in terms of both energy efficiency and maintenance considerations. Operating a fan near its BEP improves its performance and reduces wear, allowing longer intervals between repairs. Moving a fan's operating point away from its BEP increases bearing loads and noise.

Another term for efficiency that is often used with fans is static efficiency, which uses static pressure instead of total pressure in the above equation.

When evaluating fan performance, it is important to know which efficiency term is being used.

FAN LAWS

Fan laws relate the performance characteristics of any geometrically similar series of fans. CFM, RPM, SP and HP are all related to each other in a known manner and when one changes, all others change. The CFM variable is the most commonly changed measurement in an air moving system therefore the following example of Fan Law application is based on a change from an existing CFM to a new CFM. The simplified form of the most commonly used fan laws include.

$$Cfm_{final} = Cfm_{initial}\left(\frac{RPM_{final}}{RPM_{initial}}\right)$$

$$SP_{final} = SP_{initial}\left(\frac{RPM_{final}}{RPM_{initial}}\right)^2$$

$$HP_{final} = HP_{initial}\left(\frac{RPM_{final}}{RPM_{initial}}\right)^3$$

The laws governing fan operation provide a useful tool for determining requirements when changing the volume capacity of a fan is required for a given system. At a new fan rotational speed, the new operating point for a given fan can be determined from the fan laws.

Example:

A fan installed in a fixed system is operating at 10000 CFM @ 1.5"-wg, running 1000 RPM using 5 BHP. What RPM is required to move 25% more air (12500 CFM) through the system?

Solution

By rearranging the cfm fan law:

$$RPM_2 = \left(\frac{CFM_2}{CFM_1}\right) \times RPM_1$$

$$RPM_2 = \left(\frac{12{,}500}{10{,}000}\right) \times 1000 = 1250\ RPM$$

The corresponding static pressure is:

$$SP_2 = SP_1 \left(\frac{RPM_2}{RPM_1}\right)^2$$

$$SP_2 = 1.50 \left(\frac{1250}{1000}\right)^2 = 2.34''$$

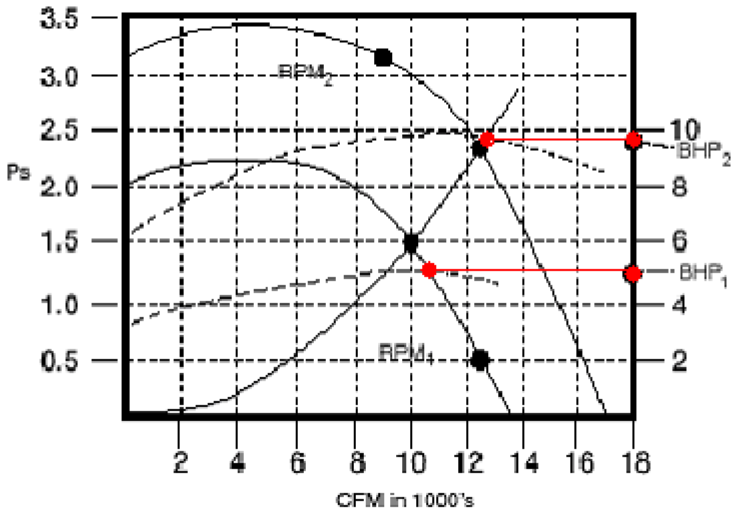
The resulting BHP is:

$$BHP_2 = BHP_1 \left(\frac{RPM_2}{RPM_1}\right)^3$$

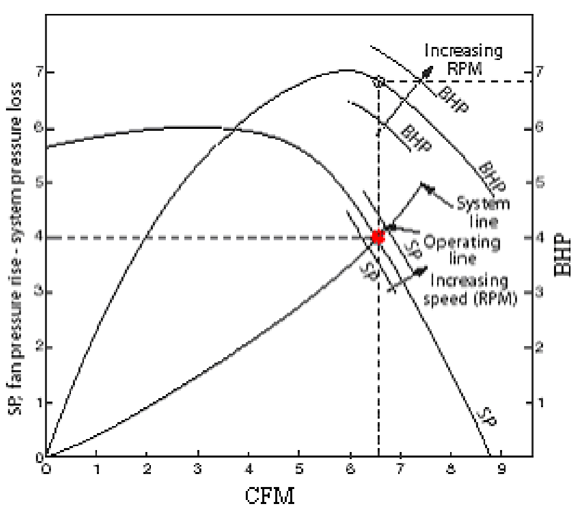
$$BHP_2 = 5.00 \left(\frac{1250}{1000}\right)^3 = 9.77\ BHP$$

According to the fan laws, in order to use the original fan, the speed must be increased from 1000 RPM to 1250 RPM; the motor must be changed from a 5 HP to 10 HP.

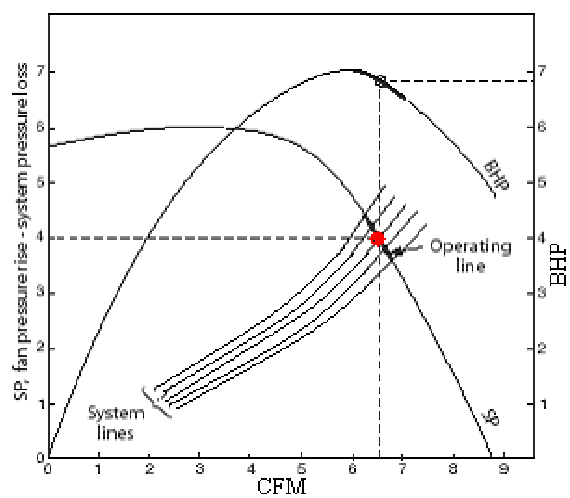
Figure below illustrates fan curves for both the original and new fan performance.



Note that these laws apply only when all flow conditions are similar. Increasing the speed of a fan causes changes in several important parameters, which may invalidate the fan laws. Sometimes a fan system does not operate properly according to the design conditions. The measured airflow in the fan system may be deficient or it may be delivering too much CFM. In either case, it is necessary to either speed the fan up or slow it down to attain design conditions. We can graphically present new SP curves and BHP curves under different control scenarios as shown below:



Volume control by altering fan speed changes the fan curve or the operating lines



Volume control by damper adjustment alters the system lines

The fan control method is the means by which the fan/system operating point is manipulated in order to change the desired flow/pressure somewhere else in the system. If you change the system resistance, the operating point will move along the fan curve. Fans equipped with an outlet damper, opposed bladed inlet box damper or a system damper are examples. If you change the fan performance capability, the operating point will move along the established system resistance curve. By having the ability to adjust both the system and the fan performance, the operating point can be moved to almost any desired position under the fan curve. Controls which move the operating point along the system curve include variable inlet vanes, parallel bladed inlet box dampers, blade angle changes on axial fans and the many techniques used to change fan speed. The speed changes represent an example of fan control that can be accomplished through drive changes or a variable speed motor. It is extremely important to know the individual performance characteristics for each type of fan when working on efficient control strategies and/or energy conservation.

FAN SELECTION

Fan selection is a complex process that starts with a basic knowledge of system operating conditions: air properties (moisture content, temperature, density, contaminant level, etc.), airflow rate, pressure, and system layout. When selecting a fan, the following information must be known.

1) Volume of air to be moved per unit time
2) Static pressure--the estimated system resistance and expected variations
3) Space available for installing fans
4) Amount of noise permitted
5) Efficiency--select fan that will deliver required volume at the expected static pressure with the minimum horsepower
6) Economic considerations

These conditions determine which type of fan—centrifugal or axial—is required to meet service needs. After deciding which fan type is appropriate, the right size must be determined. Fans are usually selected on a "best-fit" basis rather than designed specifically for a particular application. A fan is chosen from a wide range of models based on its ability to meet the anticipated demands of a system. For any one-performance point, there are many different fans, which will satisfy that rating. However, based upon any one set of priorities such as fan size, efficiency, motor size etc., there is only one best fan for that application.

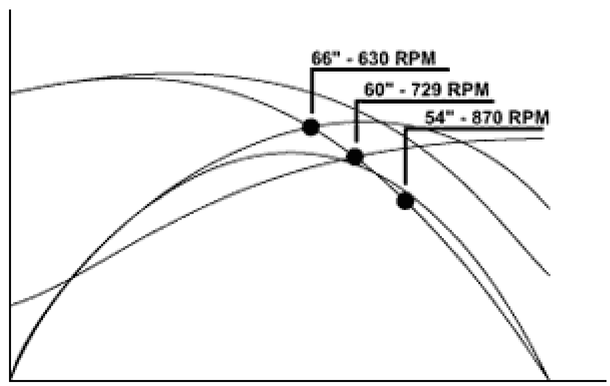
As an example, for a single operating point of 60,000 CFM at 7.0 inches total pressure at a density of .075 lbs/ft^3 (70 degrees), several different fans will satisfy that rating. The possible selections are tabulated in the chart below taking into account impeller size, operating speed, horsepower, relative selling price and recommended motor size.

Influence of fan size and type 60,000 CFM at 7" total pressure							
Item	**Axial**	**Radial**			**Backward**		
Diameter	48"	66"	60"	54"	66"	60"	54"
Speed (rpm)	1750	603	685	816	630	729	870
Power (BHP)	81.4	91.0	95.0	103	77.1	77.1	80.0
Efficiency (%)	81.0	72.5	69.5	64.0	85.5	85.5	82.5
Cost Factor	1.0	2.5	2.1	2.0	2.6	2.2	2.1
Motor (hp)	100	100	125	125	100	100	100

Inferences

Depending upon which priority is chosen with regards to acceptability, the optimum fan selection will change.

1) Based upon selling price the 48 inch axial fan would be selected. This fan is the cheapest of all, but the high RPM operation would increase the noise levels.
2) Based upon operating costs, the 48-inch axial or the 60 inch backwardly inclined fan could be selected.
3) If there was dust in the airstream, the radial tipped fan might be selected taking into account erosion. Plus these fans operate at low RPM; therefore the noise levels would be low.
4) The optimum fan to satisfy the future rating would be the 48-inch axial utilizing a blade angle change since the centrifugals would have to be increased in speed sixteen percent with a corresponding horsepower increase of fifty-eight percent. The motor for the 48-inch axial would simply require better initial insulation so as to handle the increased power.
5) Fan selection and rating point location: A very important concept is the relationship between size and the rating point location on the fan curve. The generic performance curve for the three backwardly curved fans from the chart is shown below. This figure illustrates that the relative position of the rating point on the fan curve changes with a change in fan size for a particular rating. As can be seen, the smaller the fan size used to satisfy a rating, the faster it must run and the farther it will be to the right on the fan curve. In general, it will also be less efficient. It will be louder and wear out faster. This concept is important when considering types of control and fan stability when operation is near peak pressure.



Rating Point Location

Fan selection is based on the static pressure for a given volume of air that needs to be moved. The example above provides clues for fan selection at fixed CFM and pressure. In actual practice the system pressure requirements are never exactly known. Take another example, for three different fans that are capable of supplying about 10,000 CFM (cubic feet per minute) of air.

Fan Rating Table for three fans supplying about 10000 CFM			
Item	**Fan #1**	**Fan # 2**	**Fan # 3**
Diameter	30”	36”	54”
Free air CFM	10200	11700	29100
1/8” SP	9200	10220	22300

Fan Rating Table for three fans supplying about 10000 CFM			
Item	**Fan #1**	**Fan # 2**	**Fan # 3**
¼” SP	7400	8690	14100
3/8” SP	4300	7560	10400
Fan RPM	640	650	385
Motor (hp)	3/4	1	2

Inferences

1) Fan # 1 delivers 10,200 CFM at conditions of free air. Free air means there is no static pressure. However, at 1/8” SP, the air flow rate drops to 9200 CFM and finally to 4300 CFM at 3/8” SP.
2) Fan # 2 supplies 10,200 CFM at 1/8” SP and drops to 7560 CFM at 1/8” SP.
3) Fan # 3 delivers much more than 10,000 CFM at all static pressures below 3/8” SP. Only at 3/8” SP does fan # 3 deliver 10,400 CFM.

For a desired airflow rate of 10000 CFM and the expected SP up to 1/8”, fan #2 is definitely the choice as it operates close to the best efficiency point. At 1/8” SP the fan #1 would not deliver enough air and fan #3 would deliver too much air.

Fan System Design Considerations

It is important to address the following questions related to system design:

1) What is the application and what is the fan supposed to do?
2) How many systems are there and are they interdependent?
3) How many fans per system?
4) Where is the fan equipment located (inside, outside, next to an office, on the ground or several stories up in a building?) If outside, what are the ambient conditions?
5) What space limitations exist? Is there adequate space for maintenance and removal of parts?
6) What facility limitations exist in the form of weight, electrical capability, noise or vibration?
7) What fan orientation is best suited for the application?
8) What fan arrangement is best suited for the application?
9) What are the airstream characteristics? Is fan required to handle moisture, particulate or contaminants?
10) Is the environment corrosive, flammable or hazardous?
11) Are there any leakage requirements for the fan or ductwork?
12) Are there any sound limitations including casing radiated or duct breakout noise?
13) Are there any storage requirements? If so, how long and under what conditions?

14) The cost of electricity and any support functions required?

Life cycle considerations

The overall effectiveness of the fan/system selection should be evaluated taking each of the following into account:

1) Initial cost – What is the purchase price and cost of installation?
2) Operating cost – What is the total cost per year to operate the fan, accessories and any support equipment?
3) Maintenance – Is maintenance costly and frequent? And, is the equipment accessible for ease of maintenance?
4) Frequency of repair and downtime – What is the reliability of the equipment and the cost of downtime?
5) Spare parts – Are spare parts expensive and readily available? Must an inventory of spare parts be maintained?
6) System availability – What percentage of time must the system be operable? There may be the requirement to have a "stand-by fan".
7) Expected life – What is the expected life of the equipment before it is to be replaced?

FACTORS AFFECTING FAN PERFORMANCE

Air Density (Temperature and Altitude)

Air density is a variable of elevation and temperature, and both these variables affect fan air performance. Air density will affect the total pressure that a fan can generate and the horsepower required to move the air. Most fan performance is published at a density based on air at 70°F and at sea level. This is referred to as "standard air", and is defined as clean, dry air with a density of .075 pounds per cubic foot, with the barometric pressure at sea level of 29.92 inches of mercury and a temperature of 70°F.

A fan operating at a higher elevation or temperature will move the <u>same volume</u> of air as it will at standard conditions; however it will generate <u>less total pressure</u> and will require <u>less horsepower</u>.

Note that the volume of air is not affected by variations in air density. In other words, if a fan will move 3,000cfm at 70°F it will also move 3,000cfm at 250°F. Since 250°F air weighs only 34% of 70°F air, the fan will require less BHP but also create less pressure than specified. Therefore, when selecting a fan to operate at a non-standard density using standard air density tables and curves, corrections must be made to the parameters affected by air density. These parameters are static pressure and brake horsepower.

The following formulas and table give air density correction factors for non-standard temperatures and elevations.

$$DCF = \left(\frac{T + 460}{530}\right) \times 1.037^{\frac{E}{1000}}$$

DCF = Density Correction Factor

T = Temperature (degrees F)

E = Elevation above sea level (feet)

$$\text{Air Density } (lb/ft^3) = \frac{0.075}{DCF}$$

Air Temp. °F	Elevation (Feet Above Sea Level)										
	0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000
0	0.87	0.90	0.984	0.97	1.01	1.05	1.08	1.13	1.17	1.22	1.26
50	0.96	1.00	1.04	1.08	1.11	1.15	1.20	1.24	1.30	1.34	1.40
70	1.00	1.04	1.08	1.12	1.16	1.22	1.25	1.30	1.35	1.40	1.45
100	1.06	1.10	1.14	1.18	1.22	1.27	1.32	1.37	1.42	1.48	1.54
150	1.15	1.19	1.24	1.30	1.33	1.38	1.44	1.49	1.55	1.61	1.67
200	1.25	1.29	1.34	1.40	1.44	1.50	1.56	1.61	1.68	1.75	1.81
250	1.34	1.39	1.44	1.50	1.55	1.61	1.67	1.74	1.80	1.88	1.95
300	1.43	1.49	1.54	1.60	1.66	1.72	1.79	1.86	1.93	2.01	2.08
350	1.53	1.58	1.64	1.71	1.77	1.84	1.91	1.988	2.06	2.14	2.22
400	1.62	1.68	1.75	1.81	1.88	1.94	2.03	2.09	2.19	2.27	2.37

How to use this table:

Example: 10,000 CFM @ 2½" SP @ 150°F at an altitude of 7000 feet. In this example the factor will be 1.49 from table above.

1) STEP 1: Multiply the static pressure by the factor (2.5 x 1.49 = 3.725"wg SP say select~4"wg).
2) STEP 2: Select a fan from the following fan charts for the new condition of 10,000 CFM @ 4"wg SP. For example say the appropriate selection from manufacturers catalogues narrows down to a fan delivering 10,350 CFM at 4" SP when operating at 1217 RPM. The BHP rating at this operating point is 9.93 BHP.
3) STEP 3: Correct the horsepower and static pressure in Step 2 to non-standard performance by dividing the factor. (1) 4"wg SP ÷ 1.49 = 2.68" SP (2) 9.93 BHP ÷ 1.49 = 6.66 BHP
4) STEP 4: Check for maximum safe speed from table below. At 150°F, the safe speed factor is .98. The maximum safe speed for the Class I fan is 1355 RPM x .98=1328 RPM. Our RPM selected above is 1217, which is therefore satisfactory.

Maximum Safe Speed Correction Factor						
TEMP	0	70	100	150	175	200
RPM	1.00	1.00	1.00	0.98	0.945	0.91

Final performance is: 10,000 CFM @ 2.68" SP turning at 1217 RPM using 6.66 BHP operating at 150°F at 7000 ft elevation. Note: Use of special high altitude motor is recommended, if altitude exceeds 3300 feet.

Example#2:

Let's look at another example using a specification for a fan to operate at 400°F at sea level. This example will illustrate that the fan must be selected to handle a much greater static pressure than specified. A 20" centrifugal fan is required to deliver 5,000cfm at 3.0 inches static pressure. Elevation is 0 (sea level). Temperature is 400°F.

Solution

1) Air volume delivered by the fan is not affected by density and remains at 5000 CFM
2) Using the chart, the correction factor for 400°F at sea level is 1.62.

3) Multiply the static pressure (3”) by the altitude/temperature correction factor (1.62) to find the standard air density equivalent static pressure (Corrected static pressure = 3.0 x 1.62 = 4.86”. The fan must be selected for 5” of static pressure.)
4) Select the appropriate fan at 5000 CFM and 5” corrected SP. Enter the performance chart and say the RPM at the operating point is 2056 and the BHP is 6.76. (Note: This now requires a Class II fan. Before the correction was made it would have appeared to be a Class I selection.)

What is the operating BHP at 400°F?

Since the horsepower shown in the manufacturers catalogues refer to standard air density, it must be corrected to reflect actual BHP at the less dense conditions. Therefore, divide the BHP (6.76) by the altitude/temperature correction factor (1.62). The new operating BHP is 4.17 BHP.

Important: We now know the operating BHP but what motor horsepower should be specified for this fan?

If a fan is selected to operate at high temperatures, the motor must be of sufficient horsepower to handle the increased load at any lower operating temperature where the air is denser. Assume the air entering the fan at start up is 70°F; therefore no correction should be made. The starting BHP remains at 6.76 and a 7½ HP motor is required.

Note that the BHP corrections should be used only when the starting and operating temperatures are the same.

Fan System Effect

Imagine a fan selected with great care to provide exactly the performance required in the specifications. Once installed, the air balancer reports that air performance is considerably lower than required. What went wrong?

The answer is probably system effect.

The system effect is the change in system performance that results from the interaction of system components. The Air Movement and Control Association (AMCA) define system effect as “a pressure loss, which recognizes the effect of fan inlet restrictions, fan outlet restrictions, or other conditions influencing fan performance when installed in the system.” Typically, during the design process, the system curve is calculated by adding the losses of each system component (dampers, ducts, baffles, filters, tees, wyes, elbows, grills, louvers, etc). The loss of pressure due to all of these sources, known as the system resistance, is for practical purposes proportional to the square of the velocity at the point of loss. As velocity varies directly as volume, we can say that resistance varies as the square of the volume or for a fixed system, it may be said that the pressure required to pass a given volume of air through the system will vary as the (volume flow rate) ^{2}, therefore, if it is required to double the air flow through a system, the fan must be capable of providing twice the volume flow rate at four times the original pressure! AND EIGHT TIMES THE FAN MOTOR POWER!

The governing equation for pressure loss across any particular component is:

$$\Delta p = C \left(\frac{V}{1097}\right)^2 \rho$$

Where:

Δp = pressure loss in inches of water gage (in- wg)

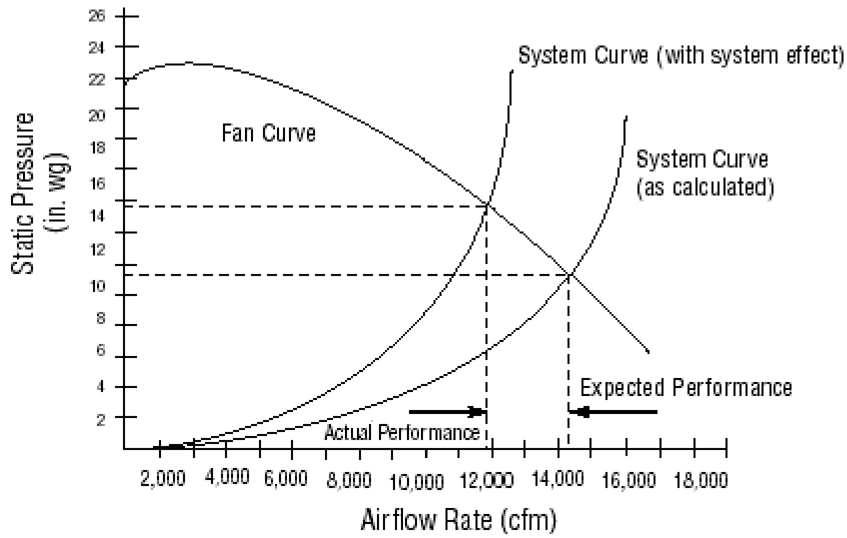
C = loss coefficient for the component

V = velocity in feet per minute

ρ = density of the airstream (0.075 pounds per cubic foot at standard conditions)

The loss coefficient, C, is a dimensionless indicator of flow resistance. The loss coefficient is based on uniform flow into and out of the system components such as ducts, fittings, and components, the values of which are typically listed in tables provided by manufacturers. During the system design phase, designers calculate system resistance curves based on the published loss coefficients for each component. However, system configurations that promote non-uniform flow conditions will create flow resistances that are higher than anticipated, leading to under-performing systems.

The result of this equation is illustrated in system curve below.



What are the penalties of system effect?

The penalty starts with fans selected at higher speeds to compensate for additional losses. Higher speeds result in larger motors, increased cost, reduced efficiencies, increased vibration, and acoustical effects. The severity depends on how inadequate the fan to system connection is. For example, if 20% more air is required, the fan speed will need to be increased 20%. The resulting static pressure will be 44% higher and the BHP will be 73% higher than the original values. New fan drives and a larger motor may be required.

Direct drive fans present a greater problem than belt drive fans. In some cases, it's not possible to use the existing fan unless the installation can be modified to eliminate or reduce the causes of system effect.

How to minimize system effects?

To solve deficient fan system performance problems, it helps to have a clear understanding of fan and system curves plus knowledge of how to apply the fan laws. To minimize system effects, air must enter or leave a fan uniformly. However, in many cases space constraints or other factors prohibit designers to allow for ideal conditions. The following conditions cover the most common causes of system effect.

Inlet Conditions

1) Elbows too close to fan inlet
2) Abrupt duct transition
3) Inlet spin due to duct design
4) Dampers not fully open
5) Damper locations

6) Poorly designed guards
7) Inlet too close to walls or bulkhead
8) Inlet boxes

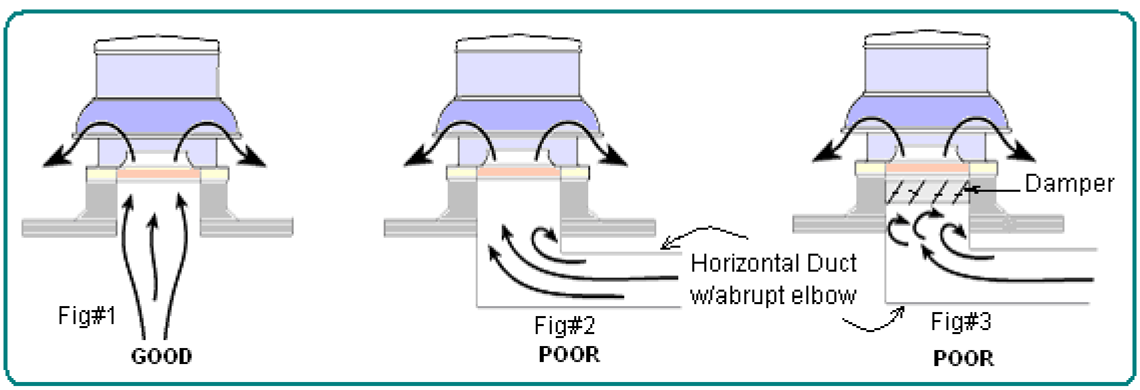
Outlet conditions

1) Elbows too close to fan outlet
2) Abrupt transitions
3) Free discharge
4) Damper location
5) Weather hoods
6) Discharge guards
7) Discharge too close to wall or bulkhead

The system effect can be particularly problematic when the airflow into or out of a fan is disrupted into a highly non-uniform pattern. Poor configuration of ductwork leading to or from a fan can severely interfere with a fan's ability to efficiently impart energy to an airstream. For example, placing an elbow close to the fan outlet can create a system effect that decreases the delivered flow by up to 30 percent. This can require an increase in fan speed, which in turn results in an increase in power and a decrease in system efficiency. The figures below illustrate the concept.

Roof Exhaust Fans

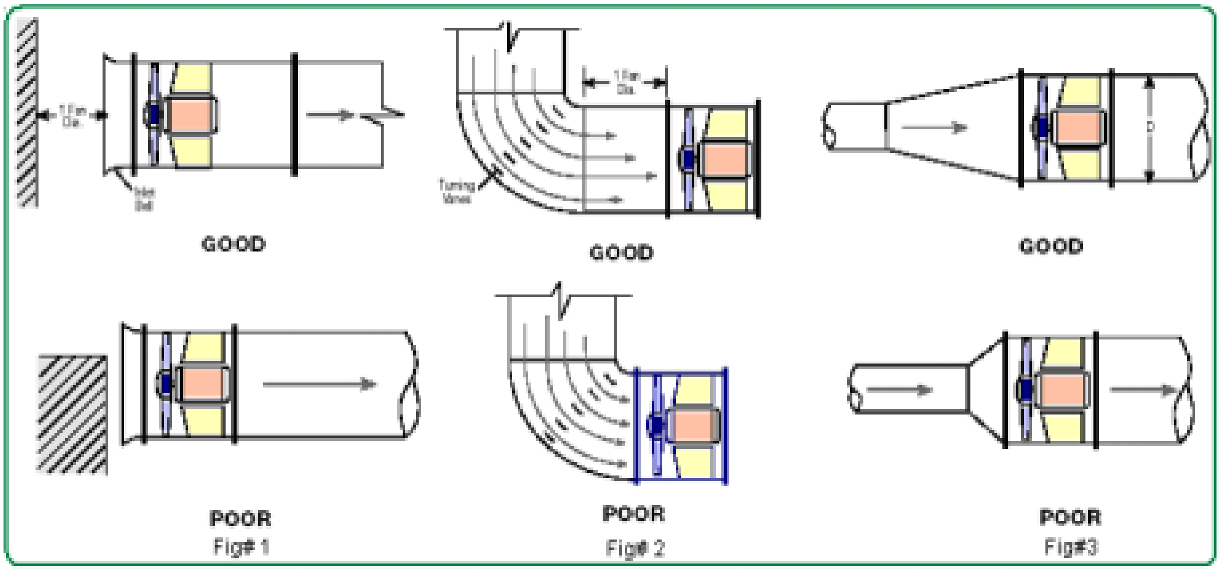
Figures below illustrate how roof exhausters are tested; AMCA refers to Fig#1 setup as "Type A: Free inlet, free outlet". Fig # 2 & 3 illustrate roof exhaust fan installations having system effects. Fig # 3 illustrates the worst case, because the damper is located in a turbulent airstream. To improve on installations where horizontal ducts are used directly under the roofline, turning vanes should be installed in the elbows. In addition, a higher curb or extended base should be used. Higher curbs result in the elbow being further from the damper and fan inlet.



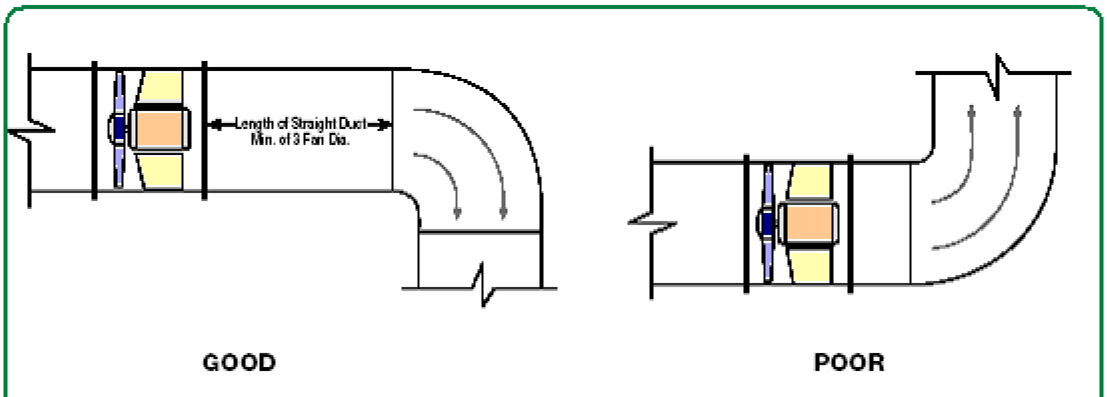
Tube Axial Fans

Figures below illustrate how tube axial fans are tested. Fig #1 is typical of how tube axial fans are tested. AMCA refers to this set-up as "Type B: Free inlet, ducted outlet." Fig # 2 illustrates a good/poor installation with an elbow directly at the fan inlet. The air entering the fan is forced to one side.

Fig # 3 illustrates good/poor installation with ducted fan inlet.



Installations with straight inlet ducts and inlet bells would result in similar performance. Without a discharge duct, a system effect will occur. Inline installations are subject to system effect both at the fan inlet and outlet. Figure below illustrates a good/poor installation with discharge elbow directly at the fan outlet. Inline fans require the appropriate length of discharge duct in order to achieve cataloged performance.



Ducted Outlet Conditions

Centrifugal Fans

Centrifugal fan installations are subject to the greatest possibilities of system effect due to the possibilities of ducted inlets and outlets, plus multiple available arrangements, discharge positions, and clockwise and counter-clockwise rotations. Fig # 1 illustrates how housed centrifugal fans are tested. Fig # 1 is typical of how centrifugal fans are tested. AMCA refers to this set up as "Type B: free inlet, ducted outlet." Fig # 2 illustrates a poor installation with an elbow directly at the fan discharge. This type of installation can be avoided by selecting a fan with the correct rotation and discharge position as shown in Fig # 3. Fig # 4 illustrates a poor installation with an abrupt discharge into a plenum.

Figure 4 illustrates another poor installation with an abrupt discharge into a plenum. A system effect results if a given length of discharge duct is not present.

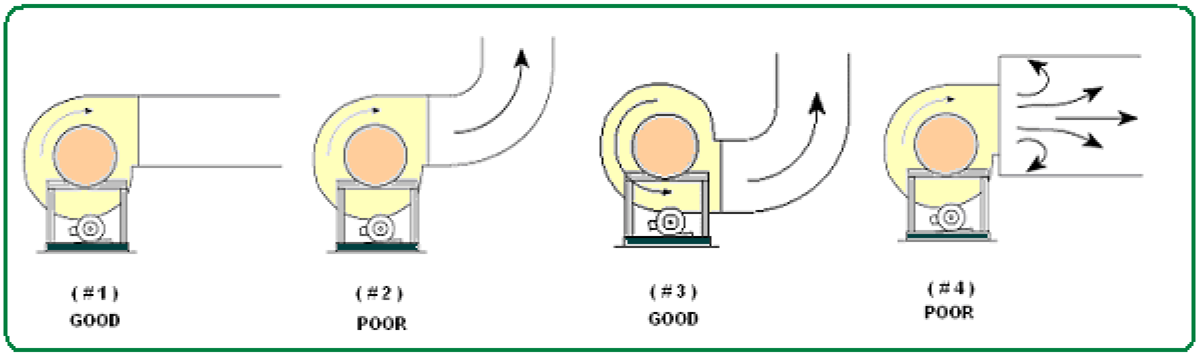
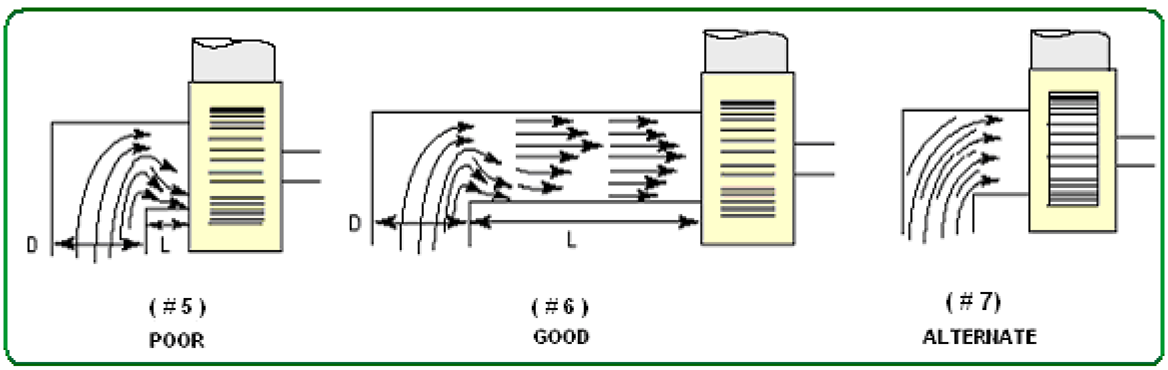
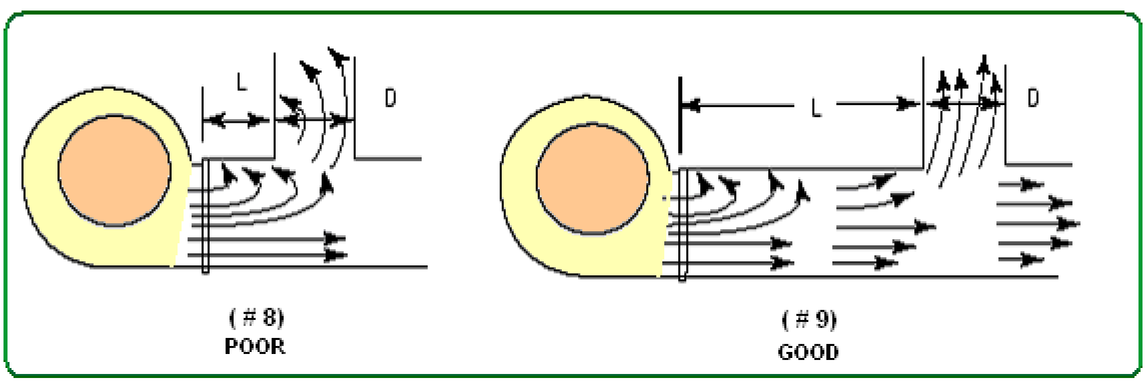


Fig # 5 illustrates installation with improper inlet conditions with an elbow directly at the fan inlet. Fig # 5 could be improved by ensuring L > 3D (refer Fig # 6). If this is not possible, the fan should be equipped with a factory inlet box or alternately, a flow straightener should be considered. Fig # 7 shows round baffles incorporated in the duct as an alternate measure.



Fan Outlet Conditions



The illustrations above show only a few of the many installation possibilities that can cause system effect. It is important to realize that fan manufacturers can only guarantee the fan to perform as tested. In the figures shown, not one of the fans is tested with obstructions, such as elbows, guards or dampers, directly at the fan inlet or outlet. These obstructions cause additional losses that are not included in the fan manufacturer's tests, and in many cases, are not included in the designers' usual system resistance calculations. AMCA Publication 210 shows four basic installation types. However, combining all the fan types, fan arrangements, and manufacturer's choice of how to test, the installation possibilities are far too numerous to cover in this course.

In summary, here are a few points to consider:

1) At the design stage, don't try to save a few dollars per square foot of space. The cost of the resulting poor installation could be much greater.
2) Carefully design the system so it can operate as intended. Personnel doing the installation and checkout should also be familiar with causes of system effect.
3) In correcting installations with system effect, changing the ductwork should not be the last consideration. Remember the penalties of system effect will remain for the life of the project.
4) Use a straight duct length on the discharge of both inline and centrifugal fans. In order to achieve a uniform velocity profile, a 100% effective duct length must be used. To calculate the 100% effective length, use 2.5 duct diameters for 2500 FPM (or less). Add one duct diameter for each additional 1000 FPM.
5) In many cases an installation will end up having an obstruction at the inlet or outlet (or both) causing system effect. If these situations cannot be avoided at the design stage, the system effect should be estimated and added to the calculated system resistance.
6) In general
   - Roof exhaust fans are affected by the inlet condition.
   - Roof supply fans are affected by outlet conditions.
   - Fan types typically affected by both inlet and outlet conditions are inline fans (both axial and centrifugal) and housed single inlet centrifugal fans.

FAN CAPACITY CONTROLS

Most fans are sized to handle the largest expected peak design condition. Because normal operating conditions are often well below these design conditions, air-moving equipment is often oversized, operating below its most efficient point and creating several types of problems. Among these problems are high energy costs, high system pressures and flow noise, and, in systems with high particulate contents, erosion of impeller and casing surfaces. Consequently, the combination of extended operating times and the tendency to oversize the air-moving equipment creates a need for efficient flow control. To accommodate demand changes, the volume of air is adjusted by four principle methods:

1) **Outlet dampers**

   Outlet or discharge air dampers are installed to add resistance at the fan. . As dampers close, they reduce the amount of flow and increase pressure on their upstream side. By increasing system resistance, dampers force fans to operate against higher backpressure, which reduces their output. As a fan works against higher backpressure, its operating point shifts to the left along its performance curve. Fans operating away from their best efficiency points suffer increased operating and maintenance costs. Sizing of discharge air dampers should be done with great care. There are many rules-of-thumb, but the recommended procedure is to size the discharge air dampers for a wide open pressure drop of from 7 to 10 percent of the system pressure.

   Advantages: Damper control is a simple and low-cost means of controlling airflow

   Disadvantages: The penalty is high resistance and increased fan horsepower.

---

2) **Inlet vane control**

   Inlet vanes often referred to as pre-rotation vanes, cause the air to swirl before it encounters the fan wheel. The fan wheel cannot "grip" the air as well and consequently, capacity is reduced more efficiently than with discharge damper control. Excess pressure is not created and wasted. The fan inlet vanes are positioned by an actuator in response to a signal received from the system static pressure receiver-controller.

These pre-rotating swirls lessen the angle of attack between the incoming air and the fan blades, which lowers the load on the fan and reduces fan pressure and airflow. By changing the severity of the inlet swirl, inlet vanes essentially change the fan curve. Because they can reduce both delivered airflow and fan load, inlet vanes can improve fan efficiency. Inlet vanes are particularly cost effective when the airflow demand varies between 80 and 100 percent of full flow; however, at lower airflow rates, inlet vanes become less efficient.

Advantages: Because variable-pitch fans maintain their normal operating speed, they avoid resonance problems that can be problematic for certain fan types. Additionally, variable-pitch blades can operate from a no-flow to a full-flow condition without stall problems. During start-up, the fan blades can be shifted to a low angle of attack, reducing the torque required to accelerate the fan to normal operating speed.

Disadvantages: Disadvantages of this flow-control option include potential fouling problems because of contaminant accumulation in the mechanical actuator that controls the blades. Also, because motor efficiency and power factor degrade significantly at loads below 50 percent of rated capacity, operating at low loads for long periods may not provide efficiency advantages and can incur a low power factor charge from the utility.

---

3) Variable Pitch Blades

Variable pitch axial-flow fans deliver an amount of air in accordance with the pitch of the fan blades. As more or less air is needed in the system, an actuator positions the pitch of the fan blades accordingly. The positioning of the blades is similar to the positioning of the inlet vanes. However, the fan is always spinning while inlet vanes remain stationary. The degree to which the blades are pitched determines how much air can be “gripped” and passed on into the system Variable-pitch fans allow the fan blades to tilt, changing the angle of attack between the incoming airflow and the blade.

---

4) Fan speed control

Changing the rotational speed is the most efficient. The laws governing fan operation provide a useful clue that power varies as cube root of RPM or airflow. When fan speed decreases, the curves for fan performance and brake horsepower move toward the origin. Fan efficiency shifts to the left, providing an essential cost advantage during periods of low system demand*. Reducing fan speed can significantly reduce energy consumption. For example, according to the fan laws, reducing fan rotational speed by 20 percent decreases fan power by 50 percent.*

If the volume requirement is constant, it can be achieved by selecting appropriate pulley sizes. If the volume varies with the process, multiple-speed fans or variable-speed drives (VSDs) can be used. VSDs allow fan rotational speed adjustments over a continuous range, avoiding the need to jump from speed to speed as required by multiple-speed fans.

Many types of ASDs are available, including mechanical (eddy current drives, variable-ratio pulley, and hydraulic drives), direct current (DC motors), and electronic. Although mechanical drives and DC motors have been applied extensively in industrial settings, they are seldom used in commercial buildings for economic or technical reasons. The mechanical variable-ratio pulley is applicable to commercial buildings (from 5 to 125 horsepower), but space requirements and mechanical problems usually make commercial applications impractical. DC motors comprise a mature technology, but they are expensive and have a reputation for high maintenance costs. The electronic load-commutated inverter has also been used in industry, but it is not an energy-conscious choice for commercial buildings. Frequency operated adjustable speed drives commonly known as variable frequency drives (VFD’s) are most commonly used ASD, largely because of their proven effectiveness in reducing energy costs.

Advantages of VFDs: Among the primary reasons for selecting VFDs are improved flow control, ability to retrofit to existing motors, their compact space advantages, and elimination of the fouling problems associated with mechanical control devices. The benefits include:

- VFDs decrease energy losses and offer substantial savings with respect to the cost-per-unit volume of air moved.
- VFDs eliminate the reliance on mechanical components, providing an attractive operational advantage, especially in “dirty” airstreams.
- VFDs decrease airflow noise during low system demand, they can improve worker comfort.FDs offer operating improvements by allowing higher fan operating efficiency and by increasing system efficiency as well.
- VFDs provide soft-start and allow the motor to be started with a lower start-up current (usually about 1.5 times the normal operating current in contrast to 5 to 6 times higher than normal operating currents in case of normal motors). This reduces wear on the motor windings and the controller. Soft starting a fan motor also provides benefits to the electrical distribution system reduces voltage sags and wear on the motor windings.

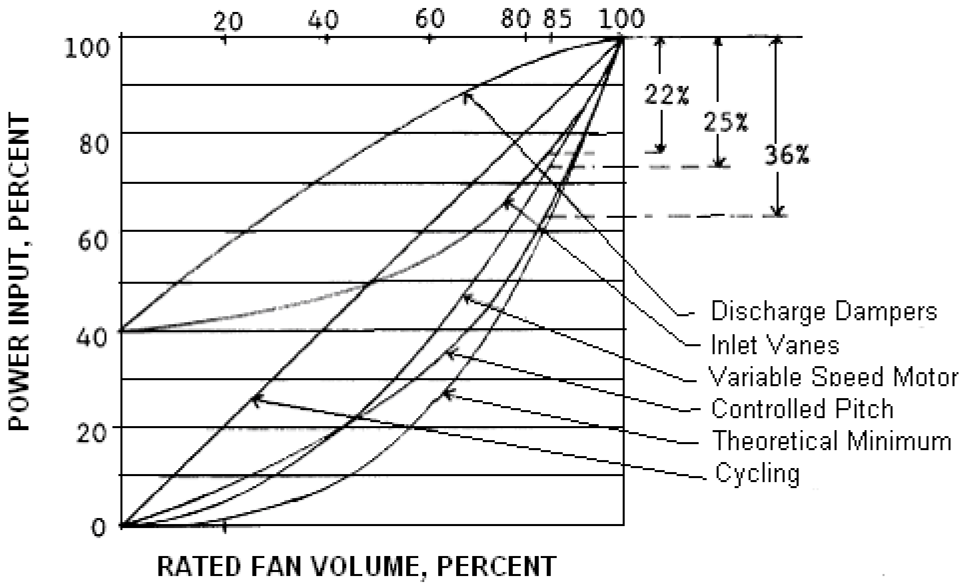
Disadvantages of VFDs: VFDs are not appropriate for all applications.

- Decreasing the rotational speed of a fan too much often risks unstable operation, especially with axial fans and some centrifugal fans, such as backward inclined airfoil and forward-curved types. With these fans, careful review of the performance curves should precede the selection of a VFD.
- Fans, like most rotating machinery, are susceptible to resonance problems. Resonance is an operating condition in which the natural frequency of some component coincides with the frequency set up by the rotation. Fans are usually designed so that their normal operating speeds are not near one of these resonant speeds. However, decreasing the rotational speed of a fan increases the chances of hitting a resonant speed. The effects of operating at resonant speeds can be damaging. Shafts, bearings, and foundations are particularly susceptible to problems with resonance.
- When a fan’s rotational speed is reduced, the fan generates less pressure, and some fans, like many types of turbo-machinery, operate poorly against shut-off conditions. If a VFD slows the fan, the static pressure requirement may exceed the pressure generated by the fan and no airflow will be generated.
- In some VFD applications, power quality can also be a concern. VFDs operate by varying the frequency of the electric power supplied to the motor. The solid-state switching that accompanies inverter operation can create voltage spikes that increase motor winding temperatures, accelerating the rate of insulation degradation. To account for the added winding heat, conventional motors usually must be de-rated by 5 to 10 percent when used with VFDs. A classification of motors known as “inverter-duty” has been developed to improve the matching of VFDs to motors.
- VFDs can also generate electrical noise that interferes with the power quality of the supporting electrical supply. These problems are typically correctable with the installation of coils or electrical filters.

Energy Comparison

Figure below shows the approximate power savings that can be obtained by reducing air quantities for the four methods of capacity control.

From power consumption standpoint, variable speed motors and blade pitch control are the most efficient. Inlet vanes save some power, while discharge dampers throttling at the fan save little. From a first-cost standpoint, dampers are the least costly. Inlet vanes and blade pitch control follow, with variable speed motors being the most expensive.



Example of energy saving

The relative efficiencies of the flow control options is illustrated below:

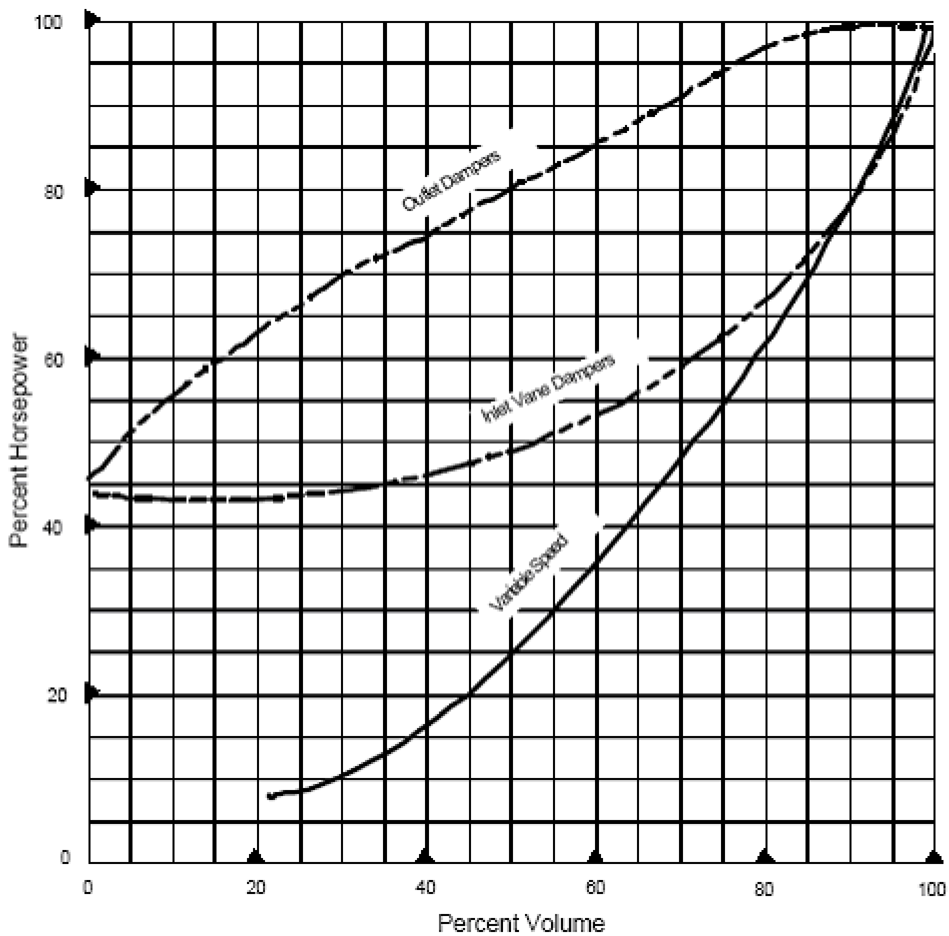


Figure above shows a horsepower comparison of various methods of fan control for typical fans. A 50 percent reduction with an outlet damper requires 80 percent of rated power; with a slower-speed motor, only 25 percent of rated power is required. (Refer to the speed control curve on the figure.) Therefore:

Annual Savings = (20 hp x 80% - 20 hp x 25%) x 6,000 hrs/yr x $0.041/hp-hr

= $2,700

The reduction in fan output will result in operation of the electric motor at less than rated capacity. If the horsepower required at the reduced flow is less than about one third of rated horsepower, the potential savings for substitution of a smaller motor should also be investigated.

FAN NOISE

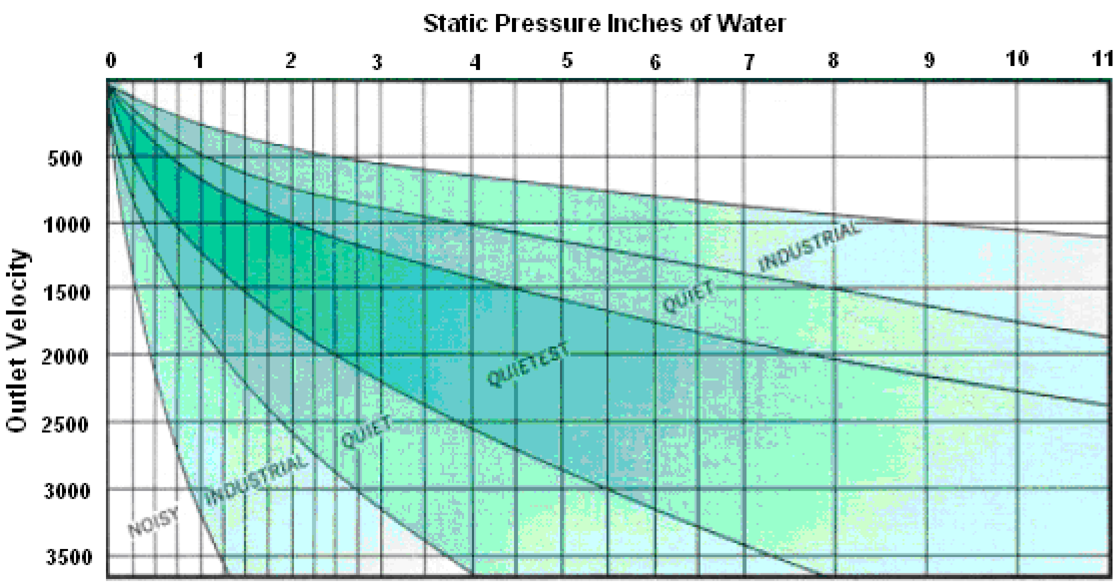
Noise is generally considered low quality, unwanted sound. The primary selection criterion for a fan is based not on its acoustical characteristics but on its ability to move the required amount of air against the required pressure. In addition, the fan must do so at a reasonable initial cost (it also may be required to handle dust-laden air, to resist abrasion and/or corrosion, to have a type of construction that can be repaired easily in the field, to withstand high temperature, etc). Once these requirements have been met, the type, size and speed of the fan are completely determined then the noise characteristics of the fan also are determined. In most cases it is not practical to substitute a fan, which generates less noise, since a quieter design of the same type probably will not meet the other operating specification for the fan. Therefore, sound power levels generated by a correctly selected fan must be accepted as the sound power levels to be used in acoustical design calculations. There are three categories of fan noise viz:

1) Aerodynamic generated noise is characterized by a continuous broadband frequency spectrum with a superimposed tone. The tone is typically objectionable when it becomes 4-6dB louder than the rest of the spectrum. The tone can be the blade frequency, which is a function of the fan type. It can become very objectionable when system effects and various controls cause it to rise higher than normal. Additional causes include turbulence, high velocities, and instabilities due to pulsation and surge. The probable causes of aerodynamically generated noise is blade frequency, system effects, damper stall, surge, hunting, or turbulence, The noise characteristics associated with aerodynamic noise are pure tone, whistles, rumble, pulsating, hiss or beating.

2) Mechanical noise due to rotation and has a different sound quality and characteristic. It has a metallic sound caused by metal-to-metal contact. This contact may be constant or intermittent. The probable causes of mechanical noise loose fittings or problems with bearings, sheaves, shaft or V-belts. The noise characteristics associated with mechanical noise are rattling, scraping, banging, screeching or clicking.

3) Electrically generated noise is a function of motors, relays, controls or unbalanced line voltages into the motor. Sometimes improperly matched VFD and motors can cause a substantial increase in the motor noise due to imperfect sine wave simulation. The noise characteristics associated with electrical noise are hum, whine or chatter.

Broadly, fan noise is a function of fan design, volume flow rate of air, total pressure, and efficiency. After making a decision on the proper type of fan for a given application, the best selection of a specific fan must be based on efficiency. The most efficiently operating fan will also be the quietest fan. Fan noise is an important consideration, particularly when the fan is located near occupied space and is operating against a static pressure of 1” or greater. The following table provides a quick method for considering the degree of quietness when selecting a fan.



There are operating characteristics typical to various generic fan types, which make them more or less suitable to variable volume operation (having smaller or larger range on the fan curve with relative operation efficiency). The shape of each curve gives some indication of sensitivity to the system effects, such as inlet and discharge conditions and potential noise. Each of the fan illustrations in the figure below shows a) system resistance intersecting the fan curve at “maximum efficiency point” (quiet operation), b) system resistance intersecting the fan curve in the region of “rotating stall” (low frequency rumble) and c) system resistance in the “surge” region (low frequency fluctuating noise). On each of these fan curves, moving further out on the fan curve (to the right of the maximum efficiency) also increases noise levels, but predominantly in mid and high frequencies and at blade passage frequency, which varies with the rotational rate according to the following equation:

BPF = (n/60) * N

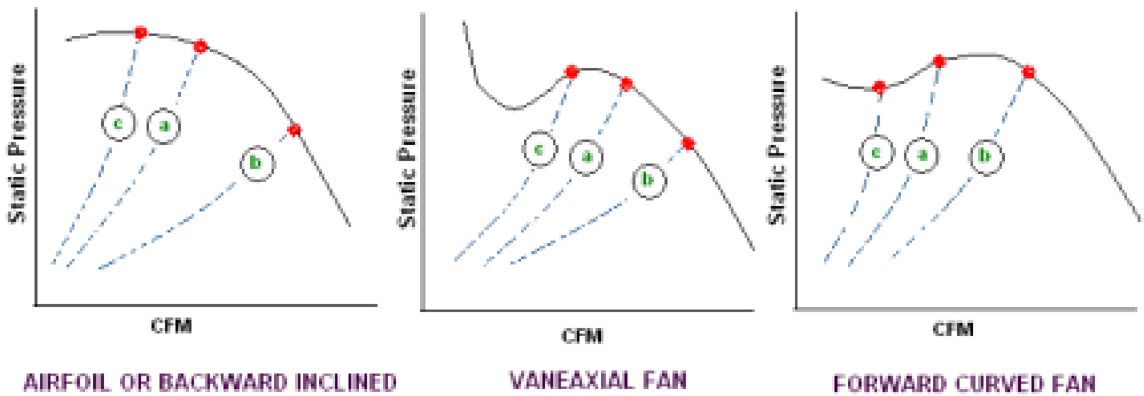
Where

BPF = blade passage frequency

n = fan wheel revolutions per minute (rpm)

N = number of impeller blades

Note that the blade passage frequency can be perceived as tone. If it coincidently occurs within a frequency span of fan noise spectrum that is prominent, it can emphasize the tonality of an otherwise relative smooth fan noise spectrum. Harmonics and sub-harmonics are possible at the multiples of the blade passage frequency, but at lower amplitude (sound level).



Noise Characteristics of Various Types of Fans

The noise characteristics of various types of fans are reasonably predictable, and if the fans are well designed, the noise characteristics are not significantly affected by minor changes in the fan geometry. Low outlet air velocity does not necessarily assure quiet operation. Noise comparisons of different types of fans on the basis of rotational speed alone may be erroneous. The only valid basis for comparison of different fan types is the actual sound power levels generated by the fans when the fans are all producing the required volume flow rate of air at the specified static pressure.

Prediction of Fan Sound Power

The sound power generated by fan performing a given duty is best obtained from manufacturer's test data taken under approved (ASHRAE Standard 68- 1986; also AMCA Standard 300-1986). However, if such data are not readily available, the octave-band sound power levels for various fans can be estimated by the procedure described here.

Fan noise can be rated in terms of the specific sound power level, defined as the sound power level generated by a fan operating at a specific capacity and pressure. The specific capacity chosen is $1 m^3/s$ and the pressure of 1 Pa. By reducing all fan noise data to this common denomination, the specific sound power level serves as a basis for direct comparison for the octave-band levels of various fans.

Sound power levels at actual operating conditions can be estimated by using the corresponding fan volume flow rate and fan pressure given by:

$$L_w = K_w + 10 \log (Q / Q_1) + 20 \log (P / P_1) + D$$

Where

L_w = estimated sound power level of fan
K_w = specific sound power level (see table below)
Q = flow rate, m^3/s
Q1 = 1 m^3/s

P = pressure drop, Pa
P_1 = 1 Pa and
C = correction factor for point of fan operation, dB

(Note - 1in-wg = 250 Pa and 1m^3/s = ~2117CFM)

Values of the estimated sound power level are calculated for all eight bands, and the BFI (Blade Frequency Increment) is added to the sound pressure level in the octaves band in which the blades passage frequency falls.

Specific Sound Power Levels of Typical Fans									
Fan Type	**Octave Band**								
	63	125	500	1K	2k	4K	8K	BK	BFI*
Centrifugal									
AF,BC, or BI									
Wheel diameter									
Over 900 mm	40	40	39	34	30	23	19	17	3
Under 900 mm	45	45	43	39	34	28	24	19	3
Forward Curved									
All wheel diameters	53	53	43	36	36	31	26	21	2
Radial Bladed									
Low pressure									
(1 to 2.5 kPa)	56	47	43	39	37	32	29	26	7
Medium pressure									
(1.5 to 3.7 kPa)	58	54	45	42	38	33	29	26	8
High pressure									
(3.7 to 15 kPa)	61	58	53	48	46	44	41	38	8
Axial Fans Vaneaxial									
Hub ratio 0.3 to 0.4	49	43	43	48	47	45	38	34	6
Hub ratio 0.4 to 0.6	49	43	46	43	41	36	30	28	6
Hub ratio 0.6 to 0.8	53	52	51	51	49	47	43	40	6
Tubeaxial	51	46	47	49	47	46	39	37	7
Over 1000 mm									
Wheel diameter	48	47	49	53	52	51	43	40	7
Under 1000 mm									
Wheel diameter	48	51	58	56	55	52	46	42	5

Propeller General ventilation									

*BFI = Blade frequency increment

Relative Noisiness

A feel of the relative noisiness of different types of fans can be had from table below for a fan-delivering 20000CFM of air against a static pressure of 4in-wg. It is assumed that the wheel diameter is within 900 mm and the hub ratio is small 9 (about 0.3) It may be noted that for this typical case, the centrifugal blower with airfoil blades, or backward curved blades or backward inclined blades, is likely to be the quietest, and the propeller type fan may be the noisiest fan. In the axial types of fan, vaneaxial may be noted to be the quietest as well as most efficient from the aerodynamic point of view.

Comparative Sound Power Levels										
Type of Fan	**Specific sound power level, K_w**									**Total Power Level**
	63	125	250	500	1k	2K	4K	8Khz	Total	Lw dB
Centrifugal AF,BC, or BI	45	45+3*	43	39	34	28	24	19	51.1	121.1
Centrifugal, forward curved	53	53	43	36+2*	36	31	26	21	56.3	126.3
Centrifugal, radial bladed	56	47+7*	43	39	37	32	29	26	58.3	128.3
Vaneaxial	49	43	43+6*	48	47	45	38	34	53.8	123.8
Tubeaxial	48+7*	47	49	53	52	51	43	40	59.2	129.2
Propeller	58+5*	51	58	56	55	52	46	42	62.8	132.8

*Octave Band of Blade Frequency Increment

If the project is a new piece of equipment, select a fan with thick, wide blades and run the fan at lower speeds to reduce the sound generated by the fan. Once a fan type has been selected and major modifications to fan geometry are no longer possible, fan speed is the most significant feature that can be changed to reduce generated sound.

If you have an existing piece of equipment, it is generally useless to experiment with changes in the fan configuration. If the changes reduce the noise level, they also will change the fan performance characteristics so that they no longer meet the specified conditions. If lower sound power levels are required than are generated by well-designed fans, it is necessary to add attenuation to the system. This may be provided by sound attenuators (also called sound traps) installed as separate unit in the field or by sound attenuators installed as an integral part of the fan assembly.

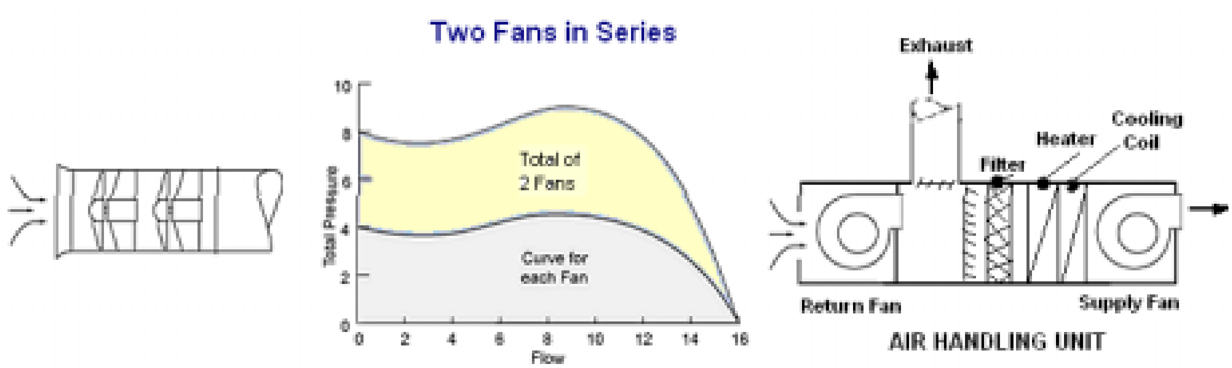
MULTIPLE FAN SYSTEMS- FAN IN SERIES & PARALLEL

In most fan systems a single fan is selected for the required system design rating. Fans often are combined in series or in parallel as an alternative to using single, large fans. Sometimes it is advantageous to use more than one fan in a system but most often there are compelling reasons to use more than one fan in a single system. These include:

1) One fan may be too large and not fit into the desired space, or it may weigh too much if supported on upper levels.
2) The required operating range of the system may necessitate multiple fans instead of one large fan controlled over a wide operating range. Multiple fans for capacity control may be more economical, if cost of operation is critical, especially at very low flow rates for long time intervals.
3) Supply and exhaust fans operating on opposite ends of a system decrease the pressure build-up in a duct or space compared to a single fan. It is usually easier to control a zero point location or maintain low pressures (such as the draft over a fire in a boiler), if supply and exhaust fans are used.
4) Critical systems are often equipped with redundant or back-up fans in case of a fire or some other emergency that requires a sudden increase in flow. Redundant fans are also used to eliminate downtime during fan maintenance.
5) Some systems for process applications may require pressures that are greater than a single fan can produce or when noise may be a special concern. When this occurs, two fans are placed in series with each taking about one-half of the pressure.

Rating Two Fans in Series

Fans configured in series tend to be appropriate for systems that have long ducts or large pressure drops across system components. Fans used in an induced-draft/forced-draft configuration can minimize the amount of pressurization in a duct or an enclosure.



There may also be by-pass ductwork around the second fan if only one fan is run for a period of time.

Advantages of fans in series include:

1) Lower average duct pressure
2) Lower noise generation
3) Lower structural and electrical support requirements.

Considerations for Putting Fans in Series

1) To simplify selection and control, two fans of the same size are typically used with the required flow rate defined by the inlet conditions of the first fan.

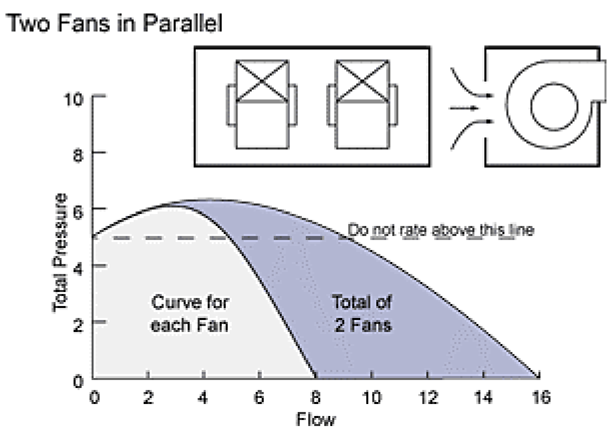
2) Establish the system requirements in terms of total pressure. If they are known only in terms of static pressure, total pressure values may be calculated by adding the velocity pressure corresponding to the velocity passing through the outlet of the second stage fan to the system static pressure requirements. The combined total pressure across both fans will be the sum of the individual total pressure of each fan. Total pressures are used instead of static pressure because the fans can actually be different sizes and a change in fan or connecting duct areas has an influence upon static pressure values.
3) If axial fans or inline fans are being considered, select each fan for the flow rate required and one-half of the system total pressure requirements.
4) If centrifugal fans are being considered, select each fan for the flow rate required and one-half of the system total pressure requirements plus an allowance for interconnecting ductwork losses, typically one inch of total pressure.

It must be realized that the above selection process is approximate in that the actual individual performance of each fan is not the same. *Both fans will handle the same mass flow of air but not volumetric flow rate.* This is the result of differences in the inlet densities of each fan caused by differences in the inlet absolute pressures and differences in the temperatures resulting from the possible heat of compression, process heating or cooling through AHU coil or motor heating etc. by the first stage fan. The greatest significance is that the rating process can be simplified by making sure the system requirements are in terms of total pressure and that the fans are selected using total pressure.

Rating Fans in Parallel

Parallel configurations may be feasible for systems with wide variations in air-moving requirements, which preclude a single fan from consistently operating close to its best efficiency point (BEP). Operating a fan away from its BEP can result in higher operating and maintenance costs. Multiple fans placed in parallel allow units to be energized incrementally to meet the demands of the system. In most instances fans in parallel will be in some form of plenum application. Unlike fans in series when typically only two fans are involved, parallel fan applications may use multiple fans and energizing or de-energizing individual fans can operate each fan operated more efficiently. In this case the selection process is straightforward in that each fan will be selected for the same static or total pressure with the flow rate being the total flow divided by the number of fans. Use care when selecting fans in parallel to ensure that the system resistance remains on a stable portion of the fan curve at all times. This is particularly true when the fans have a pronounced surge area or a dip in the fan curve and some form of control is applied.

A good rule of thumb is to ensure that the operating point with all fans running is no higher than the lowest pressure in the dip. This minimizes the possibility that the fan will hunt back and forth across the peak of the curve looking for an operating point. This policy also minimizes the likelihood that the fans will experience unequal loading causing differences in motor load or creating unequal velocity profiles within the plenum, which may result in a system effect. To allow operation of individual fans in a multiple fan arrangement, each fan must have a back-draft damper installed to prevent recirculation through the fan when it is idle.



Fans placed in parallel can provide several advantages including:

1) High efficiencies across wide variations in system demand

2) Redundancy to mitigate the risk of downtime because of failure or unexpected maintenance.

Parallel fan configurations may also be a safety requirement in case of a single fan failure. In mining and other hazardous work environments, ventilation is critical to worker safety. The existence of backup fans can help avoid production stoppages and may be a safety requirement.

Fans placed in parallel can provide several disadvantages including:

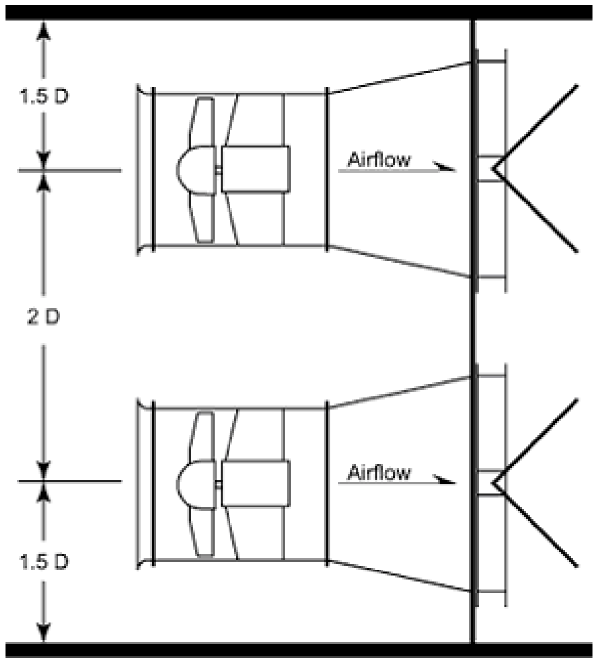
1) When placing centrifugal fans in parallel, caution should be used to ensure that one fan does not dominate another. Ideally, all fans should be the same type and size; however, differences in the duct configuration can cause one fan to operate against a higher backpressure. In severe cases, one fan will force another fan to operate far away from its BEP. Often, fans placed in parallel are the same model so that there is balanced load sharing during periods when all the fans are operating.

2) Another problem that accompanies parallel operation of fans is instability. This problem is especially applicable to fans with unstable operating regions (axial fans, forward-curved centrifugal fans, and airfoil fans). Instability results from alternate load sharing that can occur below certain airflow rates. This can occur despite the fact that each fan alone is operating outside of its stall region.

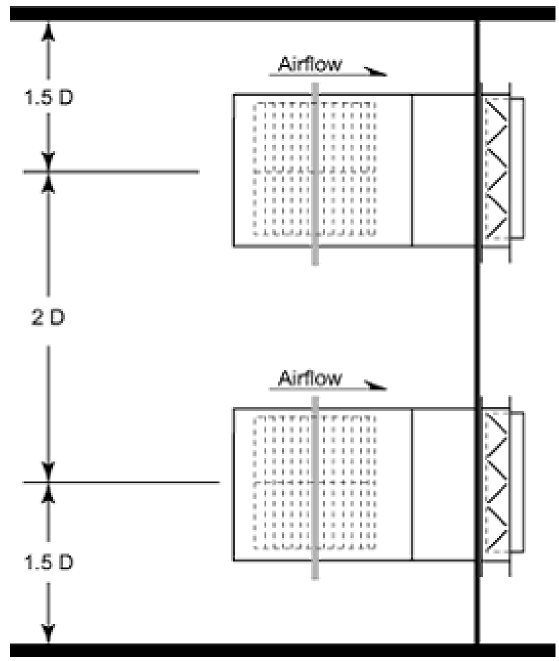
3) The combined performance curve of both fans has a region in which there are multiple combinations of airflow from each fan that can meet the system needs. The instability results from the fans' shifting between these multiple combinations (known as "hunting"), as the fans tend to load and unload. In addition to creating an annoying noise pattern, this continued hunting increases the wear on the fan drives because of repeated acceleration and deceleration. To avoid this problem, the system airflow should be kept to the right.

4) Fans in parallel should have some form of isolation damper to prevent the air from an energized fan from going back through a fan that is not energized. The damper also serves to minimize the shock during start-up of bringing a windmilling fan to a stop and then up to speed again. This is not good for the fan, motor or system. A mechanical backstop clutch can also be used to eliminate wind milling of fans installed in parallel.

5) The type of isolation damper used will vary with the type of fan. Back draft or opposed-blade control dampers are used at the discharge on double width centrifugal fans. Butterfly dampers are commonly used at the discharge of tubular inline fans (axial, centrifugal, and mixed flow). Isolation dampers for plenum fans should be located farther away from the fan, either up stream or down, in order to minimize the loss through the dampers.



Vaneaxial Fans in Parallel



Double Width Centrifugal Fans in Parallel

One Fan Operation in Multiple Fan Arrangement

All equipment will need periodic maintenance and repair. This means that at least one fan is shut down while the others are running. For fans in parallel equipped with isolation dampers, this is generally no problem. The motors of the fans left running must be sized properly by taking into account the shape and slope of the horsepower curve further out to the right. With fewer fans running, the system line will intersect the fan curve further out to the right than with all the fans running. As an example, with a vane axial fan the power may drop off but with a forward curved fan the power will increase due to the constantly rising horsepower characteristic.

For close-coupled fans in series, it is not advisable to run one fan with the other off. Obviously, an isolation damper will stop the air entirely. Without an isolation damper, the fan shut down will windmill, but the pressure drop across it when added to the reduction in pressure having only one fan running will likely make the system useless.

Advantages of Multiple-Fan Arrangements

1) **Lower Average Duct Pressure:** The series-configurations fans along different points in a system minimize the average static pressure in a duct. Because leakage in a duct system depends largely on the pressure difference between inside and outside the system, reducing the maximum system pressure can minimize energy losses attributable to system leaks.

2) **Lower Noise Generation:** Lower pressure requirements can decrease the noise generated by fan operation.

3) **Redundancy:** Failure of one unit does not force a system shutdown. In a single-fan application, a repair task on that fan requires a system shutdown. With a multiple-fan arrangement, one can be repaired while the others serve the system. In some facilities, fan failure can cause the interruption of production work. With redundant fan configurations, failure of one fan does not necessarily cause the whole process to halt. Although total fan output falls if one of the parallel units fails, the capacity of the remaining fan or fans may be sufficient for limited production.

4) **Efficiency:** Allowing each fan to operate close to its BEP can provide substantial energy savings. In addition, a potential advantage of multiple fans is a higher overall efficiency level. Although larger motors tend to be more efficient than smaller ones, operating smaller, higher-speed fans close to their BEPs can often achieve a net efficiency advantage over a single, low-speed fan.

5) **Structural and Electrical Constraints:** Two smaller fans in series may be more suitable in terms of structural and electrical requirements than a single one. Large motors have large starting currents that can affect the power supply to other parts of the facility. This concern is particularly acute if the service requires the fan to energize and de-energize relatively often. Frequent power surges that often accompany the start-up of large motors can create power quality variations that are problematic for numeric controlled machinery and other sensitive equipment.

Also, the use of multiple fans in parallel may be necessary because of space considerations. A single fan with an impeller large enough to move the proper amount of air may not fit into the available space or may encounter structural constraints.

Check Points

When more than one fan is used in a system, whether in series or parallel, the following questions must be answered.

1) What is the relative location of the rating point on each fan curve under all operating conditions? Does the operating point remain on a stable portion of the curve?

2) Does the location of the rating point allow the use of the fan control desired and the range of flow desired?

3) Taking into account the range of operation and numbers of fans, do the motors satisfy the power required?

4) Are there properly designed isolation dampers or other provisions in case some fans are shut down? Are leakage requirements satisfied?

5) Are the system requirements satisfied under all operating conditions?

6) Are sound levels acceptable under all operating conditions?

FAN CONSTRUCTION

Fans are constructed from many different materials, and like everything else, there are pros and cons to any option.

1) Heavy solid-blade fans may experience extended starting times due to the high inertia load that the motor must accelerate to speed. It is not unusual to have electric motor overloads trip from extended starting times when applied on high inertia propeller fans. With increasing market pressures of energy conservation, fans generally are applied with less of a safety factor between motor size and fan requirements.

2) Hollow construction fan blades offer thick airfoils that move large volumes of air, yet the blades are lightweight and relatively stiff.

3) Fiberglass construction provides corrosion resistance for applications in chemically aggressive environments.

4) In today's market, blades are being constructed from aluminum alternatives. For example, many heat exchangers and cooling towers operating near the ocean now use aluminum fans. Several manufacturers offer fan blades constructed of marine-grade aluminum, which provides good corrosion resistance. For more aggressive environments, special coatings can be applied to aluminum blades, offering acceptable fan life at a lower initial cost compared to fiberglass blade construction.

5) Wet cooling towers can have airborne water droplets that are surprisingly abrasive to a fiberglass fan blade's leading edge. Most molded blades have a parting line at the leading edge. This area is especially susceptible to erosion damage due to trimming material that may be caught in the mold's parting line during blade fabrication.

6) Blades constructed with epoxy resins have more natural erosion resistance than blades using vinyl ester or polyester resins, but epoxy resins are less desirable to work with from a manufacturing perspective. Make sure fiberglass fan blades with vinyl ester or polyester resin materials have a barrier material - either external or internal to the fan blade - to protect against erosion at the leading edge.

7) A high operating temperature generally is not a limiting factor for wet cooling towers, but it can be a factor for induced-draft air-cooled heat exchangers and other equipment. Fiberglass fan blades begin to experience reduced mechanical properties at much lower temperatures than aluminum fan blades. A recommended maximum temperature for fiberglass construction might be 180°F (82°C) while aluminum blades can perform satisfactorily at 300°F (149°C). Be sure to consider the maximum temperature the fan can experience when the process is operating but the fan is off.

Fan Construction for Explosive Environments

Fan applications may involve the handling of potentially explosive or flammable particles, fumes or vapors. Such applications require careful consideration of all system components to insure the safe handling of such gas streams. The following should be noted:

1) All parts of the fan in contact with the air or gas being handled shall be made of nonferrous material. Steps must also be taken to assure that the impeller, bearings, and shaft are adequately attached and/or restrained to prevent a lateral or axial shift in these components.

2) The fan shall have a nonferrous impeller and nonferrous ring about the opening through which the shaft passes. Ferrous hubs, shafts, and hardware are allowed provided construction is such that a shift of impeller or shaft will not permit two ferrous parts of the fan to rub or strike. Steps must also be taken to assure the impeller, bearings, and shaft are adequately attached and/or restrained to prevent a lateral or axial shift in these components.

3) The fan shall be so constructed that a shift of the impeller or shaft will not permit two ferrous parts of the fan to rub or strike.

4) No bearings drive components or electrical devices shall be placed in the air or gas stream unless they are constructed or enclosed in such a manner that failure of that component cannot ignite the surrounding gas stream.

5) The user shall electrically ground all fan parts.

6) The use of aluminum or aluminum alloys in the presence of steel which has been allowed to rust requires special consideration. Research by the U.S. Bureau of Mines and others has shown that aluminum impellers rubbing on rusty steel may cause high intensity sparking.

The exact method of construction and choice of alloys is the responsibility of the manufacturer; however, the customer must accept both the type and design with full recognition of the potential hazard and the degree of protection required.

FAN DRIVES

Fans are typically driven by alternating current (AC) motors. In industrial fan applications, the most common motor type is the squirrel-cage induction motor. This motor type is commonly used because of its characteristic durability, low cost, reliability, and low maintenance. These motors usually have 2 or 4 poles, which, on a 60- hertz system, translate to nominal operating speeds of 3,600 revolutions per minute (rpm) and 1,800 rpm, respectively. Although motors with 6 poles or more are used in some fan systems, they are relatively expensive. The most common class of motors for fan applications is NEMA

Design B. Service factors range from 1.1 to 1.15, meaning that the motors can safely operate at loads between 110 to 115 percent of their horsepower (hp) ratings.

The installed motor should be checked for sufficient starting torque to overcome the inertia of the fan wheel and drive package, and accelerate the fan to its design speed. A characteristic of induction motors is that their torque is directly related to slip, or the difference between the speed of the magnetic field and the speed of the motor shaft. Consequently, in many fans, actual operating speeds are usually around 2 percent less than their nominal speeds. For example, a theoretical four-pole induction motor with no slip would rotate at 1,800 rpm with a 60-hertz power supply; however, rated operating speeds for this motor are usually around 1,750 rpm, indicating that slip rates are a little over 2.7 percent at rated load. Fans that are driven by older motors are probably operating at much lower efficiencies and at higher levels of slip than what is available from new motors. Remember the fan laws; the fan operating at low RPM will provide low airflow. EPAct efficiency motors operate with less slip, which means fans rotate at slightly higher speeds.

Fan Motor Controller

All fans are suitable for direct on line starting manually or automatically up to and including 5.5kW. The number of starts should be limited to no more than four direct-on-line starts per hour or, no more than eight starts per hour for motor up to 1kW. The controller is the switch mechanism that receives a signal from a low power circuit, such as an on/off switch, and energizes or de-energizes the motor by connecting or disconnecting the motor windings to the power line voltage. In conventional systems, the high in-rush and starting currents associated with most AC motors creates power quality problems, such as voltage sag. Soft starters gradually ramp up the voltage applied to the motor, reducing the magnitude of the start-up current. As industrial facilities increase the use of computer-based equipment and control systems, soft starters are becoming important parts of many motor control systems. In fact, a major advantage associated with most VFDs is that they often have built-in, soft-start capabilities. The VFD provides multiple speed capability and provides energy savings in cube root of RPM.

Fan Drive System

Motors are connected to fans either directly, through a gearbox, or, more commonly, by a belt system. There are advantages and drawbacks to each drive option. Understanding how drives are selected can be helpful in correcting problems that are the result of poor design.

1) **Direct Drives**

In ***direct drive systems***, the fan is attached to the motor shaft. This is a simple, efficient system but has less flexibility with respect to speed adjustments. Because most fans are operated with induction motors, the operating rotational speeds of direct drive fans are limited to within a few percent of the synchronous motor speeds (most commonly 1,200, 1,800, and 3,600 rpm). The sensitivity of fan output to its operating rotational speed means that errors in estimating the performance requirements can make a direct-drive system operate inefficiently (unlike belt drives, which allow fan rotational speed adjustments by altering pulley diameters). In axial fans, direct drives have some important advantages. Applications with low temperatures and clean system air are well suited for direct drives because the motor mounts directly behind the fan and can be cooled by the airstream. This space-saving configuration allows the motor to operate at higher-than-rated loads because of added cooling. However, accessibility to the motor is somewhat restricted. Direct drives have several advantages over belt drives, including higher efficiency, compact space requirements, and lower maintenance. Although belt drives are occasionally used in fan applications over 300 hp, they are rarely found in fan applications over 500 hp. At these power levels, the efficiency advantages of direct drives are very attractive.

One way to add rotational speed flexibility to a direct-drive system is to use an adjustable speed drive (ASD). ASDs allow a range of shaft speeds and are quite practical for systems that have varying demand. ASDs can provide a highly efficient system for fans that operate over a range of conditions.

2) **Belt Drives**

Because the required rotational speed of a fan is usually less than motor rpm, belts are used to transfer power from a motor pulley (sheave) to a fan pulley with a larger diameter. The desired fan rotational speed can be achieved using various pulley sizes according to the following relationship:

$$\text{Drive Ratio} = \frac{\text{Motor RPM}}{\text{Desired Fan RPM}}$$

Or

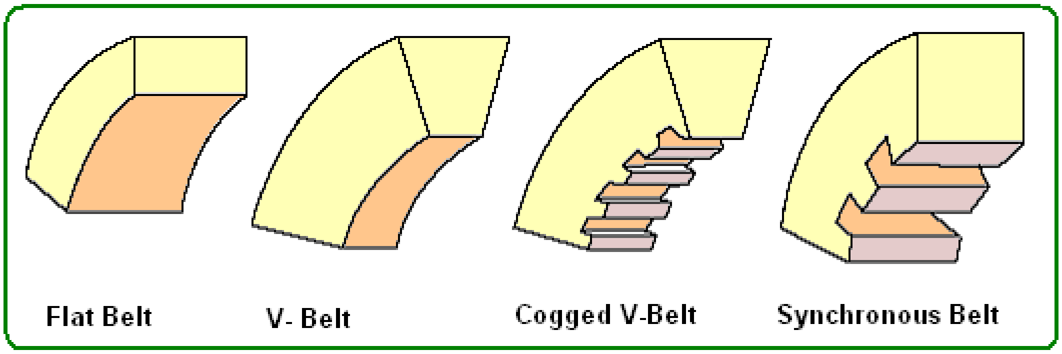
$$RPM_{driven} = RPM_{driver} \times \frac{D\ driver}{D\ driven}$$

Where D is the diameter of the pulleys

Most industrial fan belt drive applications are limited to speed ratios below 4:1 (the motor speed is 4 times faster that the fan speed); however, for small horsepower applications (less than 1 hp), this ratio can be as high as 10:1. The limiting factors on speed ratios are the practical size of the pulleys, the arc of contact between the belt and the drive pulley, and belt speed.

Belt drives offer a key advantage to fan systems by providing flexibility in fan speed selection. If the initial estimates are incorrect or if the system requirements change, belt drives allow flexibility in changing fan speed. In axial fans, belt drives keep the motor out of the airstream, which can be an advantage in high temperature applications, or in dirty or corrosive environments. The four principal types of belts are flat, V-belts, cogged V-belts, and synchronous.

- Flat belts have a uniform cross-section and transmit power through friction contact with flat pulley surfaces.
- V-belts are an improvement over the flat belt, using a wedging action to supplement friction-based power transfer. V-belts have a long history in industrial applications, which means there is a lot of industry knowledge about them. An important advantage to V-belts is their protection of the drive rain during sudden load changes. Service conditions that experience sudden drive train accelerations causes accelerated wear or sudden failure.
- Cogged V-belts offer the same advantages as V-belts; however, their notched design provides additional flexibility that allows the use of smaller pulleys. Cogged V-belts are slightly more efficient than conventional V-belts, because of their added flexibility and the fact that the notched surface transfers force more effectively. In applications where a small arc of contact is unavoidable, the use of cogged V-belts is recommended.
- Synchronous belts offer many advantages over standard flat belts and V-belts. By using a mesh engagement, synchronous belts are the most efficient type of belt drive because they do not suffer efficiency losses through slip. Synchronous belts have teeth that engage with grooves in the sheave. Synchronous belts can allow lower belt tension than conventional belts, reducing the radial loads on motor and fan bearings and extending their operating lives. Further, synchronous belts do not lose efficiency as they wear. However, synchronous belts are very noisy, which often discourages their use. They also transfer shock loads through the drive train without allowing slip. These sudden load changes can be problematic for both motors and fans. Another problem with synchronous belts is the limited availability of pulley sizes. Because the pulleys have a mesh pattern, machining them alters the pitch diameter, which interferes with engagement. Consequently, pulleys are available in discrete sizes, which preclude an important advantage of belt drives: the ability to alter operating rotational speeds by adjusting sheave diameters. Because of these factors, synchronous belts are not as widely used as V-belts in fan applications.



The required belt capacity must not only include the horsepower required by the driven load; it must also account for site-specific factors, such as temperature, service factor, and arc of contact. The effect of temperature varies according to the belt material. Rubber contracts at higher temperatures. Consequently, in belts that have high rubber content, tension and stress increase as the drive system temperature increases. Because temperature also affects the mechanical strength of a belt, belts should be sized to meet the torque requirements at the highest normal operating temperature. The belt service factor accounts for acceleration loads during start-up and under load changes. For most fans, the belt service factor is between 1.2 and 1.4.

In general, synchronous belts are the most efficient, while V-belts are the most commonly used.

---

3) **Gear Drives**

Gear drives are not as common as belt or direct drives, but are useful in a few applications that require special configurations between the fan and motor. Gear systems have a wide range of efficiencies that depend on gear design and speed ratio. Gear systems can be very robust, affording high reliability—a characteristic that is very important in applications with restricted access to the drive system. However, gears, unlike belt systems, do not allow much flexibility in changing fan speed. Gear-system efficiency depends largely on speed ratio. In general, gear efficiencies range from 70 to 98 percent. In large horsepower (hp) applications (greater than 100 hp), gear systems tend to be designed for greater efficiency because of the costs, heat, and noise problems that result from efficiency losses. Because gears require lubrication, gearbox lubricant must be periodically inspected and changed. Also, because gears—like synchronous belts—do not allow slip, shock loads are transferred directly across the drive train.

Bearing Life

Bearing life is determined in accordance with methods prescribed in ISO 281/1-1989 or the Anti Friction Bearing Manufacturers Association (AFBMA) Standards 9 and 11, modified to follow the ISO standard. The life of a rolling element bearing is defined as the number of operating hours at a given load and speed the bearing is capable of enduring before the first signs of failure start to occur. Since seemingly identical bearings under identical operating conditions will fail at different times, life is specified in both hours and the statistical probability that a certain percentage of bearings can be expected to fail within that time period.

For Example, a manufacturer specifies that the bearings supplied in a particular fan have a minimum life of L-10 in excess of 40,000 hours at maximum cataloged operating speed. We can interpret this specification to mean that a minimum of 90% of the bearings in this application can be expected to have a life of at least 40,000 hours or longer. To say it another way, we should expect less than 10% of the bearings in this application to fail within 40,000 hours.

L-50 is the term given to Average Life and is simply equal to 5 times the Minimum Life. For example, the bearing specified above has a life of L-50 in excess of 200,000 hours. At least 50% of the bearings in this application would be expected to have a life of 200,000 hours or longer.

Some common conditions to consider in designing a satisfactory drive are:

1) Drives should be installed with provisions for center distance adjustment.
2) This provision is important because all belts stretch.
3) Centers should not exceed 2-½ to 3 times the sum of the sheave diameters or be less than the diameter of the larger sheave.
4) The arc of contact on the smaller sheave should not be less than 120 degrees.
5) Sheave diameter ratios should not exceed 8:1.
6) Belt speed preferably should not exceed 5,000 ft/min, or be less than 1,000 ft/min—4,000 ft/min is the best practice.
7) Sheaves should be dynamically balanced for speeds in excess of 5,000 ft/min rim speed.

Some helpful points to watch for when installing drives are as follows:

1) Be sure that shafts are parallel and sheaves are in proper alignment. Check again after a few hours of operation.
2) Do not drive sheaves on or off shafts. Wipe shaft, key, and bore clean with oil. Tighten screws carefully. Recheck and retighten after a few hours of operation.
3) Belts should never be forced over sheaves.
4) In mounting belts, be sure the slack in each belt is on the same side of the drive. This side should be the slack side of the drive.
5) Belt tension should be reasonable. When in operation, the tight side of the belts should be in a straight line from sheave to sheave, and with a slight bow on the slack side. All drives should be inspected periodically to be sure belts are under proper tension and are not slipping.
6) When making replacements of multiple belts on a drive, be sure to replace the entire set with a new set of matched belts.

FAN SAFETY & TESTING

Safety

Rotating fan impellers and electric motors can be dangerous to personnel; only experienced qualified persons should carry out work on these products. The following precautions must be taken:

1) Isolate electrically the fan motor prior to undertaking any work.
2) Check that all fasteners, particularly impeller fasteners, are tight prior to start up. Do not re-use locking fasteners.
3) Loose objects are not only a safety hazard but may also cause major damage. Ensure all attachments of any sort are secure and cannot be drawn into the impeller.
4) Always use protective guards where fans are accessible to personnel or directly exposed to habitable areas.
5) Ensure that loose debris will not be sucked into the fan prior to fan start up. All ductwork should be clean.

Fan Testing

Fans are tested in accordance with strict requirements of AMCA (Air Moving and Conditioning Association) Standard 210. This standard specifies in detail the procedures and setups to be used in testing the various types of fans. Fans are tested under these standard test conditions so that all fans are rated on an equal basis. Thus, fans of different manufacturers and of different types can be rated and compared using the same basis of testing and obtaining performance data. Fans are tested and performance certified under ideal laboratory conditions. When fan performance is measured in field conditions, the difference between the ideal laboratory condition and the actual field installation must be considered. Consideration must also be given to fan inlet and discharge connections, as they will dramatically affect fan performance in the field. If possible, readings must be taken in straight runs of ductwork in order to ensure validity. If this cannot be accomplished, motor amperage and fan RPM should be used along with performance curves to estimate fan performance.

FAN INSPECTION

Satisfactory applications occur when all aspects of the installation are in harmony with each other. Proper operation, constant monitoring and maintenance are also part of the equation.

The following checklist from AMCA 202 contains the items to be inspected:

1) All fan parts and accessories should be installed, aligned and operational.
2) Check all tie down bolts so that the fan is firmly held in place on its foundation.
3) Check all ductwork connections so that flexible material does not "suck in", leak or become short circuited by having the fan support ductwork or other parts of the system.
4) Check that all driveline components such as bearings, couplings, v-belts, motors etc. are aligned and properly tensioned. Make sure all v-belts are matched and that bearings have been tightened to the base and to the shaft. Check that bearings are properly lubricated with the proper type and amount of grease.
5) Check that the fan wheel is properly aligned with the inlet bell and housing, is free to turn and that when momentarily energized it will turn in the right direction.
6) Check the fan and system for any obstructions, build-up, leaks, missing parts etc.
7) Run the fan at full speed. Verify that the fan is running close to the design speed. Determine whether the fan is running smooth and that the bearings are not running hot. Obtain a power measurement to make sure the fan is not overloading the motor.
8) Let the fan run for twenty-four hours. Recheck all of the items listed above once again, particularly the v-belt tension.
9) Scheduled inspection of fans is recommended. Items checked should include:
    - Bearings for over heating (lubricate or replace as required).
    - Belt drives for proper tension to prevent slipping.
    - Fan wheel for proper rotation.
    - Dust accumulations on fan blades, housings, and shutters.
    - Weeds and shrubs growing outside the greenhouse that block the fans.

The results of this initial inspection should be kept on record for future reference. If a problem does occur later on, it will serve as a beginning point of any evaluation.

FAN MAINTENANCE

To obtain maximum service life of your fans, it is recommended the following maintenance be implemented and recorded in a plant log book

1) Ensure air intake space is unobstructed to avoid overheating the motors.

2) On a weekly basis use an air hose to ensure all airways are clear and free of dust if the fans are not used.
3) Do not wash the fan motor down unless the motor is IP66 rated.
4) On a quarterly basis
   - Check motor terminals for tightness and contact.
   - If terminal lugs are discolored, re-terminate.
   - Check the operation of starting equipment, ensuring all terminations are tight.
   - Check mechanical operation of thermistor relay (if fitted).
   - Check operation of space heaters (if fitted).
5) On a six (6) monthly basis, in addition to the above item 4
   - Check stator resistance (compare to original and enter in log book).
   - Check supply voltage at motor terminals.
   - Check bearings for noise / overheating.
   - Check the tightness of fastening.
6) On an annual basis, in addition to items 4 and 5
   - Strip motor out and clean thoroughly.
   - Check bearings for wear / damage - Replace as necessary.
   - Check all fastenings and impeller blades for cracks or damage - Replace as necessary.
   - After re-assembly, check and record:
     - ✓ Full load Current
     - ✓ Full load Voltages
     - ✓ Full load speed
7) Ensure plant logbook records commissioning data and compare maintenance data to original.
8) Contact your nearest Fans Direct office for any part replacement.

Note:

Before proceeding with routine maintenance ensure that:-

1) The fan and speed control, if fitted, are electrically isolated.
2) Sufficient time is allowed for impeller to stop completely

---

FAN TROUBLESHOOTING

Troubleshooting is the effort to identify and resolve differences between what was expected and what actually happened. The following four categories narrow the focus of attention and speeds up the evaluation process.

1) **Aerodynamic Performance -** This applies to any of the five rating parameters of flow, pressure, speed, power and density and how they compare to their respective design quantities.
2) **Noise -** This applies to any problem in which the ears are the main sensor. Noise and vibration are similar in that they both have amplitude and frequency, but noise is a much lower amplitude and energy content and is measured in dB referenced to Watts. Generally speaking, noise has a much wider frequency range and a higher upper limit than vibration (63Hz to 10 KHz).

3) **Vibration -** This applies to any problem in which the hands or touching are the main sensor. Amplitude is large when there is a problem. It has much greater energy content with a smaller frequency range (3Hz to perhaps 500 Hz).

4) **Premature Failure -** Premature Failure applies to anything whose life does not meet that which was expected. The term "failure" does not necessarily mean a catastrophic failure such as when something "blows up", but a length of time considered as being the useful life of the component.

Troubleshooting must be employed when one of these areas becomes a problem.

Aerodynamic Performance Troubleshooting Symptoms

Symptoms			Possible Causes
Flow	**Pressure**	**Power**	
Low	***Low***	***Low***	• Speed is lower than design, v-belts slipping, wrong sheaves
			• Air temperature is higher than design
			• Axial fan blade settings lower than design
			▪ Suction pressure correction not included in density calculation
			• Damper mounted directly on fan outlet
			• Failure to include plenum losses in rating
			Insufficient length of duct on fan outlet
Low	***High***	***Low***	• System losses higher than design, blockage, dampers closed, dirty coils, dirty filters
			• Fan operating in stall behind peak pressure
High	***Low***	***High***	• System losses lower than design, components missing, leaks
High	***High***	***High***	• Speed is higher than design
			• Air temperature lower than design
			• Swirl opposite to fan rotation, inlet elbows
			• Wrong wheel rotation (clockwise vs. counter clockwise etc.)
			• Fan running backwards
			• Axial fan blades set higher than design
Normal	***Normal***	***High***	• Undersized motor

Symptoms			Possible Causes
Flow	**Pressure**	**Power**	
			• Excessive driveline losses, wheel or seal rubbing, tight bearings, too many oversized/misaligned v-belts
			• Motor misaligned, wrong voltage, wired wrong
			• Improperly matched motor and VFD controller
Unsteady	***Unsteady***	***Unsteady***	• Fan blocked off, operating in stall or on unstable part of curve
			• Fan and system hunting due to flat portion of curve
			• Fans in parallel and not rated properly

Premature Failure Troubleshooting Symptoms

Component	Possible Causes
Housings	• Paint failure resulting in rusting
	• Fatigue caused by loose parts which break off
	• Fatigue caused by turbulence
	• Lack of attachment to foundation
	• Ductwork or other equipment attached to fan
	• Improper storage
Wheels	• Loose rivets or bolts
	• Long term unbalance
	• Wear or corrosion
	• Loose attachment to shaft
	• Uneven build-up on wheel
Shafts	• Bent shaft causing long term vibration
	• Undersized shaft causing looseness at wheel, bearings
	• Undersized shaft causing operation near shaft critical

Component	Possible Causes
	• Improper storage
Bearings	• Lubrication: too little, too much, contaminated, wrong kind
	• Shaft to bearing clearance too large
	• Axial thrust too large
	• Minimum radial load not maintained
	• Belt pull too large due to small sheave
	• Too many v-belts
	• Operating or ambient temperature too high
	• Improper storage
Sheaves	• Loose attachment to shaft
	• Wrong v-belt cross section
	• V-belt tension not correct
	• V-belt misalignment
	• Too many start-stops
V-belts	• Under designed to take power
	• Incorrect tension or not matched, misaligned
	• Belt speed too high
	• Operating or ambient temperature too high
Motors	• Overloading of motor
	• Incorrect voltage
	• VFD controller lines too long causing voltage spikes
	• VFD controller and motor not matched causing eddy current induced fields and bearing pitting
	• Belt pull too large due to small sheave, too many belts
	• Wrong motor enclosure for environment

Some fan system problems, such as abnormally high operating and maintenance costs and ineffective airflow control, are sufficiently troublesome to justify a system assessment. If the system problems are significant, then a change to the fan, its drive system, or the airflow control devices may be justifiable.

Selecting a new, larger fan requires consideration of the same factors that are involved in any initial fan selection. A new fan may be more feasible if the existing one has degraded or requires extensive refurbishment. In high run-time applications, the purchase of a new fan with an energy-efficient motor may provide an attractive payback.

Belt Drive Maintenance Guidelines

Belt tension and alignment should be checked periodically. Proper belt tension is typically the lowest that prevents a belt from slipping at peak load. An important maintenance practice to avoid is the use of belt dressing. Belt dressing is a surface treatment that increases the level of friction between a belt and pulley. Because it masks the fundamental cause of slippage, belt dressing only provides a temporary means of reducing noise. Belt slippage should be corrected by either cleaning the drive system or adjusting belt tension.

When installing or replacing belts, ensure they are oriented correctly in accordance with the directions of the manufacturer. Belts are often tagged to show the preferred direction of rotation. Although some belts can be operated in either direction, belt manufacturers often test their belts in one direction and package them with an indication of this direction.

In high-temperature applications, new belts should be operated under low-load conditions and at normal operating temperature for a reasonable period. This run-in time increases the creep strength of the belt.

1) Drives should always be installed with provision for center distance adjustment.
2) If possible centers should not exceed 3 times the sum of the sheave diameters nor be less than the diameter of the large sheave.
3) Be sure that shafts are parallel and sheaves are in proper alignment. Check after first eight hours of operation.
4) Do not drive sheaves on or off shafts. Be sure shaft and keyway are smooth and that bore and key are of correct size.
5) Belts should never be forced or rolled over sheaves. More belts are broken from this cause than from actual failure in service.
6) In general, ideal belt tension is the lowest tension at which the belt will not slip under peak load conditions. Check belt tension frequently during the first 24-48 hours of operation.

In summary, note the following key points:

Problem	Possible Cause of Problem
Air Flow Low	1) Air leakage in duct system 2) Poor inlet / outlet conditions i.e. bends or abrupt terminations, loose flexible connections 3) Fans running backwards 4) Measurements using pitot traverse is required 2 - 3 meters away from fan on straight run of duct. 5) No turning vanes on bends 6) Site measurements may be 10% out with manometers

Problem	Possible Cause of Problem
	and 20% out with anemometers 7) Dampers not adjusted properly 8) System resistance higher than specified 9) No guide vanes to fan inlet when swirl at fan inlet is the same as fan rotation 10) Filters dirty 11) If fan fitted with adjustable pitch impeller check pitch angle is correct
Air Flow High	1) Dampers not adjusted properly 2) System resistance is lower than specified 3) Wrong type of fan selected for application 4) If fan fitted with adjustable pitch impeller check pitch angle is correct
Electrical	1) Wrong supply voltage 2) Three phase motor has been single phased 3) Check motor overloads have been set correctly 4) Some external rotor motors must be wired in star (check the correct wiring diagram) 5) Check electrical supply at switchboard.
Noise	1) Axial fan may be operating in stall position 2) Poor inlet conditions to fan causing instability 3) Poor fan selection i.e. speed of fan too high 4) Poor inlet and outlet conditions 5) Flexible connections too slack 6) No attenuators fitted to reduce system noise 7) Site conditions can effect noise rating of fans

Annexure- 1

DEFINITIONS & TERMINOLOGY

1) Adjustable Pitch - Vane axial rotor blades, which may be manually adjusted to various pitches. Fan must be off, electrical power locked out, blade retaining nuts loosened and blades manually set to desired pitch (within horsepower limitations).

2) Air Density (ρ): The mass per unit volume of the air, expressed in Kg/m^3 or Lbs/ft^3

3) Air Flow Rate (Q): The volume of air moved by the fan per unit of time, expressed in CFM (cubic feet per minute) or m^3/s.

4) Air Power (Static): That part of the energy, per unit time, imparted by the fan to the air in overcoming Static Pressure (PS) from that at the inlet to that at the outlet.

5) Air Power (Total): That part of the energy, per unit time, imparted by the fan to the air by increasing its Total Pressure (Pt) from that at the inlet to that at the outlet.

6) Air Volume- The cubic feet per minute (cfm) of air handled by a fan at any air density. This is different from the cubic feet per minute of standard air (scfm), which is at 0.075 lb/ft.

7) Axial Flow Fan: A fan in which the flow of air is substantially parallel to the axis of the impeller rotation.

8) Brake Horsepower- The actual horsepower required driving the fan. This number is greater than a theoretical "air horsepower" because it includes loss due to turbulence and other inefficiencies in the fan, plus bearing losses. It is the power furnished by the fan motor.

9) Controllable Pitch - Also known as “response control” or “in-flight controllable pitch”. Rotor blades are capable of changing pitch while the fan is running. Pitch change is accomplished by electric or pneumatic actuation.

10) Efficiency: The measure of how well a fan does work against the total pressure to move a given volume of air is termed as Fan Total Efficiency. In general, a non-uniform air stream (flow) results in a less efficient fan. A fan equipped with a duct having proper inlet and outlet shapes, allows for a more uniform air stream (flow) and thus dramatically improves the efficiency of the fan; in fact, if properly designed, a ducted fan can achieve efficiencies of up to 85%.

11) Fan: A device for moving air, which utilizes a power driven rotating impeller. A fan shall have at least one inlet and one outlet. A fan induces airflow by virtue of its blades; a blade moves air by generating a lift force when in motion through the air.

12) Fan Characteristics: The curves depicting the relationship between airflow rates, total pressure, fan power input and fan total efficiency at a specified pitch angle and RPM.

13) Fan Duty (Static): The volume of air to be moved, by the fan, at a specified static pressure (Ps)

14) Fan Duty (Total): The volume of air to be moved, by a fan, at a specified total pressure (Pt)

15) Fan Power Input (HI): The energy input, per unit time, required driving a fan, expressed in Break Horse Power (BHP) or Break Kilowatt (BKW).

16) Fan Speed (N): The number of revolutions of the fan about its axis per unit time, expressed in revolutions per minute (RPM).

17) Fan Total Efficiency: The ratio of air power (total) to the fan power input.

18) Fan Static Efficiency: The ratio of the air power (static) to the fan power input.

19) Hub - The center of the rotor. Hubs contain a provision for attachment to the driven shaft and machined sockets or holes for attaching the blades. The hub is usually covered by a nosecone (a spun aluminum cover for streamlining the hub).

20) Outlet Velocity- The theoretical velocity of the air as it leaves the fan outlet. This velocity is calculated by dividing the air volume in cfm by the fan outlet area in square feet. Since the velocity

varies over the cross-section of all fan outlets, this value is only a theoretical value that could occur at a point removed from the fan. Because of this, all velocity readings, including total pressure and static pressure, should be taken farther along in a straight duct connected to the fan discharge where the flow is more uniform.

21) Rotor - A term used to describe the vane axial propeller. The rotor consists of a hub and blades plus the pitch control mechanism for in-flight controllable rotors.

22) Scroll: A volute shaped casing in which impeller rotates and it is this casing which identifies the centrifugal fan

23) Stall: The region of instability in fan performance caused by the separation of the air flow from the surface of the fan blade. The stall condition is depicted by a dip in the performance curve.

24) Standard Air: Atmospheric air having a specific weight of 1.2 kg/m^3 which is dry air at 20° C and 50% relative humidity with a barometric pressure of 760 mm Hg.

25) Static Pressure Margin: The pitch angle margin available between the selected operating point and stall point on the performance curve.

26) Static Pressure (Ps): A fan moving air must work to overcome resistance to the flow that arises due to various obstructions in the flow path. This resistance is termed as Static Pressure. The total Static Pressure is expressed in millimeters or inches of H2O.

27) Static Regain - Conversion of the energy of motion (kinetic energy) or velocity pressure to potential energy or usable static pressure. An example is the increase in static pressure as velocity is reduced across an outlet cone.

28) System Effect - A pressure loss resulting from fan inlet or outlet restrictions or other condition within the system affecting fan performance. System effect is difficult to quantify and results in poor efficiency, noise and vibration.

29) Tip Clearance: The clearance between the fan blade tip and the fan casing wall.

30) Tip Speed (TS): The linear speed at the fan blade tip at a given fan RPM, expressed in ft/min or m/s.

31) Total Pressure (Pt): The total work a fan must do to move a specified volume of air against the static pressure plus the velocity pressure is defined as Total Pressure of the system and is expressed in millimeters or inches of H2O.

32) Turndown - A reduction in volume created by a blade pitch change on controllable pitch vane axial fans. Turndown ratio is the ratio of air volume from minimum to maximum blade pitch. The amount of turndown available depends upon the operating point selected, fan size and hub size.

33) Vane Axial Fan - An air moving device with axial airflow and straightening vanes to reduce vortex created by the rotor.

34) Velocity (V): The rate of airflow divided by the net area of the airflow, and is expressed in M/S or ft/min.

35) Velocity Pressure (PV): The portion of air pressure, which exists by virtue of the rate of motion only, expressed in mm or inches of water column, or, Pascals (Pa).

36) Vortex - Airflow rotating perpendicular to the intended axis of airflow. It is a swirling movement of air generated by the vane axial rotor.

Annexure-2

USEFUL FORMULAS

Parameter	Formula	Nomenclature
Air Velocity	$V = \frac{Q}{A}$	V = Air velocity in FPM Q = Air flow rate in CFM A = Net area in ft^2
Total Pressure	$P_t = P_s + P_v$	P_t = Total pressure, in-wg P_s = Static pressure, in-wg P_v = Velocity pressure, in-wg
Velocity Pressure	$P_v = \left(\frac{V}{4009}\right)^2$	P_v = Velocity pressure, in-wg V = Air velocity in FPM
Fan Output Power	$H_o = \frac{P_t \times Q}{6360}$	H_o = Fan output power, HP P_t = Total pressure, in-wg Q = Air flow rate in CFM
Total Efficiency	$\eta_t = \frac{H_o}{H_i} \times 100$	ήt = Total efficiency H_o = Fan output power, HP H_i = Fan input power, HP
Tip Speed	$T_s = \pi \times D \times N$	Ts = Fan tip speed, FPM D = Fan diameter, ft N = Fan speed, RPM
Net Free Area	$A = \frac{\pi}{4}(D^2 - d^2)$	A = Net free fan area, ft^2 D = Fan diameter, ft d = Seal disc diameter, ft

Parameter	**Formula**	**Nomenclature**
Solidity Ratio	$\dfrac{\text{Sum of fan blade tip chords}}{\pi \times D}$	D = Fan diameter, ft
Effect of Air Density	1) Air flow rate remains the same with a change in air density 2) Pressure varies in proportion to the air density $P_2 = P_1 \times \left(\dfrac{\rho_2}{\rho_1}\right)$ 3) Fan input power varies in proportion to the air density $H_2 = H_1 \times \left(\dfrac{\rho_2}{\rho_1}\right)$	P_2 = Pressure at duty point 2, in-wg P_1 = Pressure at duty point 1, in-wg H_2 = Fan input power at duty point 2, in-wg H_1 = Fan input power at duty point 1, in-wg ρ_2 = Density at duty point 2, lb/ft^3 ρ_1 = Density at duty point 1, lb/ft^3
Effect of Fan Speed	1) Fan air flow rate varies directly with fan speed ratio $Q_2 = Q_1 \times \left(\dfrac{N_2}{N_1}\right)$ 2) Fan pressure varies with square of fan speed ratio $P_2 = P_1 \times \left(\dfrac{N_2}{N_1}\right)^2$ 3) Fan power varies with cube of fan speed ratio $H_2 = H_1 \times \left(\dfrac{N_2}{N_1}\right)^3$	Q_2 = Air flow rate at duty point 2, CFM Q_1 = Air flow rate at duty point 1, CFM P_2 = Pressure at duty point 2, in-wg P_1 = Pressure at duty point 1, in-wg H_2 = Fan input power at duty point 2, in-wg H_1 = Fan input power at duty point 1, in-wg N_2 = Fan speed at duty point 2, RPM N_1 = Fan speed at duty point 1, RPM

Annexure-3

STANDARD UNITS & CONVERSION

1) **Air Flow (Q) - m^3/s**
   - $m^3/h = 3600 \times m^3/s$
   - $CFM = 0.59 \times m^3/h$
2) **Velocity (V) – m/s**
   - FPM = 196.8 x m/s
3) **Area (A) – m^2**
   - $ft^2 = 10.76 \times m^2$
4) **Pressure (P) – Pa (Pascals)**
   - mm(water column) = Pa/9.8
   - in (water column) = mm/25.4
5) **Power (W) – Watts**
   - hp = watts / 746
6) **Density (ρ) – kg/m^3**
   - Standard air (ρ) = 1.2 kg/m^3

Annexure-4

TYPICAL CUSTOMER APPLICATIONS

Fan and blower selection depends on the volume flow rate, pressure, type of material handled, space limitations, and efficiency. Overall system efficiency will be determined by the type of fan or blower, its interaction with the air distribution system and the method of control. Table below lists a few of the many applications and the type of equipment typically used.

Application	Type of Fan or Blower
Material conveying systems with high air/material ratio and fine, granular materials	Radial, Backward inclined fans Centrifugal blowers
Material conveying systems with low air/material ratio and materials prone to clogging distribution system	Radial fans and Positive-displacement blowers
Supplying air for combustion	All fan types
Boilers, forced draft	Airfoil, Backward inclined, Vane-axial fans
Boosting gas pressures	Centrifugal blowers
Boilers, induced draft	Forward curved, Radial fans
Kiln exhaust	Radial fans
Kiln supply	Airfoil, Backward inclined, Vane-axial fans
Process drying	Airfoil, Backward inclined, Radial, Vaneaxial, Tube-axial fans Centrifugal blowers
Aeration and agitation systems	Centrifugal, Positive-displacement blowers
Plant ventilation and HVAC (clean air only),	Airfoil, Backward inclined, Forward curved Vane-axial, Tube-axial, Propeller fans
Air knife blow off systems, clean-up air supply, vacuum cleaning systems	Centrifugal blowers
Process Heating/Cooling Air Handling	Centrifugal backward inclined,

units	Forward curved, Radial
Kitchen/toilet extract	Propeller, Axial

****