Application Engineering Handbook

176 Colchester Road, Bayswater VIC 3153 Australia
+61 3 9738 1999 | sales@aerotechfans.com.au

9. FAN APPLICATIONS

Fans can be used for numerous applications. Here are some typical applications, some for blowing and some for exhausting.

- Adhesive Industry
- Aeration Systems
- Agricultural Machinery
- Air Blow-Off/Air Knifes
- Air Cushions & Air Tables
- Air Slides
- Air-Conditioning
- Aquaculture- Pond Aeration
- Aquarium Aeration
- Bag Houses
- Bake Ovens
- Boilers
- Bottling Machinery
- Bulk Material Handling
- Canning Machinery
- Cement Industry
- Ceramic Industry
- Chemical Industry
- Clothing & Textile Machinery
- Collieries
- Container Manufacturing
- Conveyor Cooling
- Cooling Systems
- Cryogenic Applications
- Cyclones
- Dental Equipment
- Drying Systems
- Dust Collection
- Electronics Cooling
- Electroplating Tank Agitation
- Environmental Control
- Filtration
- Fish Farm Hatcheries
- Floatation Systems
- Fluidised Beds
- Food Processing
- Forced Draft
- Foundry Industry
- Furnace Equipment
- Gas Evacuation
- Glass Manufacturing
- Grain Handling
- Heat Recovery
- Incineration Systems
- Induced Draft
- Industrial
- Iron And Steel Industry
- Labelling Machinery
- Laboratory Filtration
- Liquid Media Agitation
- Marine Industry
- Materials Handling
- Media Agitation
- Mine Ventilation
- Mineral Processing
- Miscellaneous Applications
- Oil & Gas Industry
- Packaging Equipment
- Paper And Pulp Processing
- Petrochemical Industry
- Pharmaceutical Industry
- Pickling Exhaust
- Plastics Industry
- Pneumatic Air Tube Systems
- Pneumatic Conveying
- Pollution Control
- Power Generation
- Printing Machines
- Pulp & Paper Industry
- Rubber Processing
- Screen Printing Machines
- Scrubbers
- Solvent Recovery
- Spa/Hot Tub Aeration
- Spray Booths
- Steam Presses
- Steel Mill Processing
- Sterilisation
- Suction
- Textile Industry
- Timber Industry
- Tunnel Ventilation
- Vacuum Hold-Down & Pick-Up
- Vacuum Lifting Gear
- Vacuum Transport Systems
- Ventilation
- Waste Treatment
- Wastewater Treatment
- Welding Fume Extraction
- Woodworking Machinery
- Wool Processing
- Etc.

10. DUST COLLECTION AND FUME REMOVAL

10.1 Hood Design
This is a lengthy subject and quite well covered in many available publications.

10.2 Minimum Conveying Velocity
All airborne material except fumes and the finest of dusts will settle in ducts and fans if a minimum velocity is not maintained. Since in dust collecting the air quantity is determined by the hood size and velocity it is usually most economical to maintain the lowest duct velocity that will keep the material in suspension. Raising the velocity raises the absorbed power.

Minimum conveying velocities imply that all factors, including surface condition of the material, must be considered. A flat particle requires a lower velocity to convey than a round. A sticky, wet, or odd-shaped piece may require a considerably higher velocity than that indicated in the chart. If the material contains moisture, the total weight should be used in calculating air volume and velocities.

10.3 Capture Velocity
The capture velocity is the velocity of air necessary to move contaminants external to the hood into the hood and overcome competing air currents.

The terminal or settling velocity is the final settling velocity of a particle in air. The terminal velocity (V) of a particle can be estimated by the following relationship.

$V = 26.42 \times S.G. \times D^2$ where

D = particle diameter in mm (or the equivalent diameter for non-spherical particles)
S.G. = specific gravity
V = terminal or settling velocity in m/s

Example
What is the settling velocity of saw dust of 0.045mm diameter, S.G. = 0.9

$V = 26.42 \times 0.9 \times (0.045)^2 = 0.048$ m/s

11. PNEUMATIC / MATERIAL CONVEYING

The system resistance, when conveying material, will be considerably higher than if the system is handling air alone. Elbows, vertical runs, and whether the material passes through the fan or not, must all be given careful consideration.

To cope with unpredicted overloads, fan ratings should be selected in the steep portion of the fan's pressure volume curve to avoid operation in the unstable region. Motor power should be selected to prevent overload when system is operating below design load conditions.

There are several factors which must be considered when planning an exhaust or conveying system, of these probably the most important are:

a. Risk or damage to the material being conveyed. e.g., materials such as coffee beans, wheat, sugar etc., can be broken or otherwise seriously damaged, particularly if conveyed through the fan.

b. Risk of fire or explosion. e.g., airborne finely divided solids having high calorific value should be treated as explosive. These include powdered aluminium, flour, grain dust, pulverised fuel, hard rubber dust etc.

11.1 Friction Loss With Material

$$\frac{\text{Friction loss of material and air mixture}}{\text{Friction loss of air}} = 1 + 0.359 \times \frac{\text{Weight of conveyed solids}}{\text{Weight of conveying air}}$$

Table 1 below shows the conveying properties of various materials.

A	B	C	D	E	F
Material	Approx. weight (kg/m^3)	m^3 of std. air per kg of material	Kg of material per kg of std. air	Minimum Conveying Velocity (m/s)	Pickup suction (kPa)
Ashes, Coal	480	2.6	0.32	23	0.75
Barley	610	2.4	0.35	25	0.88
Beans, Soy	755	2.2	0.37	26	1.00
Bran	255	3.5	0.24	18	0.50
Cement, Portland,	1600	2.2	0.38	36	1.25
Cinders, Coal	720	2.2	0.37	30	1.00
Coal, Powdered	480	2.6	0.32	20	0.75
Coffee, Beans	675	2.2	0.37	18	0.75
Cork, Ground	225	3.7	0.23	18	0.38
Corn, Cobs	400	2.7	0.30	25	0.63
Corn, Meal	640	2.4	0.35	28	0.88
Corn, Shelled	720	2.2	0.37	28	0.88
Cotton, Dry	80	5.9	0.14	20	0.50
Dust, Grinding	480	2.6	0.32	25	0.75
Fruit, Dried	480	2.6	0.32	20	0.75
Hair or Feathers, Dry	80	5.9	0.14	15	0.38
Lime, Hydrated	480	2.6	0.32	25	0.75
Malt, Dry	560	2.4	0.34	24	0.75
Oats	420	2.7	0.30	23	0.75
Paper, Shredded	320	3.1	0.27	25	0.75
Plastic, Granulated	560	2.6	0.32	27	0.75
Rags, Dry	480	2.6	0.32	23	0.63
Salt, Coarse	720	2.2	0.37	28	1.00
Sand, Dry	1680	2.2	0.38	36	1.25
Sawdust, Dry	210	3.9	0.21	19	0.63
Wheat, Dry	740	2.3	0.36	29	1.00
Wood Chips, Heavy	385	2.8	0.30	23	0.75
Wood Shavings, Light	145	4.6	0.18	17	0.50
Wool, Dry	80	5.9	0.14	25	0.50

Table 1 : Conveying properties of various materials

11.2 Material Conveying Through Fan

The problem of inducing the material into a conveying system is often a difficult one. The best overall method is one that feeds the material into the airstream evenly by either mechanical or gravity means. However, it is often required that the fan pick the material up as well as convey it. One misbelief frequently encountered is that the ability of a system to pick up material is due to the fan's suction pressure. Suction in itself is useless. It is the velocity moving past the material that induces it to flow. For this reason it is important not to plug up the entrance of the duct with material to be conveyed. It should be remembered when figuring entrance loss to a conveying system that where an appreciable amount of bulky material is to be moved it may reduce the effective area of the inlet and thus increase the entrance velocity and loss.

Since the purpose of a conveying system is to move a lot of material (as contrasted to dust collecting) the ratio of material to air volume is quite important.

Whenever material is airborne the fan must provide the energy to move the material. It is reflected as an increased resistance.

11.2.1 Feeding Devices

Material handling systems are normally fed by gravity or by some mechanical means. Wool and cotton are usually handled directly by suction hoods while dryers generally drop the material from the dryer apron into open suction hoppers. Rotary valves, belt and screw conveyors, may be used for feeding chips, grain and granular materials, to maintain uniform flow in the system.

12. PRESSURE BLOWER APPLICATIONS

A pressure blower is basically a centrifugal fan used where pressure is sufficiently high in relation to the volume of air required to make pressure development the primary design problem.

Most pressure blower designs involve the use of radial bladed fan wheels because:

First the peak of the static pressure curve of a radial blade wheel is quite broad, allowing a relatively wide range of air volume at an almost constant pressure as shown in Figure 11.This has resulted in repeated reference to a "flat performance curve" which is actually nothing more than the exaggerated broadening of the peak portion of a fan's pressure curve.

Secondly the radial blade design wheel in a properly designed housing can deliver stable performance even when dampered to almost no air delivery. Since these blowers are selected at the peak of their pressure curve, dampering in a system results in operation to the left of the pressure peak or very close to no delivery performance. Holding a stable pressure at this point without pulsation and surging is particularly important for proper "turn down" in combustion air supply applications of pressure blowers.

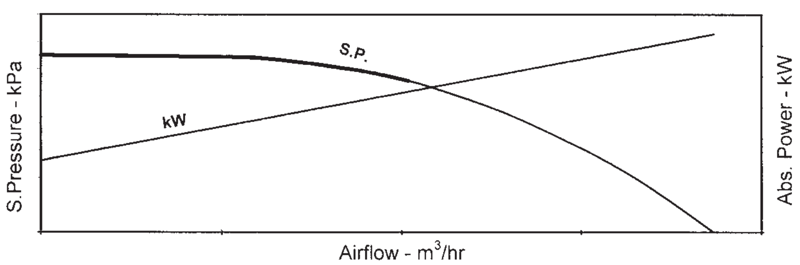


Figure 11. Flat performance curve of pressure blowers

Thirdly, the inherent strength of radial blades in centrifugal action as opposed to other designs allows relatively high tip speeds which are essential to the efficient development of high pressures. Pressure is approximately proportional to the square of the tip speed.

12.1 Selection Principles

Most pressure blower requirements are not very complicated. For the average system, with all of the pressure on the discharge side of the blower, the blower can be selected from the manufacturer's catalogue like any other centrifugal fan.

The blower can be selected near the peak of the pressure curve where considerable margin for error in the air volume can be allowed.

However, suction systems and certain critical conveying systems can be more complicated. In order to understand how to select fans for these applications, it is wise to examine precisely what happens in all types of situations .

The key to understanding pressure blower systems is to think in terms of differential atmospheric pressure and its effect on air density. At sea level, atmospheric pressure is 14.7 psi, or 407 inches W.G., or 101.325 kPa, or 29.92 in.Hg. Therefore, if a blower is operating at standard sea level conditions with an unrestricted inlet and develops 10 kPa gauge pressure on its discharge side, it actually has an inlet pressure of 101.3 kPa and a discharge pressure of 111.3 kPa. The blower in effect moves air into a semi restricted area at a sufficient rate to compress it approximately 10%.

A similar change takes place in a suction system. Atmospheric pressure at the system entrance is 101.3 kPa. If, in order to move a given quantity of air through the system, it is determined that a fan must create a suction pressure of minus 10 kPa, the actual pressure reading at the fan inlet will be 101.3 – 10 = 91.3 kPa. In other words, the air at the fan inlet is rarefied (becomes less dense) by approximately 10%. Since most manufacturers rate their blowers with standard air (1.2 kg/m³ or 20°C and sea level) entering the blower inlet, the blower must be selected at a 10% higher pressure rating in order to develop the actual requirement with the lighter inlet air. With sudden pressure drops and rises, a temperature change also takes place that affects the density of the air. In blow-through systems, where air is compressed suddenly as it goes through the blower, a good rule of thumb is to figure one degree Celsius temperature rise for every kPa pressure rise. In pull-through systems, where there is a pressure drop between the system entrance and the blower inlet, temperature change should be ignored unless the pressure drop is a sudden one. (Example: air going through a minute orifice plate across which there is a sudden pressure drop of 7.5 kPa or more). In such cases, a temperature loss of one degree Celsius for every kPa pressure drop may be figured.

Pressure and temperature changes, with their resultant effect on air density, can be summarised in the following set of principles for selecting pressure blowers.

Density of air in a pressure system is directly proportional to absolute pressure and inversely proportional to absolute temperature.

12.2 Illustrative Examples:

Example 1

Figure 12 shows a pressure blower used for supplying combustion air.

Required: 2500 m^3/hr standard air for proper combustion and 10 kPa static pressure at B.

What actually happens in the system?

1) 2500 m^3/hr of air at 20˚C and 101.3 kPa atmospheric pressure enters blower inlet at A.
2) At B, we require +10 kPa gauge pressure or 101.3 kPa + 10 kPa = 111.3 kPa absolute.

Temperature would increase by approximately one ˚C for every kPa = 20˚C + 10˚C = 30˚C

Density ratio will be $\dfrac{111.3}{101.3} \times \dfrac{(273+20)}{(273+30)} = 1.0625$

At point B, density = 1.2 kg/m^3 x 1.0625 = 1.275 kg/m^3

Capacity = $\dfrac{(2500\ m^3/hr)}{1.065} = 2353\ m^3/hr$

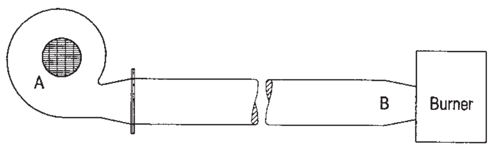


Figure 12. Combustion Air Supply

For combustion air supply application, change in volume and density does not matter as proper amount of oxygen by weight is still available for combustion. Select fan for 2500 m^3/hr of 1.2 k/m^3 density inlet air at 10 kPa S.P.

Example 2

Figure 13 shows a pressure blower used for pneumatic conveying.

Required: 2500 m^3/hr at point B and 10 kPa S.P. at point B

The requirements at point B are the same as in Example 1.
However to get 2500 m^3/hr at point B, a larger volume of standard air must be taken in at fan inlet ie. 2500 m^3/hr x 1.0625 = 2656 m^3/hr. Select fan for 2656 m^3/hr of 1.2 kg/m^3 inlet air at 10 kPa S.P.

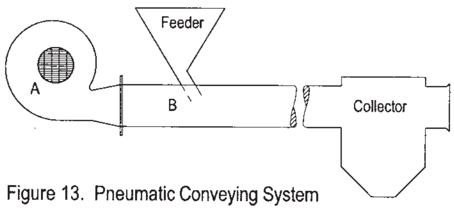


Figure 13. Pneumatic Conveying System

Example 3

Figure 14 shows a pressure blower used on the suction side for pneumatic conveying.

Required: 8000 m^3/hr of standard air based on weight per kg of material to be moved, and -10 kPa S.P. at point C.

What actually happens in the system?

1. Air enters system at A at 20˚C and 101.3 kPa atmospheric pressure.
2. At point C, pressure reading required is –10 kPa gauge pressure or 101.3 kPa –10 kPa = 91.3 kPa absolute pressure.Temperature change can be ignored as change is gradual through the system.
3. Density ratio = 91.3/101.3 = 0.90
   At point C, density = 1.2 kg/m^3 x 0.90 = 1.08 kg/m^3
   Capacity = 8000 m^3/hr / 0.90 = 8889 m^3/hr
4. Required: 8889 m^3/hr at 10 kPa S.P. at 1.08 kg/m^3 density.
   Corrected pressure = 10 / 0.90 = 11.1 kPa.

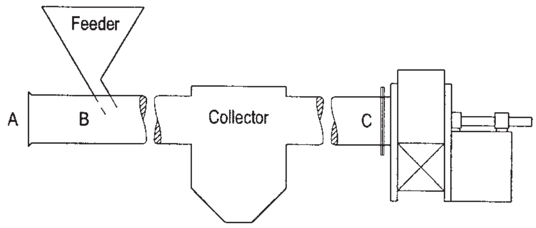


Figure 14. Pull-Through Pneumatic Conveying System

* Select fan for 8889 m^3/hr @ 11.1 kPa S.P. standard air. The actual operating S.P. and absorbed power should be multiplied by 0.90.

13. FAN LAWS

Suppose a fan of a certain size and speed has been tested and its performance has been plotted for the standard air density of 1.2 kg/m^3 (0.075 lb/ft3). We then can compute the performance of another fan of geometric similarity by converting the performance data in accordance with these fan laws without running a test on this second fan (beware of size effect where there is a minor increase in efficiency for larger sizes). We call them general fan laws because they apply to any type of fan: axial-flow, centrifugal, and mixed-flow fans, roof ventilators, cross-flow blowers, and vortex blowers.

13.1 Variation In Fan Speed

The air volume (Q) varies directly with the speed:

$$\frac{Q_2}{Q_1} = \left(\frac{rpm_2}{rpm_1}\right)$$

The static pressure (SP) varies as the square of the speed:

$$\frac{SP_2}{SP_1} = \left(\frac{rpm_2}{rpm_1}\right)^2$$

The absorbed power varies as the cube of the speed:

$$\frac{kw_2}{kw_1} = \left(\frac{rpm_2}{rpm_1}\right)^3$$

The efficiency remains constant but, of course, shifts to the new air volume values.
The sound power level is increased (or decreased) by 50 times the logarithm (base 10) of the speed ratio:

$$Lw_2 - Lw_1 = 50 \log_{10}\left(\frac{rpm_2}{rpm_1}\right)$$

Increasing the fan speed results in a steeper static pressure curve. In other words, an increased speed boosts the static pressure more than the air volume.

13.2 Variation In Fan Size

Another important fan law concerns the conversion of fan performance if the fan size is varied.

The fan laws for size, however, can be used only if the two fans are in geometric proportion. Here is what **geometric proportionality** means:

1. Both fans have the same number of blades.
2. Both fans have the same blade angles and any other angles on the fan wheel and fan housing.
3. If the diameters of the two fan wheels are D1 and D2, for a size ratio D2/D1, all other corresponding dimensions of wheel and housing have the same ratio.

The air volume (Q) varies as the cube of the size:

$$\frac{Q_2}{Q_1} = \left(\frac{D_2}{D_1}\right)^3$$

The pressures vary as the square of the size:

$$\frac{SP_2}{SP_1} = \left(\frac{D_2}{D_1}\right)^2$$

The absorbed power varies as the fifth power of the size:

$$\frac{kw_2}{kw_1} = \left(\frac{D_2}{D_1}\right)^5$$

The efficiency remains almost constant. There is a minor increase in efficiency for larger sizes. For a size ratio of 1.5, the maximum efficiency will increase by less than 1 percent. This is called the **size effect.**

The sound power level is increased (or decreased) by 70 times the logarithm (base 10) of the size ratio:

$$Lw_2 - Lw_1 = 70 \log_{10}\left(\frac{D_2}{D_1}\right)$$

The geometrically similar, larger fan results in a flatter static pressure curve. In other words, the increased size boosts the air volume more than the static pressure.

13.3 Variation In Density

This fan law is used when the fan operates at high altitude where the air density is less, where the fan handles hot or cold air (the air density is inversely proportional to the absolute temperature), or where the fan handles a gas other than air, while the size and speed of the fan remain constant. Our conversion rules then will read as follows:

The air volume remains constant: $\frac{Q_2}{Q_1} = 1$

The pressure varies directly as the density r: $\frac{SP_2}{SP_1} = \frac{r_2}{r_1}$

The absorbed power varies directly as the density r: $\frac{kw_2}{kw_1} = \frac{r_2}{r_1}$

The sound power level is increased (or decreased) by 55 times the logarithm (base 10) of the density ratio

$$Lw_2 - Lw_1 = 55 \log_{10}\left(\frac{r_2}{r_1}\right)$$

The efficiency remains constant.

13.4 Variation In Fan Size, Fan Speed, And Density

If the fan size (D), fan speed (rpm) and density (r) are varied, the three sets of rules discussed above can be applied. The combined rules then will read as follows:

$$\frac{Q_2}{Q_1} = \left(\frac{D_2}{D_1}\right)^3 \times \left(\frac{rpm_2}{rpm_1}\right)$$

$$\frac{SP_2}{SP_1} = \left(\frac{D_2}{D_1}\right)^2 \times \left(\frac{rpm_2}{rpm_1}\right)^2 \times \left(\frac{r_2}{r_1}\right)$$

$$\frac{kw_2}{kw_1} = \left(\frac{D_2}{D_1}\right)^5 \times \left(\frac{rpm_2}{rpm_1}\right)^3 \times \left(\frac{r_2}{r_1}\right)$$

$$Lw_2 - Lw_1 = 70 \log_{10}\left(\frac{D_2}{D_1}\right) + 50 \log_{10}\left(\frac{rpm_2}{rpm_1}\right) + 55 \log_{10}\left(\frac{r_2}{r_1}\right)$$

14. FAN TESTING

The Australian Standard for the determination of performance characteristics for Industrial Fans known as the SAA FAN TEST CODE is AS2936-1987. It is published by the Standards Association of Australia.

Air is a difficult item to measure. It cannot be seen, nor can it be caught in a container and weighed. Furthermore, its motion is 3 dimensional and turbulent. Therefore many measurements of velocity and direction would be needed to integrate a total volume flow.

14.1 Measuring Pressure & Velocity

For most industrial applications, the only air measurements needed are those of static pressure, total pressure and temperature. With these, air velocity and volume can be quickly calculated. Volume of air flowing past a point in a duct (m^3/s or cfm) is determined by multiplying air velocity (m/s or fpm) by the cross sectional area of a duct (m^2 or ft^2).

By far, the best method of measuring either air velocity or static pressure is with the pitot tube and manometer. Figure 15 shows the cross section of a pitot tube where total pressure and static pressure are measured. Figure 16 shows the pitot tube connected to a U-tube manometer. Lack of moving parts and fundamental simplicity make this set of instruments inexpensive and accurate.

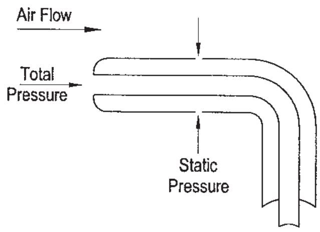


Figure 15. Pitot Tube Cross Section

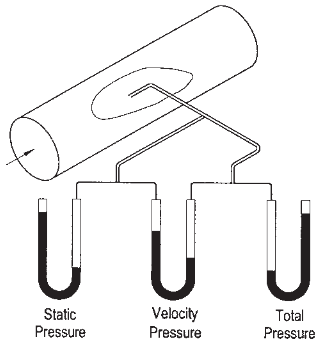


Figure 16. Pitot Tube In Outlet Duct

The duct diameter should be at least 30 times the diameter of the pitot tube. The 5/16" (8mm) diameter standard pitot tube should be used for ducts above 250mm diameter. The 1/8" (3.2mm) diameter pocket pitot tube can be used for ducts above 100mm.

In small ducts or where traverse operations are impossible, an accuracy of ± 5% can frequently be achieved by placing pitot tube in the centre of duct. The center velocity reading should be multiplied by 0.9 for an approximate average.

Manometers have a liquid column and can be roughly classified according to the pressure to be measured as follows:

For less than 100 Pa	:	a micromanometer
For 100 to 1000 Pa	:	an inclined-tube manometer
For 1000 to 15,000 Pa	:	a vertical, or U-tube manometer
For 15,000 to 110,000 Pa	:	a mercury manometer or barometer

The "U" tube is probably the most common of manometers. The bore of the glass or plastic tube should be 3/16 to 1/4 inch (5 to 6 mm) in diameter and the walls perfectly clean.
For inclined tube manometers, the multiplying factor is the sine of the angle of inclination to the horizontal. In some models, the measuring scale may be drawn to allow for direct reading by allowing for the angular setting and the density of the fluid. For inclined manometers, mineral oil, kerosene or other liquid may be used in place of water to extend the scale and flatten the meniscus.

14.2 Vertical Height of the Liquid Column

Force = mass x acceleration = kgm/s^2 = 1N
Pressure = Force per unit area = N/m^2 = kg/ms^2 = 1 Pa

If the vertical height of the liquid column in the manometer is h, then,

$$\text{Pressure (Pa)} = rgh = \frac{kg}{m^3} \times \frac{m}{s^2} \times m = \frac{kg}{ms^2} \text{ where}$$

r= density of the liquid
g = gravitational acceleration = 9.81 m/s^2

If the liquid used is water, then r = 1000 kg/m^3, for h = 1m

$$\text{Pressure} = 1000 \frac{kg}{m^3} \times 9.81 \frac{m}{s^2} \times 1m = 9810 \text{ Pa}$$

In other words, 1 meter height of water column corresponds to 9810 Pa. 1mm water column corresponds to 9.81 Pa. One inch water column corresponds to (25.4 x 9.81) = 249 Pa = 1 inch W.G.

When filled with water, it gives direct reading in inches W.G. by the difference in levels of the two sides.

14.3 Laminar And Turbulent Flow

Airflow in industrial ducts is almost always turbulent, with a small boundary layer at the surface of the duct. Figure 17 below shows typical velocity profiles in ductwork.

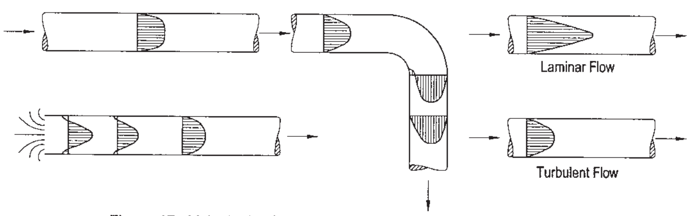


Figure 17. Velocity Profiles In Ducts

Laminar flow is airflow in which air molecules travel parallel to all other molecules and is characterised by the absence of turbulence. Turbulent flow is airflow characterised by transverse velocity components as well as velocity in the primary direction of flow in a duct.

Since velocity varies with distance from the edge of the duct, a single measurement is not usually sufficient. However, if the measurement is taken in a straight length of round duct, 3-6 diameters downstream and 2-3 diameters upstream from obstructions or directional changes, then the average velocity can be estimated at 0.9 of the centerline velocity. (The average velocity pressure is about 0.81 of center line velocity pressure.)

A more accurate method is the pitot traverse. In a pitot traverse, six or ten VP measurements are made on each of two traverses across the duct, 90 degrees opposed. Measurements are made in the center of concentric circles of equal area.

14.4 Laboratory Fan Test

Laboratory test installations are designed to give smooth air flow to and from the fan (figure 18). Inlet ducts are of the same area as the fan inlets and 10 diameters long. Outlet ducts have the same area as the fan outlets and are also straight for a distance of 10 diameters. Both ducts have straighteners before the point of reading. This set-up avoids turbulence and spinning flow in the ducts and entering the fan. Such installations are seldom found in actual practice and the greater the deviation, the greater the error to be expected.

Figure 19 shows the flow and pressure measurement set up at the fan outlet as per Australian Standard AS 2936-1987.

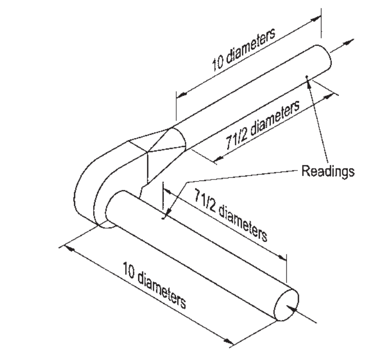


Figure 18. Laboratory Test Installation Designed For Smooth Air Flow

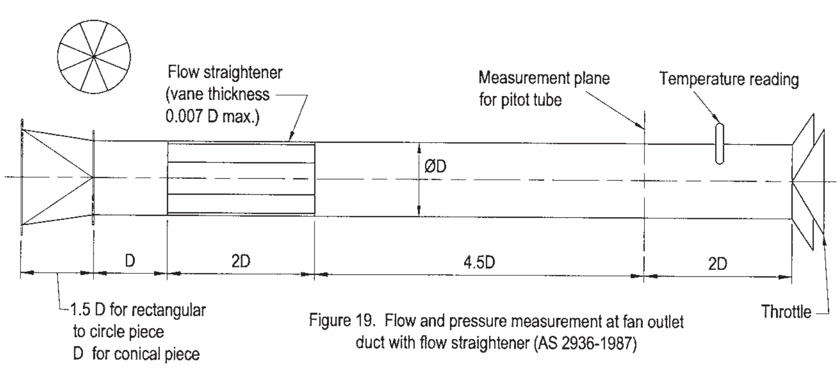


Figure 19. Flow and pressure measurement at fan outlet duct with flow straightener (AS 2936-1987)

14.5 Anemometer

The rotating vane anemometer can be used to determine airflow through large supply and exhaust openings. The cross sectional area of the instrument should not exceed 5% of the cross sectional area of the duct. The anemometer is relatively accurate when measuring a constant volume of air that flows straight through it at a uniform velocity perpendicular to its face. It can be quite misleading if used in non-uniform velocities such as discharges from a grille or where air approaches from an angle.

14.6 Aneroid Gauges

The best known of this type of gauge is the Magnehelic gauge, which is suitable for the measurement of low pressures encountered in the field.

The principal advantages of this gauge can be listed as follows: easy to read, greater response than manometer types; very portable - small physical size and weight; absence of fluid means less maintenance; and mounting and use in any position without the loss of accuracy. Principal disadvantages are that the gauge is subject to mechanical failure, requires periodic calibration checks, and occasional recalibration.

14.7 Measuring Fan Speed

Fan performance is quite sensitive to variations in fan speed. Rotational speed can be measured with various types of tachometers, or slip counting with stroboscopic light for speeds close to synchronous speeds.

14.8 Barometer

A barometer is required to measure the barometric pressure to calculate the air density. Mercury barometers will give the most accurate readings. It is customary to assume that the barometric pressure is standard, 29.92 inches of mercury.

14.9 Temperature

A psychrometer consisting of two thermometers is used to measure the ambient dry-bulb and wet-bulb temperatures.

15. FAN CONSTRUCTION

15.1 Fan Arrangement, Rotation And Motor Positions
Fans can be built with variations in rotation, arrangement, and motor position. Table 2 on page 118 outlines these variations as per AMCA Standard 99-86. Fans can also be built in various class of construction, bearing type, to handle air at elevated temperatures, with spark resistant construction and to resist the effects of corrosive air or gas by using protective coatings or by using special metals.

15.2 Spark Resistant Fans
Table 3 below shows the AMCA Standard Classification for the construction of Spark Resistant Fans:

Table 3 : Types of Spark Resistant Fan Construction

Type A	All parts of the fan in contact with the air or gas being handled shall be made of non-ferrous material.
Type B	The fan shall have an entirely non-ferrous wheel and non-ferrous ring about the opening through which the shaft passes.
Type C	The fan shall be so constructed that a shift of the wheel or shaft will not permit two ferrous parts of the fan to rub or strike. It is common to fit a non-ferrous ring on the fan inlet cone.

Notes:
1. Bearings shall not be placed in the air or gas stream.
2. The user shall electrically ground all fan parts.
3. Explosion proof motors and static resistant belts may also be required by the application.

15.3 High Temperature Fans
The correct fan arrangement, special construction, and limitations placed upon the maximum operating speeds, are important considerations that must be taken into account when elevated temperatures are involved.

The rates of expansion of the wheel hub and shaft must be carefully reviewed to insure continued trouble-free operation.

When operating at elevated temperatures, the maximum allowable fan speed must be reduced.

Most metals become characteristically weaker at high temperatures. This weakness is measurable in terms of yield strength, and modulus of elasticity and can be translated into formulae that accurately determine the safe speed of a wheel and shaft assembly in relation to its tested maximum speed at standard conditions. Below -30˚C, ordinary mild steel can be brittle.

15.3.1 High Temperature Construction (As a Guide)
Table 4 below outlines the construction features for high temperature centrifugal fans.

Arrangement	Features	Maximum Temperature
4 & 4F	Motor shaft exposed to airstream	65°C
2 & 3	Bearing housing exposed to airstream	65°C
1, 8 & 9	Fan Shaft exposed to airstream	120°C
1/ID1 8/ID1 9/ID1	Bearing pedestal cut away to fit aluminium split type cooling fin to the fan shaft between casing and inboard bearing	250°C
1/ID2 8/ID2 9/ID2	ID1 features with deflector plate inserted between casing and cooling fin	325°C
1/ID3 8/ID3 9/ID3	ID2 features with bearing pedestal separated from fan housing and bolted on to a combined base. Backward withdrawal of bearing pedestal and wheel without disturbing the inlet and outlet connections is available as an option.	400°C

Table 4 : High Temperature Construction For Centrifugal Fans

Centrifugal fan impellers for operation above 400˚C have to be constructed using special material such as stainless steel, incolloy, etc. This is dependant on the operating temperature and speed of the fan.

Table 5 below outlines the construction features for high temperature axial flow fans.

Arrangement	Features	Maximum Temperature
4	Motor exposed to airstream	50°C
2	Bearings exposed to airstream	65°C
2/ID1	Bearings vented to atmosphere. Cooling fin fitted to shaft to draw air over bearings through the belt tunnel and breather pipe	180°C

Table 5 : High Temperature Construction For Axial Flow Fans

Axial flow impellers - polypropylene impellers can be used up to 65˚C maximum temperature. Nylon impellers with aluminium hub can go up to 100˚C. Aluminium impeller maximum operating temperature is 160˚C. For temperatures above 160˚C, steel impellers should be used.

16. ACCESSORIES FOR FANS

16.1 Access Doors
Access doors, also called clean-out doors, should be used whenever there is a possibility of dirt collecting inside the fan. If inlet and outlet are inaccessible without removing ductwork, it is wise to purchase a door; even if no dirt build-up is anticipated. An unanticipated need to inspect the fan's interior will then not pose a difficult problem.

16.2 Flanges
Flanged inlets or outlets are usually used on ventilating fans when ducts are not fastened to the fans with canvas sleeves and wrap around slip connectors.

16.3 Drain Plug
A drain plug should be specified wherever rain, condensation, or washing water can get into the fan casing.

16.4 Shaft Seal
Shaft seals are useful in reducing the hissing noise emitted between the fan casing and drive shaft. Shaft seals are also advisable when the fan is to handle abrasive material that can be blown out of the shaft hole, into the bearings. The seal should be of the lubricated labyrinth type.

16.5 Cooling Fin
Cooling fin and guards should be specified whenever the temperature inside the fan exceeds 150˚C. The cooling fin extracts heat from the shaft and prevents it from harming the bearings. It should be covered with a finger proof guard. Aluminium or brass cooling fins are preferred to cast iron or steel ones because of their higher heat transfer coefficients. Cooling fin should be shrunk or clamped on the shaft to assure solid heat transfer bonds. Since cooling fins fling away any dirt coming out through the housing shaft hole, they also serve to protect the bearing from abrasion.

16.6 Dampers
Dampers can be applied to either fan inlets or outlets. If long periods of reduced-capacity operation are needed, along with the capability of opening up the control for other periods of maximum capacity, then the inlet damper is advisable. It will reduce horsepower and electrical power consumption.

If dampers are to be used to reduce capacity severely for short periods to allow for small adjustments in capacity, as perhaps when the resistance of filters or coils change, then outlet dampers should be used.

All too often, dampers are set in one position and never changed. In these cases the fan should be slowed down to deliver the desired capacity without dampering.

16.7 Vibration Isolators
The materials usually used for anti-vibration mounts are steel springs and rubber-in-shear.

Vibration isolation bases are becoming increasingly popular on ventilating jobs. Their function is to prevent the several sources of mechanical and electrical noise and vibration created in a fan assembly from being transmitted to the building structure. Isolation bases should always be accompanied by flexible duct connections.

Isolation bases are not intended to reduce air-borne or air-created noise. However, they do break up the sounding-board effect, thereby contributing to the reduction of noise levels. In critical instances, silencers that match the fan aerodynamics may be desirable.

16.8 Inlet Box
An inlet box is a device, designed and fitted to ensure optimum inlet air flow characteristics, particularly where the air stream must be turned sharply through a substantial angle.

Inlet box dampers may be used to control the air flow volume through the system. Either parallel or opposed blade types may be used. The parallel blade type is installed with the blades parallel to the fan shaft so that, in a partially closed position, a forced inlet vortex will be generated. The effect on the fan characteristics will be similar to that of inlet vane control. The opposed blade type is used to control air flow by changing the duct system resistance.

16.9 Variable Inlet Vanes
Variable inlet vanes (VIV) helps to maintain the fan efficiency at reduced flow conditions by controlling the air flow quantity at the fan inlet. These are arranged to generate a forced inlet vortex which rotates in the same direction as the fan impeller (pre-rotation). Inlet vane arrangements may be of two different basic types: (1) Integral (built in) or (2) Cylindrical (add on).

17. CONTROL OF AIRFLOW

It is frequently necessary to control the rate at which air is moved through a system. This may be a once only adjustment to suit actual operating conditions, an occasional control to give, for example, a summer and winter condition, or a continuously variable adjustment to maintain an environment using variation in air flow for control or to satisfy a process.

The rate of flow of air through a system is determined at the intersection of the system resistance curve and the fan performance characteristic curve.

Control may be achieved either by changing the effective resistance of the system, or by altering the performance of the fan. The method chosen will depend largely upon the changed running costs at the changed flow rates, weighed against the capital cost of the changed system or fan and drive. In some situations noise may also be a factor.

17.1 Control by Varying System Resistance
The simplest means of flow adjustment is by use of a damper at a suitable point in the ducting system.
Closing the damper will increase the resistance to flow and the quantity of air will fall as dictated by the fan characteristic curve. Dampers can be operated manually or by an automatic control system.

17.1.1 Parallel Blade Outlet Dampers
These dampers are the simplest and most economical to manufacture. The outlet damper control arm swings through a relatively large arc to reduce fan capacity a small amount from its wide open value. This makes the outlet damper particularly useful when installed on a continuous process manufacturing system where sensitive control of air volumes between wide open and 70% or 80% of wide open is desired. The large control arm swing means predetermined settings may be repeated with good accuracy.

Since an outlet damper operates by adding resistance to a system, its only effect upon fan absorbed power is to move the point of operation further to the left on the fan curve. The dampered power may be less, the same, or more than the wide open absorbed power, depending upon the shape of the fan absorbed power curve and where the fan operates on the fan curve.
Generally the closing of an outlet damper results in a reduction of power.
The airflow at the discharge is thrown to one side of the fan (figure 20).

17.1.2 Opposed Blade Outlet Dampers
Opposed blade outlet dampers are used where a straight line relationship between fan volume and control arm swing is desired. Alternate blades are made to turn in opposite directions. The resulting change in air delivery is almost exactly proportional to the amount of control arm swing. Opposed blade dampers cost slightly higher to manufacture than parallel blade dampers. Opposed blade dampers are used when it is necessary to maintain an even distribution of air immediately downstream from the damper (figure 21).

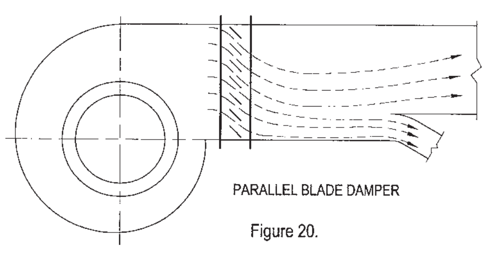


Figure 20.

Although cheap to install, the inherent pressure loss across an outlet damper is a waste of energy and may create noise. A more efficient form of control may be effected by adjusting the performance of the fan itself.

17.2 Control by Varying Fan Performance
There are several means by which the performance of a fan may be controlled. They divide conveniently into those which are continuously variable and those which are adjusted occasionally.

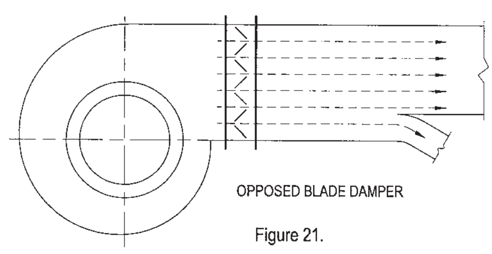


Figure 21.

17.2.1 Continuously Variable Controls

17.2.1.1 Speed Control
One of the most efficient methods of controlling the performance of a fan is by varying its running speed. When working in a constant duct system, the point of operation will move down the duct system curve as the speed is varied. This has the advantage of maintaining the fan's efficiency, resulting in a corresponding maximum drop in power consumption and noise level as the speed is reduced.

The essence of this method is, therefore, the provision of a suitable means of controlling the speed of the fan. This may be achieved either by varying the speed of the prime mover or by changing the ratio of the drive. Examples of speed control methods include: Multispeed or continuously variable speed electric motors, Diesel or petrol engine drive, Hydraulic motor, Variable speed gearbox, Fluid coupling, Magnetic coupling, Variable ratio pulleys and belt drive.

17.2.1.2 Inlet Vane Control
A second method of continuous control of fan performance is by the introduction of specially designed adjustable vanes into the airstream entering the fan inlet so as to generate a swirl of air in the direction of the impeller rotation. This produces a reduction in the performance capability of the fan.
Note however, that there is an angle of the blades beyond which swirl will become ineffective and throttling will occur with a resultant uneven change to the systems performance.
Power reduction with inlet vane control is usually much less than with fan speed control.

17.2.1.3 Blade Angle Control
A further very efficient method, at present confined to axial fans, is the adjustment of blade pitch angle while the fan is running. In this method the angles of all the blades are altered simultaneously by a suitable mechanism. Blade angle control progressively reduces pressure/volume and power curves with a very substantial power reduction.

17.2.2 Pulley Change
Where the adjustment of performance is required once only, or at infrequent intervals, this can be most easily obtained by an alteration to the fan (or drive) itself. This is particularly useful as a means of adjusting the performance of the fan at the installation stage to suit actual on-site operating conditions. In belt driven fans the speed of the fan can be adjusted by changing one or both of the drive pulleys (very often belt driven fans are chosen just for this facility).

17.3 Factors Affecting Choice of Control Method
The choice of method is one of assessing the change in power costs over the life of the machine against the initial and maintenance costs, bearing in mind the degree of control required and the frequency of its operation. The following are some of the factors which can influence the choice: Initial manufacturing and installation costs, Maintenance and replacement costs, Power costs, Degree of control required, stepped or continuous, Accuracy and repeatability of control settings, Range of flow over which control is required, Temperature, toxicity or corrosiveness of gas handled, Period of time over which each setting is effective, The control system (if automatic operation is necessary), and Noise levels.

18. FANS IN SERIES AND PARALLEL

18.1 Fans In Series
Fans in series must all handle the same amount of gas by weight measurements, assuming no losses or gains between stages. The combined total pressure will be the sum of the individual fan total pressures. The velocity pressure of the combination can be defined as the pressure corresponding to the velocity through the outlet of the last stage. The static pressure for the combination is the difference between its total and velocity pressures and is therefore not equal to the sum of the individual fan static pressures.

If one fan discharges directly into the inlet of the second fan, the velocity pressure of the first fan is lost. Practically, the losses are greater then this because the non-uniform flow into the second stage fan will create even greater losses.

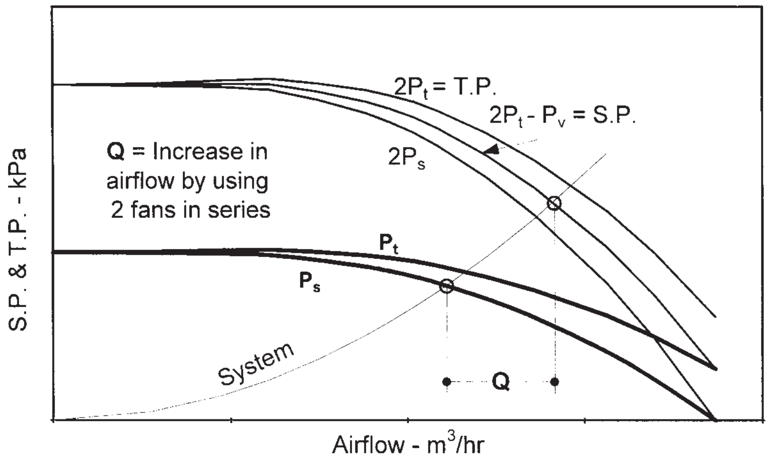


Figure 22. Identical fans operated in series

Air volume through each fan will be the same if air is considered incompressible. To establish combined fan curve, the combined total pressure is the sum of individual fan pressures at equal air volumes.

Consider the simplest case of two identical low pressure fans in series as represented by their characteristic curves in Figure 22 on page 92. Neglecting connection losses between the fans and assuming the inlets and outlets to be of the same size, the total pressure of the series is $2P_t$ while the static pressure is $2P_t - P_v$ or, which is the same in this case, $2P_s + P_v$. It will be seen that the new static pressure of the system is slightly greater than the sum of the individual static pressures. This becomes of less importance where the inlet and outlet velocity pressures are low relative to the static pressure of the fan. Actually when the resistance of the connecting ducts and elbows are taken into consideration, the combined static pressure will be less than the amount shown and may be less than the sum of the two individual static pressures.

If the fans are of the high pressure type then the air entering the second fan will have a greater density than that of the first and will produce a correspondingly greater pressure rise. Thus, if a fan is designed to produce a static pressure rise of 9 kPa with atmospheric pressure on the inlet and the actual static pressure on the inlet is 10 kPa due to another fan in series with it, then the static pressure rise of the second fan will be

$$\frac{(101.325 + 10)}{101.325} \times 9 = 9.89 \text{ kPa}$$

If the gas is considered compressible, the volumetric capacity of the second stage will not equal the volumetric capacity of the first stage. The individual total pressure must be chosen accordingly at the same volume before they are combined.

Multistage blowers are, in effect, two or more fans in series in the same casing. Fans may also be in series but at opposite ends of the system.

Series operation can be used as a method of controlling the flow through a system by shutting down fans as appropriate, but the resistance to flow of those fans not being driven should be allowed for in the calculations.

Fans are usually placed in series when the pressure required is greater than can be obtained with a single fan or when one fan can act as a pressure booster under varying system resistance.

18.2 Fans In Parallel

When two or more fans receive air from and deliver into a common system, they are said to operate in parallel. Identical fans may be operated in parallel when two such fans will deliver twice the volume of air at the same pressure as a single unit (figure 23).

Non-identical fans, too, may be operated in parallel, but care must be taken to select a good working position on the combined characteristic and even then maximum efficiency is unlikely to be achieved at the same time by each fan.

Care must be taken in selection of fans for parallel operation to avoid the possibility of stalling.

Two fans operating on the same system, it should be noted, do not give twice as much air as one of them would give when working alone on the system. As the resistance of the system usually increases as the square of the volume flow of air, the latter settles down at some value which is less that twice the volume given by one fan.

A form of volume control is feasible by switching off one or more units, but generally it will be necessary to provide anti-backdraught devices to prevent short circuiting of the air back through the fans not in use.

Fans are usually operated in parallel when lack of space forbids erection of a single large fan. Sometimes, too, a number of small fans may be installed at a lower capital expenditure than a single unit capable of the combined duty. The risk of complete shut down is minimised as individual fans may be taken out of service for maintenance without closing down the system.

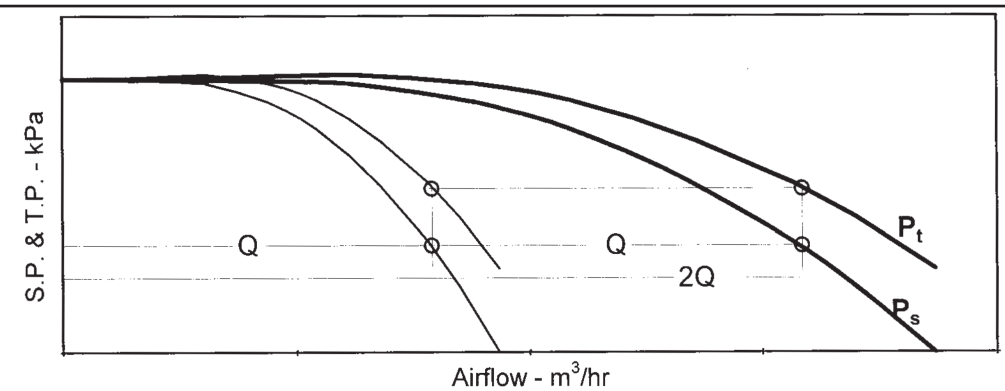


Figure 23. Identical fans operated in parallel

A noise problem often encountered with fans operating in parallel is beating. This is caused by a slight difference in speed of rotation of the two theoretically identical fans. The resulting low frequency beating noise can be very annoying and difficult to eliminate.

The question may arise, therefore, when should fans be placed in series and when in parallel. This will be governed by the relative point of operation on the curve as well as the application of the fan.

19. EFFECT OF SYSTEM DESIGN ON FANS (SYSTEM EFFECT)

By system is meant the path through which air is pushed and/or pulled. Fan performance ratings are based on laboratory tests made under ideal conditions. Actual fan installations can be expected to deliver the fan's full rating only when the actual system allows the fan to operate as it would in a laboratory test. Any deviation will cause a change, invariably to the detriment of the system's performance in actual installations.

19.1 Important Causes of Poor Fan Performance

Fan rating tests are made under conditions that effectively eliminate the three most important causes of poor fan performance in the field. These are:

(1) Eccentric flow into the fan inlet.
(2) Spinning flow into the fan inlet.
(3) Fan discharge ductwork that does not allow full development of fan pressure.

Figure 24 shows an even air distribution from a straight length of duct at the fan inlet.
Figure 25 shows an eccentric air distribution from an elbow at the fan inlet.
Figure 26 shows examples of inlet duct connections that spin the air into the impeller of the fan.
Figure 27 shows the use of guide vanes to counteract the spinning air at the fan inlet.

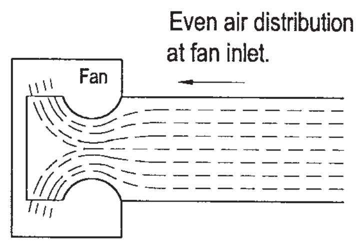


Figure 24.

Air should be sucked into a fan inlet with the same velocity at all points in the inlet. This allows all portions of the fan wheel to do equal portions of the fan's work. If the air is caused to have a higher velocity through one part of the inlet, work is unevenly distributed, and the amount of work done will be diminished.

Where an elbow on a fan inlet causes the air to load one side of the inlet, total air flow would probably be reduced 5% to 10% below what it would be with even distribution.

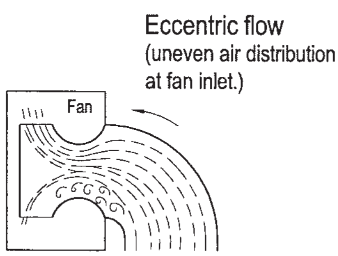


Figure 25.

Fans perform correctly when air flows straight into the inlet. That direction the air is flowing when it meets the fan wheel is quite important because the wheel works on the air by changing its direction and velocity. If air enters the fan inlet with a spinning (vortex) motion, capacity of the fan will be lowered.

A 30° spin will reduce air flow 15%, and a 45° spin, 25%. Spinning is often caused by inlet boxes. An ideal fan inlet connection creates neither eccentric nor spinning flow. Such inlet could be a long straight duct containing straightening vanes. However, it is usually necessary to adapt the system to the available space. When this is necessary, the system designer has a choice of either:

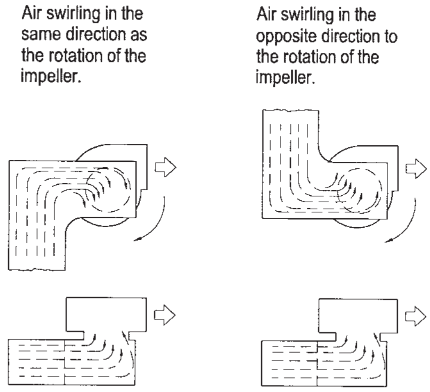


Figure 26. Examples of duct connections which give rise to air spinning at the fan inlet.

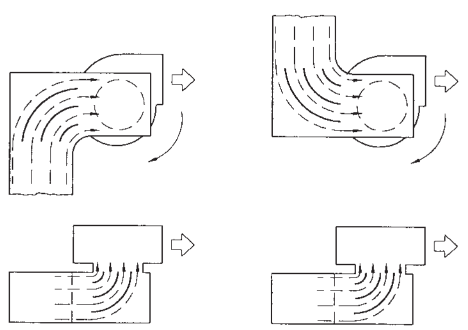


Figure 27. Guide vanes which counteract spinning at the fan inlet.

(1) Putting corrective devices in the duct to prevent eccentric flow, or
(2) Letting the eccentric and/or spinning flow go unchecked but compensating for them in selection of the fan.

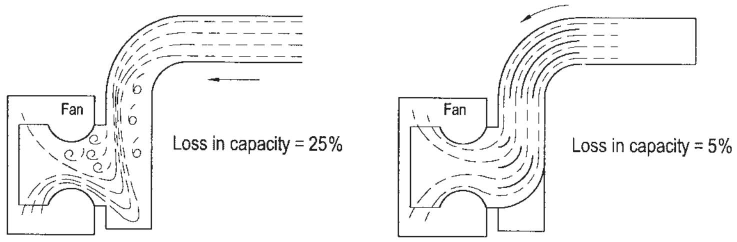


Figure 28. Inlet box causing 25% loss due to eccentric flow, and use of vanes to remedy condition.

The first choice is preferable, but the second is often necessary, particularlyif the air is carrying material or the duct needs to be clear of obstructions for maintenance.

Spinning occurs principally from one of two reasons. Either air makes two consecutive turns in perpendicular planes that form a corkscrew path, or air is introduced to the duct or plenum tangentially.

If the air spins in the same direction as the wheel is rotating, the net effect is similar to slowing down the fan. This is the same effect as is obtained with an inlet damper. Both capacity and absorbed power are reduced. If air spins in a direction opposite to wheel rotation, the net effect is usually reduced fan capacity, and always increased power.
Figure 28 shows the effect of inlet boxes reducing the fan capacity and the utilisation of inlet vanes to remedy the situation.

The connection made to a fan's outlet cannot affect its performance as much as can the inlet connection, as the fan has done its work on the air by the time it reaches the outlet. However, the discharge connection does affect fan performance, and its design is important.

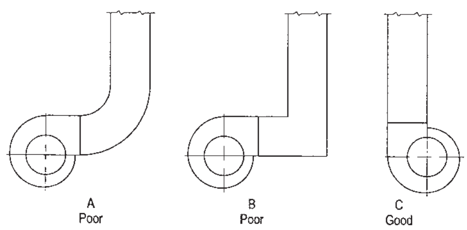
Air is not discharged from a fan with uniform velocity across the fan outlet. This is because the air has weight and is thrown to the outside of the scroll. The velocity becomes substantially uniform at about 2.5 times nominal duct diameter down a straight duct for velocities up to 12 m/s (2400 fpm) and additional 1 duct diameter for every additional 5 m/s (1000 fpm). If the duct is rectangular with sides a and b, the equivalent duct diameter is 2ab/(a+b).

Velocity pressure is proportional to velocity squared, the summation of velocity pressures at the fan discharge is higher than the same summation performed several diameters down the duct. Since total pressure is the same at both points, neglecting the small friction drop, the static pressure should be higher down the duct than at the fan. Tests bear this out.

If the discharge duct is omitted, a loss equal to roughly half the average fan outlet velocity pressure occurs, and a system resistance calculation should include this pressure loss as additional required static pressure.

Outlet connections shown in Figure 29(A) and Figure 29(B), though often seen, are poor. Imagine the path of the high velocity air leaving the fan scroll. A connection of this type should be considered to have a loss equivalent to its entire fan outlet velocity pressure. Arrangement shown in Figure 29(C) is good.

Figure 29.



19.2 Static Regain Using Outlet Evase'

Static regain at fan outlets is often wastefully overlooked, especially when the fan is discharging to the atmosphere. (Also refer to section 2.6 on page 74)

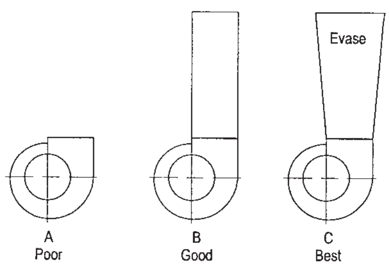
Referring to figure 30 , if each fan is to have a 1000 Pa inlet pressure and have an average outlet velocity V_1 of 15 m/s, then for:

1. Fan A without any outlet duct,
   Outlet loss = 0.5(VP) = $0.5(0.5rV_1^2)$ = 0.5 x 0.5 x 1.2 x 15^2 = 68 Pa
   Select fan A for (1000 + 68) = 1068 Pa

2. Fan B with straight length of outlet duct,
   Outlet loss = Friction loss of outlet duct
   Select fan B for 1000 Pa + Friction loss of outlet duct

3. Fan C with evase', if outlet velocity V_2 at the end of evase' = 5 m/s, then
   Static regain = $(0.5rV_1^2) - (0.5rV_2^2) = (0.5 \times 1.2 \times 15^2) - (0.5 \times 1.2 \times 5^2)$
   = 120 Pa
   Select fan C for (1000 - 120) = 880 Pa + Friction loss in evase'

Figure 30.



19.3 Other Major Causes of Poor Fan Performance

Other major causes of deficient performance are:

- The air performance characteristics of the installed system are significantly different from the system design engineer's intent.
- The system design calculations do not include adequate allowances for the effect of accessories and fittings in the system or the fan selection was made without due allowance for fan accessories.
- The "performance" of the system is determined by inappropriate or inaccurate field measurement techniques.
- Maintenance of filters, coils etc is neglected. Dirty filters, dirty ducts, and dirty coils will increase the system resistance and consequently reduce the air flow, often significantly.

19.4 Precautions To Prevent Poor Fan Performance
Use appropriate allowances in the design calculations when space or other factors dictate the use of less than optimum arrangement of the fan outlet and inlet connections.

Design the connections between the fan and the system to provide, as nearly as possible, uniform straight flow conditions at the fan outlet and inlet.

Include adequate allowances for the effect of all accessories and fittings on the performance of the system and fan.

19.5 The 'System Effect'
Point 1 in Figure 31 depicts the calculated design point with the system pressure losses accurately determined and a suitable fan selected for operation at that point. If, however, no allowance has been made for the effect of the system connections on the fan's performance the actual duct system curve may cut the fan curve at Point 4. To obtain the design air flow the fan should then be selected to operate at Point 2. To compensate for this 'system effect' it will be necessary to include additional losses in the calculated system pressure loss to determine the total system curve.

If the actual system pressure loss is greater than design, an increase in fan speed may be necessary to achieve the design volume flow rate at Point 2. Before attempting to increase fan speed, a check should be made with the fan manufacturer to determine if the speed can be safely increased and also to determine the expected increase in power. The connected motor may not handle the increased fan power required.

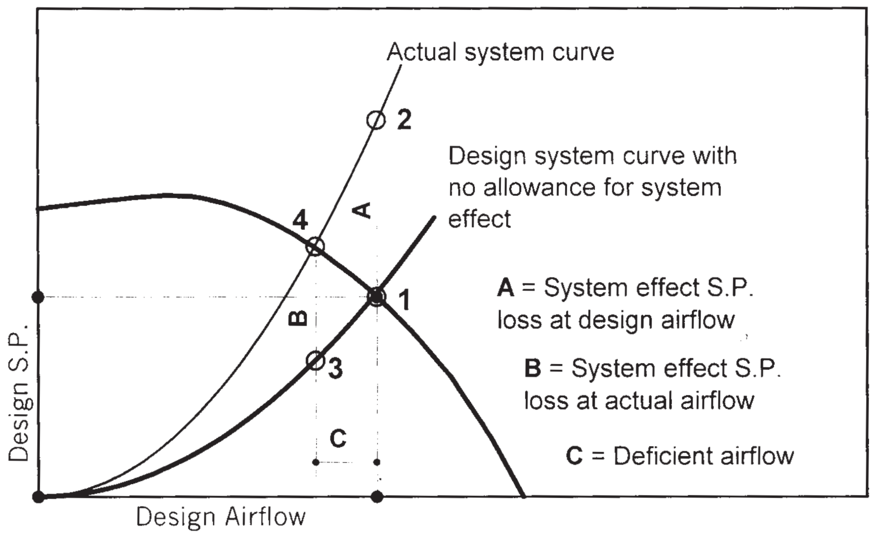


Figure 31. Performance loss due to system effect

20. DUCT DESIGN
If the duct system conforms to recommended design practice with special attention to those areas where turbulence is likely, both aerodynamic and acoustic efficiencies will improve.

20.1 Hood Coefficient Of Entry (C_e)
The hood coefficient of entry (C_e) is a measure of the ratio of actual flow to ideal flow. C_e is defined as the square root of the ratio of duct velocity pressure (VP) to hood static suction, which is the static suction in the duct near the entrance (SP)

$C_e = \sqrt{(VP/SP)}$

If there were no losses, then SP = VP and $C_e = 1.0$
However, hoods always have some losses, and C_e is always less than 1.0. Refer Figure 33 on page 97.

The velocity of air entering an orifice(duct) = C_e x Velocity corresponding to static pressure in pipe if completely converted to velocity pressure.

20.2 Hood Entry Loss (H_e)

The hood entry loss (H_e) is the static pressure loss when air enters a duct through a hood. The majority of the loss is usually associated with a vena contracta formed in the duct (figure 32). The vena contracta is a layer of turbulent air just inside a duct created by the entry of air into the duct. The narrowed opening of the vena contracta results in a higher velocity pressure and a subsequent loss of static pressure.

20.3 Usefulness Of C_e And H_e

Figure 33 below outlines the C_e and H_e for various types of orifices. The hood entry loss (H_e) is usually expressed in percentage of velocity pressure VP.

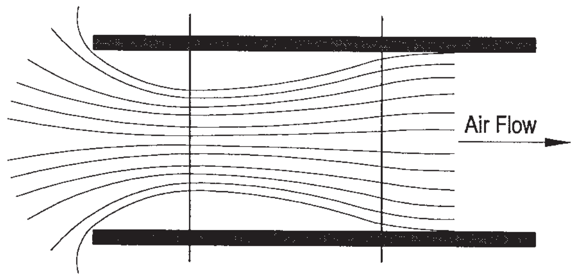


Figure 32. Vena Contracta Airflow

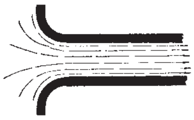
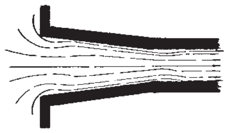
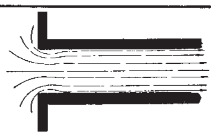
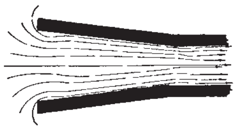
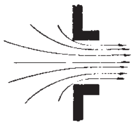
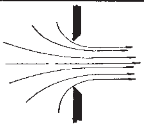
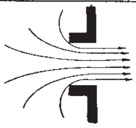
Type of orifice	Description of Orifice	C_e	H_e
	Smooth bell mouthed	0.98	0.04VP
	Flanged cone with 14° included angle	0.94	0.13VP
	Flanged cone with 30° included angle	0.90	0.24VP
	Flanged pipe	0.82	0.49VP
	Unflanged cone with 14° included angle	0.82	0.49VP
	Unflanged cone with 30° included angle	0.79	0.6VP
	Unflanged pipe	0.72	0.93VP
	Short flanged pipe, less than 1.5 diam	0.60	1.78VP
	Thin plate orifice	0.60	1.78VP
	Short re-entrant pipe, less than 1.5 diam.	0.53	2.56VP

Figure 33. Properties of orifices

H_e is useful when it is necessary to find the suction required to produce a given average velocity at the pipe mouth.

C_e is useful when the static suction is known and the pipe velocity is required.

Examples for unflanged pipe, $C_e = 0.72$, $H_e = 0.93VP$

Example 1
Given: Require average pipe velocity, V = 20 m/s
$VP = 0.5 \times 1.2 \times 20^2 = 240$ Pa
$H_e = 0.93 \times 240 = 223$ Pa
Static pressure (suction) required = 240 + 223 = 463 Pa

Alternatively $C_e = \sqrt{(VP/SP)}$, $SP = VP/C_e^2 = (0.5rV^2)/C_e^2 = (0.5 \times 1.2 \times 20^2) / (0.72^2) = 463$ Pa

Example 2
Given: Static pressure (suction) required = 500 Pa
What is the velocity, V in the pipe.
$C_e = \sqrt{(VP/SP)}$, $VP = C_e^2 \times SP = 0.5rV^2$
$V = \sqrt{(C_e^2 \times SP/ 0.5r)} = \sqrt{[(0.72^2 \times 500) / (0.5 \times 1.2)]} = 20.78$ m/s

20.4 Good & Bad Duct Design

Figure 34 shows the dimensions for a good flared (bell mouthed) inlet.
Figure 35 shows the dimensions for a good conical inlet, and the clear space required at the inlet.
Figure 36 shows the design criteria for achieving good circular duct transformations.
Figure 37 shows the design criteria for achieving good rectangular to circular transformation for fan outlet.
Figure 38 shows examples of good and bad duct expansions and contractions.
Figure 39 shows examples of good and bad duct entries.
Figure 40 shows examples of good and bad bends.
Figure 41 shows the design criteria for achieving good elbow radius.
Figure 42 shows examples of good and bad branch entries for ducts.
Figure 43 shows the design criteria for achieving proper duct sizing for branches.
Figure 44 shows examples of good and bad airflow pattern in bends.
Figure 45 shows examples of good and bad airflow patterns in axial fans.
Figure 46 shows the design criteria for achieving a good discharge stackhead.

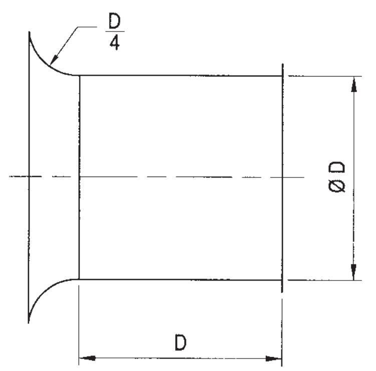


Figure 34. Flared Inlet

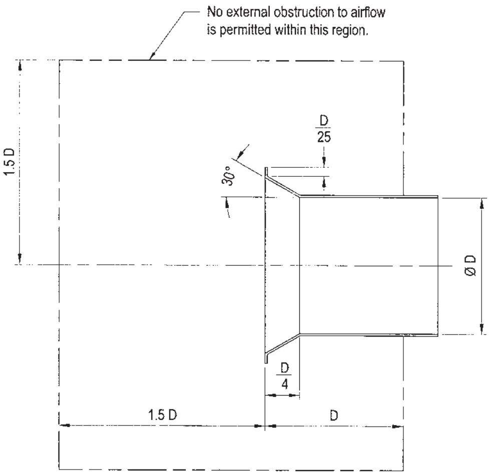
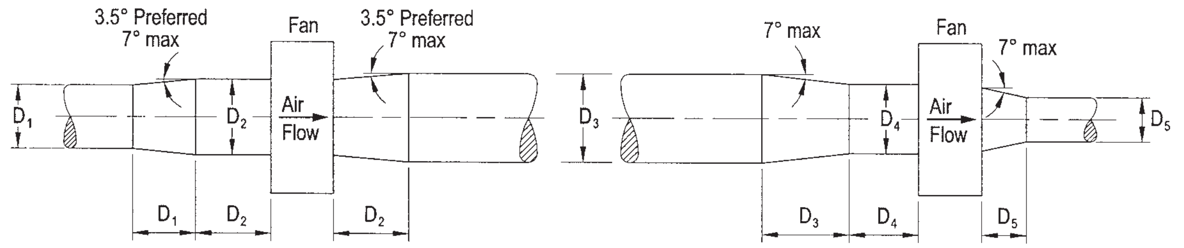


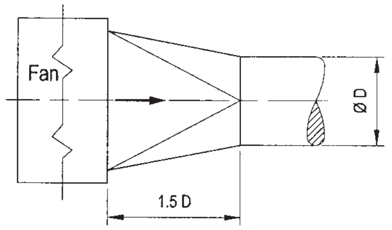
Figure 35. Conical Inlet



Circular expansion before and after a fan

Circular reduction before and after a fan

Figure 36. Circular Duct Transformation



Circular/rectangular area ratio should be greater than 65 %

Figure 37. Rectangular-to-circular transformation for fan outlet.

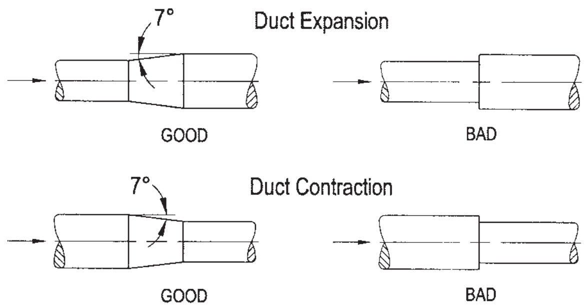


Figure 38

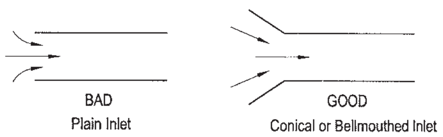


Figure 39. DUCT ENTRY

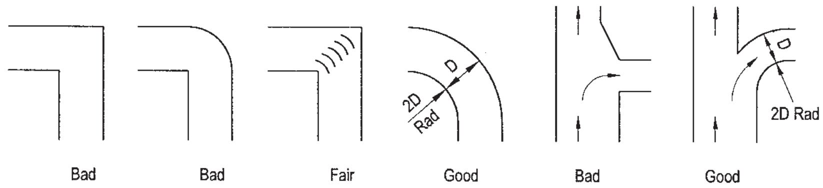


Figure 40. BENDS

ASPECT RATIO $\left(\frac{W}{D}\right)$

Elbows should have $\left(\frac{W}{D}\right)$ and $\left(\frac{R}{D}\right)$ equal to or greater than (1)

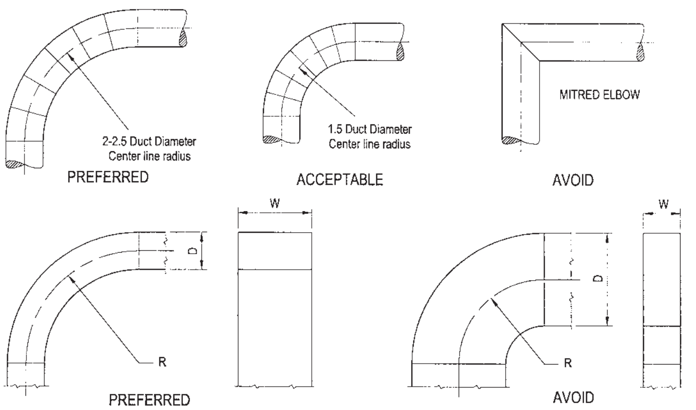


Figure 41. ELBOW RADIUS

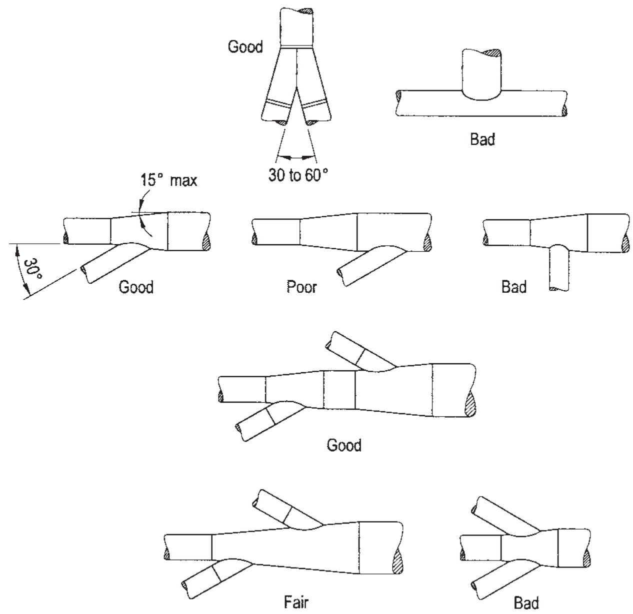


Figure 42. BRANCH ENTRY
Branches should not directly enter opposite each other.

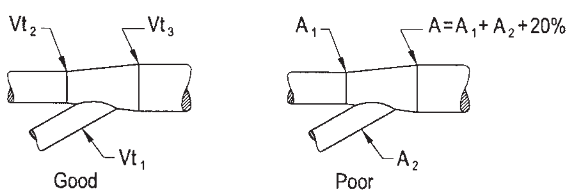


Figure 43. PROPER DUCT SIZING

Vt = Minimum transport velocity

A = Cross section area

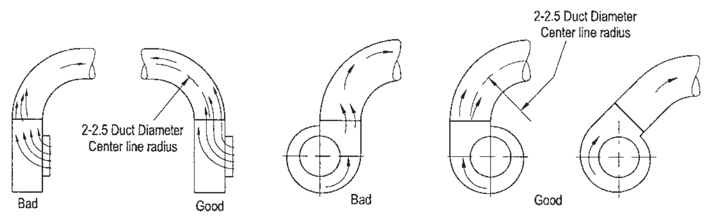


Figure 44. AIR FLOW IN DUCTS

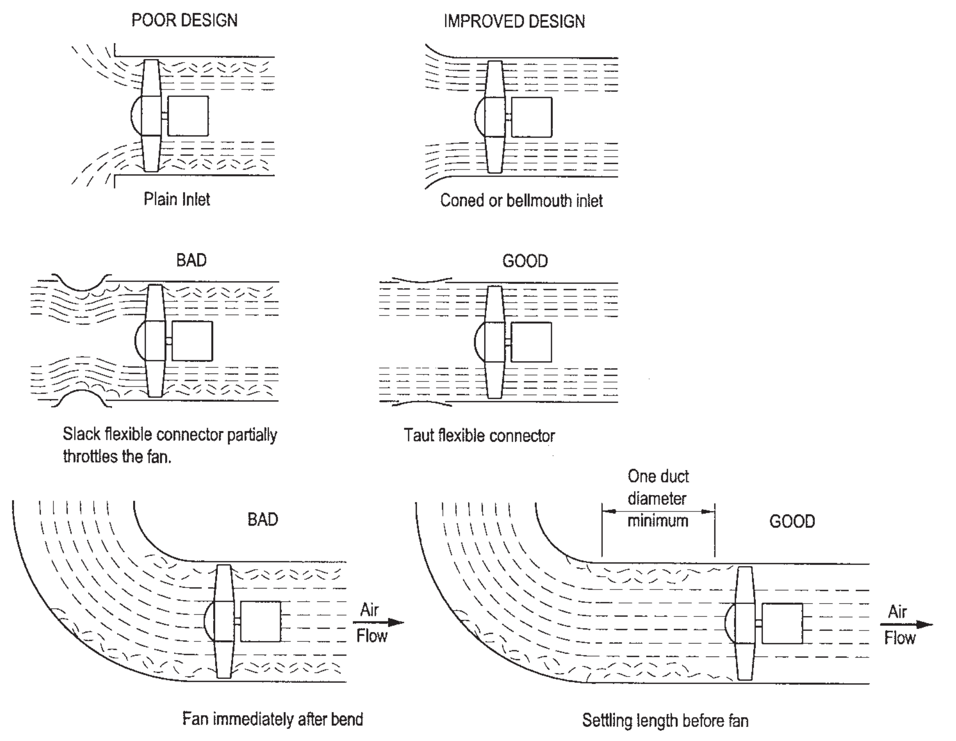


Figure 45. Air Flow In Axial Fans

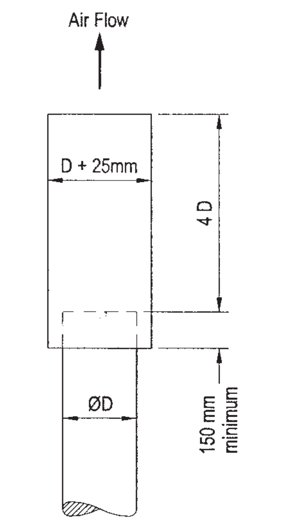


Figure 46. Stack Head Design

21. FAN VIBRATION – CAUSES AND REMEDIES

Fans are subject to vibration because they are relatively limber machines that operate at high speeds. Two types of vibration can cause trouble – aerodynamic and mechanical.

21.1 Aerodynamic Vibration (Pulsation)

If the vibration is aerodynamic it will usually disappear or change remarkably with substantial change in the volume of air flowing through the fan. This is usually accomplished by opening dampers, or by opening an access door or removing a section of duct near the fan. These reduce the system's pressure and increase air flow. Some causes of aerodynamic vibration are:

(a) Unstable fan performance – Some fans pulsate when operating to the left of the peak of their static pressure curves. Pulsation may be due to an error in system pressure calculation or design. This causes the fan to operate at a point other than the one for which it was originally selected.

It may be possible to eliminate pulsation by moving the fan wheel, so as to increase the overlap of the wheel and inlet cone. By introducing a large amount of recirculation, more air is allowed to circulate through the fan wheel than travels through the fan. The fan wheel gets a volume of air which allows it to perform without pulsating. In general, increasing the overlap by a distance equal to two per cent of the wheel diameter will eliminate pulsation.

This form of pulsation occurs when the fan wheel does not move enough air to fill the blade passages. Centrifugal fans with backward-inclined blades (the most efficient, and non-overloading type) are particularly subject to this phenomenon. However, fans with backward-inclined airfoil blades can be designed to be extremely stable.

(b) Faulty duct or system design (particularly at the fan inlet) – Aerodynamic vibration may be due to a poor inlet connection to the fan. Inlet boxes and inlet elbows should be vaned. When air is made to flow through a sharp turn as it enters the fan it tends to load just part of the fan wheel. The result is always decreased performance, and pulsation can occur.

21.2 Mechanical Vibration
If vibration in a system does not change when the airflow is increased and the pressure reduced, then it is more than likely to be mechanical in nature. If vibration is mechanical, perform the following steps:

a. Check for cleanliness. Even a thin layer of dirt can cause a surprising amount of unbalance. Use solvent, wire brushes and scrapers to get the wheel really clean.
b. Examine the wheel for indication of abrasion or corrosion.
c. Check the V-belt drive or coupling for alignment. Examine the grooves of sheaves and the surfaces of belts. Couplings can shift a few thousandths of an inch in shipment, or when the fan is bolted to a slightly uneven support.
d. Bearings can cause vibration. Listening with an industrial stethoscope or a screwdriver will often tell an experienced mechanic the condition of the bearing.
e Study the structure supporting the fan. Is it adequate? Quite frequently fans are mounted on supports that have a natural vibration frequency near that of the fan. Under such conditions it is almost impossible to balance all the rotating components finely enough to prevent an objectionable amount of vibration. If one of the system's natural frequencies coincides with the rotational frequency, resonance results.

If the motor is not the cause of unbalance, the cause is probably an unbalanced fan wheel. Or it could be a crooked shaft or an unbalanced half coupling or sheave still mounted on the fan shaft.

Avoiding pulsation and vibration is a matter of proper equipment specification, system design and installation.

21.3 Vibration Transmission
Vibrations are induced in a fan by unbalanced centrifugal forces and by aerodynamic forces. Some net force will be transmitted to the supporting structure,which forms part of the vibrating system. Many fans can be installed without vibration isolators, but the system must be designed accordingly. Vibration isolation is necessary whenever the vibrations in the supporting structure must be limited because they are annoying or even destructive.

The amount of force transmitted between members of a vibrating system depends on the masses, stiffnesses, and damping present.

21.4 Vibration Isolation
The materials usually used for vibration mounts are steel springs, rubber-in-shear, and cork. Cork or a similar material can be used in sheets under direct loads, since it will compress. Rubber, is usually bonded to two steel parts and loaded in shear. Steel springs are loaded directly in compression.

Isolators should be installed according to the distribution of the load, which may be due to thrust forces as well as to the dead weight of the equipment. If the fan is to be held level, the deflections must be uniform. The deflection should be chosen to give the proper natural frequency. Usually, steel springs are required for fan speeds below about 700 rpm but can be used at any speed. Rubber-in-shear can be used for speeds above 700 rpm, but steel springs will often be needed to limit the transmissibility to acceptable values. Cork can be used above about 1200 rpm.

It is important for a fan and its driving motor be mounted on a common rigid base. If isolation is required, it should be provided between the base and the supports so that the base, rather than the isolators, must withstand any torque or belt pull. Although the additional mass of the base will limit the system's amplitude of vibration, it will not alter the magnitude of the forces transmitted.

Flexible connections must be used between the fan and any ductwork on either the inlet or the outlet.

Inertial masses may be needed to restrict the amplitude of vibration regardless of whether the fan is mounted on isolators or not. As a rule to thumb, use a mass of concrete two to three times the mass of the fan and drive.

22. BALANCING

22.1 Purpose of Balancing
An unbalanced impeller will cause vibration and stress in the impeller itself and in its supporting structure. Balancing of the impeller is therefore necessary to accomplish one or more of the following: (a) Increase bearing life, (b) Minimize vibration, (c) Minimize audible noises, (d) Minimize operating stresses, and (e) Minimize power losses

Unbalance in just one rotating component of an assembly may cause the entire assembly to vibrate. This induced vibration in turn may cause excessive wear in bearings, shafts etc., and substantially reducing their service life. Vibrations cause highly undesirable alternating stresses in structural supports and frames which may eventually lead to their complete failure. Vibrations may be transmitted through the floor to adjacent machinery and seriously impair its proper functioning.

22.2 Causes Of Unbalance
The excess of mass on one side of a rotor is called unbalance. This may be caused by a variety of reasons including the following:

a) Tolerances in fabrication, including casting, machining, and assembly.
b) Variation within materials, such as voids, porosity, inclusions, grain, density and finishes.
c) Nonsymmetry of design, including motor windings, part shapes, location, and density of finishes.
d) Nonsymmetry in use, including distortion, dimensional changes, and shifting of parts due to rotational stresses, aerodynamic forces, and temperature changes.

22.3 Types Of Unbalance

22.3.1 Static Unbalance
This type of unbalance is found primarily in narrow, disc-shaped impellers. It can be corrected by a single mass correction placed opposite the center-of-gravity in a plane perpendicular to the shaft axis, and intersecting the centre of gravity. Static unbalance, if large enough, can be detected with conventional gravity-type balancing methods.

22.3.2 Dynamic Unbalance
It is the most frequently occurring type of unbalance and can only be corrected by mass correction in at least two planes perpendicular to the shaft axis.

23. SOUND

23.1 Definitions
Decibel (db) is a term borrowed from electrical terminology. Whenever ratios of power (highest divided by lowest) are so large as to be unwieldy when expressed in ordinary numbers, they can be expressed in "bels" or "decibels". First, a base, or reference level is established. Then the value to be expressed is divided by the reference level. The logarithm (to base 10) of the ratio is the number of Bels. 10 Decibels equal 1 Bel.

A **Pure Tone** is a sound with just one frequency, such as a clear whistle.

A **Broad Band Sound** is one that is devoid of pure tones, and is spread more or less evenly over a range of frequencies.

A **White Noise** is a broad band sound that is constant through the entire audible spectrum.

Absorption occurs when sound is absorbed rather than reflected. Acoustical materials absorb sound. Hard surfaces reflect it. Perfect absorption is an open area (free field).

A **Reverberant Room** is one that has a large proportion of hard walls, with little absorption. Sound is reflected and reverberates.

A **Free Field** is the opposite of a reverberant room. Sound is able to travel away from the source without being reflected back. A sound generated at the top of an outdoor tower would be in a perfect free field.

An **Anechoic Chamber** is a room so thoroughly lined with absorption material that it approximates a free field.

Attenuation is another word for absorption, but usually used in connection with duct borne sound.

An **Attenuator** is a device placed in a duct or plenum to absorb sound from the air passing through it. Attenuators are also called "silencers," "mufflers" or "traps".

Equal Loudness Contours are curves of sound pressure level versus frequency drawn such that the average human ear hears all sounds on a curve as seeming to be equally loud.

Sones are linear units of loudness. A sound pressure spectrum is converted to a Sone value by weighting each octave band pressure level in relation to how loud it sounds. The weighting is done according to the equal loudness contours. A sound of 10 Sones seems twice as loud as one of 5 Sones, etc.

Noise Criteria Curves are approximately equal loudness contours tagged with numbers. The number is the sound Pressure Level of the curve as it passes through 1000 Hz. The NC-40 curve passes through 40 db at 1000 Hz. A specification for a room to meet NC-40 means that when an octave band sound pressure level test is plotted on the NC chart, no octave band will extend above the NC-40 curve.

The Absorption Coefficient a, is the decimal fraction of the sound that is absorbed when the sound strikes the surface perpendicularly. If one forth of the sound is absorbed and three fourths reflected, a = 0.25.

Sound is caused by air pressure fluctuations.

23.2 Sound Wave and Octave Band Frequencies

The energy of a pebble when thrown into a pond creates surface waves which radiate out in concentric circles from the point of impact and which are sinusoidal in shape. Similarly, a vibrating object, such as a tuning fork, alternately compresses and rarifies the air particles adjacent to it and sends out invisible sinusoidal sound waves. These waves radiate from the source in a more or less spherical pattern. Sound waves are minute variations in the atmospheric pressure and, since our ears are attuned to the relatively unchanging atmosphere, the waves are picked up and interpreted by our ears as sound. Human beings with healthy ears can perceive oscillations of air pressure between approximately 16 Hz (infrasonic) and 16000 Hz (audible sound).

Both the loudness and frequency of the sound wave are important considerations in the analysis and attenuation of sound. A sine wave consists of crests and troughs which represent the rarifactions and compressions of the vibrating source in the air. The height (amplitude) between the crests and troughs is an indication of the loudness of the sound. The number of crests (cycles) occurring per unit of time designates the frequency (pitch) of the sound and is usually expressed as cycles per second

(Hertz). Sound produced by mechanical and electrical equipment is made up of a great number of frequency components. The human ear responds differently to these various frequencies. Low frequency sound is not as noticeable as higher frequency sound, therefore, the measurement of both loudness and frequency are necessary. For fan applications, only frequencies between 45 to 11,000 Hz are of interest. Audible sound can be divided into 8 octave bands.

Table 6 below outlines the octave band mid frequencies.

Octave Band	1	2	3	4	5	6	7	8
Frequency range (Hz)	45 - 90	90 - 180	180 - 355	355 - 710	710 - 1400	1400 - 2800	2800 - 5600	5600 - 11200
Approx. Geometric Mid Frequency	63	125	250	500	1000	2000	4000	8000

Table 6 : Octave Band Mid Frequencies

23.3 Sound Power Level

Sound power is the rate at which sound energy leaves the fan and travels along a duct attached to the inlet or outlet. If the duct is removed, the quantity of sound power radiated into the surrounding atmosphere is called the "open inlet (or outlet) sound power."

The sound power level is denoted as L_w and is defined as: $L_w = 10 \log_{10} \dfrac{\text{sound power of source, watts}}{10^{-12}\text{ watts}}$

and is expressed in decibels, dB

23.4 Sound Pressure Level

The human ear is, in effect, a sound measuring device which can distinguish the minute pressure variations of the sound wave.

The decibel reading of the sound meter indicates the level of sound (sound pressure level) at the point of measurement. This pressure is the root mean square (RMS) value of the minute variation in the atmosphere created by the sound wave. The base or reference value is established at the quietest sound that can be heard by the human ear. This is equivalent to pressure variation of $2\text{x}10^{-5}$ Pa.

The sound heard by the ear and measured by the sound level meter is defined as the sound pressure level ,

$$L_p = 20 \log_{10} \frac{\text{sound pressure, Pa}}{\text{reference pressure, } 2 \times 10^{-5}\text{ , Pa}}$$

and is expressed in decibels, dB

The sound meter reading includes not only the effect of the immediate sound source but also the effect of the floor, walls and ceiling which may absorb or reflect sound depending upon their surfaces.

When measuring noise levels, care must be taken to eliminate the effect of all sources of noise other than the one to be measured. With two or more sources in an area, it is difficult to determine the exact amount of noise generated by each source.

23.5 Directivity

The sound level of a fan varies at different locations around the fan. The noise level in front of the fan outlet is higher than the noise level behind the fan at the same distance away. However this effect is difficult to measure.

Hard surfaces near the fan also affect the directivity. Each additional wall can double the pressure level (add 3 dB). The directivity variable "D" can be assigned the following values:

D = 1 No reflecting surfaces (free field)
D = 2 Floor only
D = 4 Floor and 1 wall only
D = 8 Floor and 2 walls (corner)

23.6 Relationship between Sound Pressure Level and Sound Power Level

This is a complex subject, however the following equation is a good estimate

$$L_p = L_w + 10 \text{ Log}_{10} \{ (D/4\pi r^2) + 4/R \} + 10.5, \text{ where}$$

L_p = sound pressure level
L_w = sound power level
D = directivity factor
r = radius in feet from source (1m = 3.28 feet)
R = room constant = Sa / (1-a), a = absorption coefficient, S = surface area in square feet

If the value for S is small (small room), the value of 4/R will be large and the difference between the sound pressure and sound power becomes very small. The value a is dependent on both the construction material of the wall and the frequency. For estimation purposes, a value of a = 0.2 can be used for all frequencies.

23.7 Weighted Sound Pressure Level, dBA

In order to adapt sound measurements to the true audible sensations of the ear, sound pressure levels are weighted at low and high frequencies by means of filters fitted to sound measuring instruments. These weighted sound pressure levels are designated dBA, dBB, dBC, and dBD. High frequencies are perceived to be more disturbing than lower ones. For example, a sound of 80 dB @ 1000 Hz would subjectively sound the same as a sound of 106 dB @ 63 Hz. To evaluate noise at work places or with regards to annoyance to neighbours, measurements in dBA should be used. The dBA weighting takes into account the sensitivity of the human ear for moderate noise levels experienced in engineering.

For practical purposes, the sound level of each octave band can be corrected for A-weighting as shown in Table 7 below:

Octave Band	1	2	3	4	5	6	7	8
Correction, dB	-26	-16	-9	-3	0	+1	+1	-1

Table 7 : dBA Corrections

Examples of sound pressure levels:
Threshhold of pain (135 dBA) ; Pneumatic drill (125 dBA) ; Discotheque & Boiler plate shop (105 dBA) ;
Shouting & Noisy factory (90 dBA) ; Factory machine shop (80 dBA) ;
Crowded street & Restaurant (70 dBA) ; General office (60 dBA) ; Low conversation (50 dBA) ;
Whisper (25 dBA) ; Threshold of audibility (0 dBA)

23.8 Sound Level Meter
A sound level meter (with a microphone) and an octave band analyzer are usually used to measure sound. An octave band analyzer includes electronic filters which screen out all the octave bands except the one in which measurements are being taken. A reading of the sound pressure level for each octave band is taken and the result indicates the loudness of the sound (in decibels) in each band at the point of measurement. This measurement is useful in determining whether the sound level is acceptable or within specifications that may have been established. The sound meter does not and cannot measure the sound power level of the source.

23.9 Effect Of Distance On Sound Pressure Level
In a free field, the sound pressure level will decrease by approximately 6 dB when the distance from the source is doubled. In a room having surfaces which reflect some sound, the sound-pressure level will decline by 6 dB especially near the source. The pressure will cease to decrease significantly if you move beyond a certain distance. This area of relatively constant sound pressure is called the "reverberant" field.

23.10 Addition Of Two Sounds
When two sounds of different intensities are present together, the resultant level is not merely the addition of the two intensities in decibels. If the two sounds are equal, the combined sound powers are twice the power of one sound.

$$\text{Difference in db} = 10 \log_{10} \frac{\text{Combined level}}{\text{Level of one sound}} = 10 \log_{10} ({}^{2}/1) = 10 \times 0.3010 = 3.0 \text{ db}$$

A sound twice the intensity of another is said to be 3 db higher

24. NOISE FROM FAN SYSTEMS

If one compares the noise spectra for different types of fans it can be seen that centrifugal designs produce most of their noise at low frequencies whereas axial flow designs generate higher frequency noise. Most people will accept higher levels of low frequency noise and this is one of the reasons why centrifugal fans are generally used if noise is an important consideration.

Care must be taken to ensure that the airflow into the fan impeller itself is uniform; otherwise the fan will produce considerably more noise than specified.

24.1 Causes Of Noise In Ventilating Systems
In many installations of ventilating systems it is desired to reduce the noise due to the system. Before this is possible, it is necessary to consider the causes of noise.

Noise is transmitted through the ducts:
a) From equipment such as fans,
b) From outside, and transmitted through duct walls into the air stream
c) Due to eddying from obstructions both in the duct and at discharge grilles
d) Due to mechanical vibration of sheet metal duct sides, etc.

The noise generated by a fan is largely due to the creation of vortices or eddies from the blade and is dependent on blade shape, pressure, and velocity. This appears to manifest itself in two ways, one in the form of unpitched noise associated with general air movement, and the other a pitched sound due to the fan blade passing a fixed point.

24.2 Noise Attenuation In Ventilating Systems
Noise in ventilating systems could be attenuated (reduced in intensity) in three ways.

a) By physical barriers to its transmission through building materials
b) By acoustical filtering, absorption, or reflection during its airborne passage.
c) By reducing the air velocity to below about 2.5 m/s (500 fpm). Strictly speaking this is not attenuation of existing noise, but reduction in the generation of noise.

It must be realised however, that the higher frequency tones can be more easily suppressed using simple attenuator devices whereas reducing the lower frequency noise usually requires larger, more expensive solutions. It is sometimes possible to install a high speed, small diameter axial flow fan fitted with an attenuator, less expensively than a slower speed centrifugal fan generating similar noise levels and giving the same aerodynamic performance.

Care must be taken to ensure that the airflow into the fan impeller itself is uniform; otherwise the fan will produce considerably more noise than specified.

If the sound power level of a fan is too high, and no other selection is possible, attenuation must be introduced at appropriate locations in the system to prevent unacceptable noise being transmitted to the occupied spaces.

Attenuation can be provided by a variety of means including lining the ducts with absorptive material, changing the layout, using a plenum or by inserting proprietary attenuator units. Duct lining with absorptive material, especially at bends, is often adequate, providing sufficient length of duct is available. Bends are particularly useful for reducing high frequency noise, but are not very effective at low frequencies, particularly if the duct dimensions, are small.

Where a certain amount of noise is present as vibration in the fan body, the best action which can be taken to reduce this noise to a minimum is to isolate the fan from both mountings and connections. Anti-vibration mountings, if properly fitted will effectively isolate the fan from its mountings. Anti-vibration mountings usually comprise a bracket which is isolated by resilient means from the portion of the mounting which carries the fan.

Connections to inlet and outlet ducts should be made through resilient sections made from materials such as rubber or canvas. Where direct duct connections must be made, subsequent ductwork should be made of sufficiently stiff construction as to prevent drumming.

24.3 Preventive Measures For Sound Reduction
Preventive measures are required if the noise emitted by an installation reaches an unacceptable level in the surrounding area.

The most important primary preventive measures to avoid noise are the selection of the correct type and size of fan in order to operate at or near to the maximum efficiency point.

Further primary preventive measures are:

- Using vibration absorbing material between the baseplate and the foundation.
- Avoiding sudden changes of the pipe cross-sectional area.
- Use of large radius bends.
- Connection of the pipes to the fan by rubber expansion joints.

If no satisfactory results can be obtained in this way secondary preventive measures have to be used to reduce noise by means of insulating or silencing the sound.

The active preventive measures are mainly sound damping (reflecting) and sound silencing (absorbent) walls, and complete enclosures. Purposeful steps to reduce noise are only possible if the sound spectrum i.e. the distribution of the sound pressure level within the relevant band of frequencies is known.

Passive protective measures are achieved by sound deadening or sound absorbing cabins, e.g. control rooms, and by ear plugs or ear muffs.

25. ELECTRIC MOTORS

Polyphase induction motors are the normal standard drive for industrial fans, where the necessary three phase AC (alternating current) supply is almost always available. The usual squirrel-cage rotor has the merits of minimum maintenance, robustness and low cost.

25.1 Synchronous Speed
The stator (stationary iron member) is wound with coils of insulated wire inserted into slots and connected to the supply. The winding pattern determines the number of poles or peaks of magnetic flux round the circumference.

These magnetic poles rotate at a synchronous speed which is also the speed of the motor on no-load.

$$\text{Synchronous speed (rev/s)} = 2 \times \frac{\text{supply frequency (Hz)}}{\text{number of poles}}$$

Table 8 below outlines the synchronous speed of electric motors at 50 Hz supply frequency.

Synchronous Speed At 50 Hz Supply Frequency		
Number of poles	Rev/sec (rps)	Rev/min (rpm)
2	50	3000
4	25	1500
6	16.7	1000
8	12.5	750
10	10.0	600
12	8.3	500
14 (rare)	7.1	429
16	6.25	375

Table 8 : Synchronous speed

When an induction motor is loaded the speed falls, the drop from synchronous speed being known as the slip. The slip at full load ranges from 1% of synchronous speed for large motors to 10% for smaller motors.

The direction of rotation for a 3-phase induction motor depends on the motor connection to the power supply. The rotation can be easily reversed by interchanging any two input leads to match the fan required fan rotation.

25.2 Classes Of Protection

The majority of motors are supplied with classes of protection as shown in Table 9 below.

IP 54	Complete protection against contact with electrically live internal moving parts. Protection against injurious deposits of dust. The penetration of dust is not completely prevented, but the dust may not penetrate in such quantities that the functioning of the machine is adversely affected.	Water, sprayed against the machine from any direction, must not have any injurious effect.
IP 55		A jet of water from a nozzle, directed against the machine from any direction , must have not injurious effect.
IP56		Water flooding over the machine, eg. Due to heavy seas, must not enter the machine in injurious quantities.
IP65	Complete protection against contact with live or moving parts inside the enclosure and against the ingress of dust.	Water projected by a nozzle against the machine from any direction shall have no harmful effect.

Table 9 : Classes Of Protection for Electric Motors

25.3 Classes of Insulation

The three most commonly used classes are B, F and H. The ambient temperature range for Class B is up to 40°C, Class F from 41°C to 65°C and Class H from 66°C to 90° C.

Operating these motors slightly above these limits will reduce its life expectancy. For example, operating a Class F motor at 75° C, which is 10° C above the limit can decrease the motor's life expectancy as much as 50%.

25.4 Enclosures

The two most commonly used enclosures are the Totally Enclosed Fan Cooled (TEFC) and Totally Enclosed Air Over (TEAO).

The totally enclosed motors are not airtight but are adequately enclosed to prevent the free exchange of air between the inside and outside of the enclosure.

The TEFC motors have an external fan mounted on the motor shaft. The fan directs the air over the motor frame for cooling. The TEFC motor is used for indoor or outdoor applications where dust, dirt, mild corrosives, and water are present in moderate amounts.

The TEAO motor does not have an external fan mounted on the motor shaft and is not self cooling. It should only be used in applications where the fan itself provides the airflow over the motor frame for cooling, eg. direct driven axial flow fan.

25.5 Starting Three-Phase AC Induction Motors

Direct-on-line (DOL) starting is the ideal method of starting, since it is the simplest. Full supply voltage (mains supply line voltage) is applied directly to the motor, by way of a contactor. At the instant of connection, a starting current of from four to six times the rated current flows. As a result of this, the voltage in the supply lines falls and other consumers may be affected. For this reason electricity supply undertakings place power limitations on the direct-on-line method of starting.

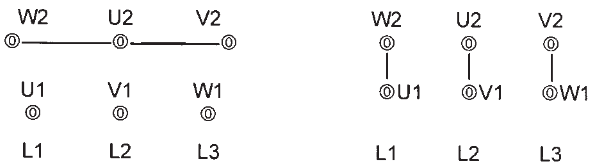
Larger motors (rated power above 5.5 kW, as a guideline) may only be connected to the mains supply with reduced voltage, in order to alleviate the starting current surge. The best-known of these is the star-delta (Y-Δ) starting method.

For star-delta starting, star-delta starters or corresponding combinations of contactors are used, to which all six ends of the windings are led. Starting takes place with star-connection, when the phase voltage in the windings is only $1/\sqrt{3}$ of the supply line voltage, in contrast to delta-connection (the operating connection), in which the full line voltage is applied to the phase winding.

25.6 Wiring Instructions

Electrical connections described in this section should be carried out by a qualified electrician only. All connections should be carried out in accordance with relevant load regulations.

1. Fit or check position of bridges on motor terminal block. The motor nameplate will show type of connection. Y (Star) or Δ (Delta) for the required voltage. Refer nameplate or inside of terminal box lid for diagram.



Star Connection Delta Connection

2. If fan/motor is to be started in a Direct on Line (D.O.L.) configuration with a contactor and overload, then an electronic overload is to be used. For high pressure blowers, a bimetal thermal overload **must not** be used as an extended starting time due to the high impeller inertia, will cause it to trip prematurely and not allow the motor sufficient time to reach its full speed. An electronic overload (Alan Bradley SMP-2 or equivalent) set to a class 20 or 30-trip mode will overcome this problem.

   Other suitable starting methods are as follows:

   Star – Delta Starters
   Soft Starters
   Variable Frequency Controllers

3. High pressure fans have an overloading characteristic. Hence if the motor running current is higher than noted on the nameplate, it will need to be reduced by dampening (partially closing off the fan inlet or discharge). This is particularly important especially if an overload relay is not used and if not corrected will eventually burn out the motor.

25.7 Types Of Hazardous Location Motors

Ex d Flameproof Motors for Industrial Use (Class 1 Zone 1, Flameproof gas groups I, IIA & IIB)
Ex e Increased Safety Motors (Class 1 Zone 1, Gas group IIA, IIB, IIC)
D.I.P. Dust Ignition Proof Motors (Class II Division 1 & 2 , IP65)
Ex n Non Sparking Motors (Class 1 Zone 2, Gas group IIA, IIB, IIC)

25.7.1 Hazardous Areas

There are a number of defined Hazardous Areas covering Gases & Dusts, it is therefore strongly recommended that the relevant Australian Standards & Statutory Authorities be consulted prior to final selection of the motor.

Many Gases, Vapours & Dusts which are generated, processed, handled & stored in industry are combustible. When ignited they may burn rapidly & with considerable explosive force if mixed with air in the appropriate proportions.

Areas where Gases, Vapours, Dusts & Fibres occur in dangerous quantities are classified as hazardous.
Classification of areas are:

Gases, Vapours, Mists	-	Class I
Dusts	-	Class II

Gas groupings are further defined for either

Coal Mining (Methane)	-	Group I
Or Other Industries	-	Group II

With Group II Gases, they are further sub-divided into sub groups IIA, IIB, IIC depending upon the ignition point of the gas.

25.7.2 Zone Classification
Zonal classification is also required where explosive gas atmospheres are present & they indicate the probability of the presence of a flammable, combustible or explosive material, the extent dimension & shape of the hazardous area together with the volume in which the hazardous material can be expected. There are 3 zones.

Zone 0 - An area in which an explosive gas atmosphere is continuously present or is present for long period of time.
Zone 1 - An area in which an explosive gas atmosphere is likely to occur in normal operation.
Zone 2 - An area in which an explosive gas atmosphere is not likely to occur in normal operation & if it does occur it will exist for a short period only.

25.7.3 Temperature Classification
Hot surfaces can cause ignition of gases, vapours & dust, therefore It is necessary to ensure that the maximum surface temperature of equipment introduced into a hazardous area does not exceed the ignition temperature for the gas, vapour or dust in the hazardous area.

Group l Gases – Maximum Surface Temp. 150˚C

Group ll Gases & Class ll Dusts are given a Temperature Class (T) based on the maximum surface temperature of the equipment.

Temperature Classes are:- T1:450ºC, T2:300ºC, T3:200ºC, T4:135ºC , T5:100ºC, T6:85ºC

25.8 Fan Starting Requirements
A fan is an energy converter. Electrical energy rotates the fan wheel through a driving motor and increases the static pressure (potential energy) of the air handled by the fan in order to overcome resistance to air flow offered by the duct system.
The wheel also increases the velocity pressure (kinetic energy) of the air which is the energy required to maintain the air in motion. The driving motor must be capable of starting the fan from rest and accelerate it to operating speed, with a minimum of disturbance to the electrical system.

To start and accelerate a fan to operating speed it is necessary to:

1. Overcome bearing resistance. This resistance can vary with the type of bearing used. It is low for anti-friction types and relatively high for sleeve types.
2. Accelerate the inertia of the fan wheel and shaft. This inertia is generally designated as the moment of inertia or (WR^2). The motor must provide energy to accelerate it together with the inertia of the drive pulleys or coupling. The moment of inertia for heavier wheels is greater.
3. Provide energy to the fan wheel as it begins to deliver air into the duct system. The power required varies with the cube of the fan speed. It is insignificant at low speeds, but increases rapidly as the fan wheel comes up to operating speed.

Fans, when selected for low static pressure, may be specified with motors which are not large enough to start the fan, accelerate and operate at the design RPM without overheating the motor or overloading the electrical system. Minimum motor sizes become critical for large fan sizes. In general, smaller fans do not present a starting problem.

Direct driven units will require larger motors to accelerate the fan inertia load to the designed speed, since the WR^2 referred to the motor is

$$\left(\frac{\text{Rpm fan}}{\text{Rpm motor}}\right)^2 WR^2 \text{ of fan}$$

Whenever inlet vanes or outlet dampers are used, the starting load and motor heating are reduced, if such devices are kept closed until after the fan has accelerated to operating speed.

25.9 Power Requirements To Start Fan
Two separate power requirements must be considered when selecting the motor for a fan:

1. The absorbed power required to turn the fan at the proper speed to deliver the design volume at the necessary static pressure.
2. The minimum motor power required to bring the fan to the necessary speed by overcoming the inertia load of the wheel and shaft.

The inertia load of the wheel and shaft is measured as moment of inertia (WR^2). This value is used to determine the capability of a motor to bring the load up to the required speed before the motor overheats.

Fans selected in the lower static pressures, may specify motors, which are not large enough to start the fan without overheating the motor or the electrical system. Generally, smaller fans do not present a starting problem. Whenever devices such as inlet vanes and outlet dampers are used and kept closed until the fan has reached operating speed, both motor heating and starting load are decreased.

26. CORROSION PREVENTION

26.1 Material Selection
The most common method of preventing corrosion is to select the proper metal or alloy for a particular corrosive application. One of the most popular misconceptions to those not familiar with metallurgy or corrosion engineering concerns the uses and characteristics of stainless steel. Stainless steel is the generic name for a series of more than 30 different alloys containing from 11.5 to 30% chromium and 0 to 22% nickel together with other alloy additions. Stainless steels have widespread application in resisting corrosion, but it should be remembered that they do not resist all corrosives.

In alloy selection, there are several "natural" metal-corrosive combinations. These combinations of metal and corrosive usually represent the maximum amount of corrosion resistance for the least amount of money. Some of these natural combinations are as follows:

1. Stainless steels-nitric acid
2. Nickel and nickel alloys-caustic
3. Hastelloys (Chlorimets)-hot hydrochloric acid
4. Aluminium-nonstaining atmospheric exposure
5. Tin-distilled water
6. Titanium-hot strong oxidizing solutions
7. Tantalum-ultimate resistance
8. Steel-concentrated sulfuric acid

26.2 Design Allowance For Corrosion
The design of a structure is frequently as important as the choice of materials of construction. Design should consider mechanical and strength requirements together with an allowance for corrosion.

Design systems for the easy replacement of components that are expected to fail rapidly in service. Frequently, fans in chemical plants are designed so that they can be readily removed from a piping system.

Avoid excessive mechanical stresses and stress concentrations in components exposed to corrosive mediums. Mechanical or residual stresses are one of the requirements for stress-corrosion cracking.

The most general rule for design is to avoid heterogeneity. Dissimilar metals, uneven heat and stress distributions, and other differences between points in the system lead to corrosion damage. Hence, in design, attempt to make all conditions as uniform as possible throughout the entire system.

26.3 Acids

26.3.1 Mineral Acids
Most of the severe corrosion problems encountered involve the mineral acids or their derivatives. The most common mineral acids include sulfuric acid, nitric acid, hydrochloric acid and phosphoric acid.

26.3.2 Sulfuric Acid
More sulfuric acid is produced than any other chemical in the world. The principal uses of sulfuric acid are for production of hydrochloric acid, other chemicals, and their derivatives; pickling of steel and other metals: manufacture of fertilizers, dyes, drugs, pigments, explosives, synthetic detergents, rayon, and other textiles; petroleum refining; storage batteries; metal refining; and production of rubbers.

Ordinary carbon steel is widely used for sulfuric acid in concentrations over 70%.

Ordinary grey cast iron generally shows the same picture as steel in sulfuric acid. Cast iron shows somewhat better corrosion resistance in hot strong acids, and in very hot and very strong acid it is better than most materials, including stainless high alloys. However, the corrosion rates are high. The better resistance in this case is probably due to the graphite network interfering with the reaction between the acid and the metallic matrix.

26.3.3 Nitric Acid
The choice of metals and alloys for nitric acid services is quite limited as far as variety is concerned. For most plant applications the choice is usually between only two general classes of materials – the stainless steels and alloys, and the high-silicon irons.

26.3.4 Hydrochloric Acid
Hydrochloric acid is the most difficult of the common acids to handle from the standpoints of corrosion and materials of construction. Extreme care is required in the selection of materials to handle the acid by itself, even in relatively dilute concentrations, or in process solutions containing appreciable amounts of hydrochloric acid. This acid is very corrosive to most of the common metals and alloys.

The limits for so-called acceptable corrosion rates are usually raised when considering materials of construction for handling hydrochloric acid. Good judgment is required to obtain a good balance between service life and cost of equipment. Hastelloy B, Hastelloy C, tantalum, and molybdenum are used in hydrochloric acid applications.

26.3.5 Phosphoric Acid
The most widely used alloys are type 316 stainless steel. Copper and high-copper alloys are not widely used in phosphoric acid. Aluminium, cast iron, steel, brass, and the ferritic and martensitic stainless steels exhibit poor corrosion resistance.

26.4 Alkalies
The common alkalies such as caustic soda (NaOH) and caustic potash (KOH) are not particularly corrosive and can be handled in steel in most applications where contamination is not a problem. However, one must guard against stress corrosion in certain concentrations and temperatures. Rubber-base and other coatings and linings are applied to steel equipment to prevent iron contamination.

Nickel and nickel alloys are extensively used for combating corrosion by caustic. Nickel is suitable under practically all conditions of concentration and temperature.

Aluminium is a very poor material for handling caustic. This metal and its alloy are rapidly attacked even by dilute solutions.

Many metals and alloys exhibit low corrosion rates in caustic, but steel, cast iron, nickel, and nickel-containing alloys are suitable for a large majority of applications.

Ammonia and ammoniacal solutions generally do not present difficult corrosion problems. In manufacture and handling, steel and cast iron are satisfactory except for high temperatures, where types 430 and 304 stainless steels are required. The major warning is not to use copper and copper-base alloys because even traces of ammonia can cause stress corrosion.

26.5 Atmospheric Corrosion
Corrosion by various atmospheres accounts for more failures on a cost and tonnage basis than any other single environment. Atmospheres can be classified as industrial, marine, and rural. Corrosion is primarily due to moisture and oxygen but is accentuated by contaminants such as sulfur compounds and sodium chloride. Corrosion of steel on the seacoast is 400 to 500 times greater than in a desert area. Steel specimens 25 meters from the seashore corroded 12 times faster than those 250 meters away. Sodium chloride is the chief contaminant.

Industrial atmospheres are more corrosive than rural atmospheres, primarily because of sulfur gases generated by the burning of fuels. SO_2 in the presence of moisture forms sulfurous and sulfuric acids, which are both very corrosive.

Small amounts of copper (0.1 percent) increase resistance of steel to atmospheric corrosion because it forms a tighter, more protective rust film. Small amounts of nickel and chromium produce similar effects. Nickel and copper are helpful in industrial atmospheres because they form insoluble sulfates that do no wash away and thus afford some protection. For almost complete rust resistance in ferrous alloys, we must go to the stainless steels. Copper, lead, aluminium, and galvanised steel are also widely used for atmospheric applications.

26.6 Seawater And Freshwater
26.6.1 Seawater
Seawater contains about 3.4% salt and is slightly alkaline, pH 8. It is a good electrolyte and can cause galvanic corrosion and crevice corrosion. Corrosion is affected by oxygen content, velocity, temperature, and biological organisms. Greatest attack from sea water occurs in the splash zone because of alternate wetting and drying and also aeration.

26.6.2 Fresh Water
Corrosivity in fresh water varies depending on oxygen content, hardness, chloride content, sulfur content, and many other factors. Fresh water can be hard or soft, depending on minerals dissolved. In hard water, carbonates often deposit on the metal surface and protect it, but pitting may occur if the coating is not complete. Soft waters are usually more corrosive because protective deposits do not form.

Low-alloy steels do not offer any advantage over ordinary steel in water applications (as compared with atmospheric corrosion). As is the case in atmospheric corrosion, complete corrosion resistance would require the more expensive stainless steels.

27. MAINTENANCE

Fans must be inspected periodically and maintained properly with regard to lubrication, belt tension and alignment, clean surfaces etc. Preventive maintenance is better than unexpected shutdown and subsequent repairs.

27.1 Foundations
Foundations (preferably concrete) should be level, rigid and should be equal to 3 or 4 times the fan weight. For structural steel foundation, it must be rigid enough to prevent excessive vibrations. The minimum natural frequency of any foundation should be 25 to 50% higher than the fan speed.

27.2 V-Belt Drives
V-belt drives require careful alignment of the pulleys and adjustment of the belt tension. This has to be done after the fan has been mounted on its foundation because the tightening of the mounting bolts may distort some of the previously aligned surfaces. Fan shaft and motor shaft should be parallel - adjust and shim the motor if necessary. Fan pulley and motor pulley should be adjusted axially so that their faces are not only parallel but also aligned. This can be checked with a straight edge as

shown in figure 47 .

It is important that the correct tension be maintained on the belts at all times, if power is to be transmitted as efficiently as possible, over the life of the belts.

The consequences of an under tensioned drive are excessive belt flap, slippage and overheating of the belts, premature belt wear and even belt breakage. On the other hand, an over tensioned drive can result in a premature bearing or shaft failure or belt breakage.

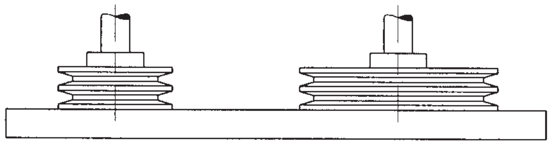


Figure 47. Checking Pulley Alignment

After running for the first hour, the belts are to be re-tensioned to the correct value and after a further few hours of running, the belts are to be re-tensioned again.

This procedure is necessary to take up the initial stretch that occurs with new belts and after this stretch has been taken up, the drive should perform satisfactorily provided that periodic adjustments to the belt tension are maintained. Loose belts can result in 10 to 20% reduction in fan speed.

The vee drive should be periodically checked for correct belt tension, conditions of belts, loosening of pulleys on shaft, groove wear etc.
Should belts need replacing, it is important that a matched set is obtained.

Figure 48 below shows three possible shapes a fan belt will assume, depending on its tightness. The belts should be adjusted until the belts have only a slight bow on the slack side of the drive while operating under load.

It is common for V-belt drives handling more than 15 kw to squeal a little on start up. Do not overtighten belt.

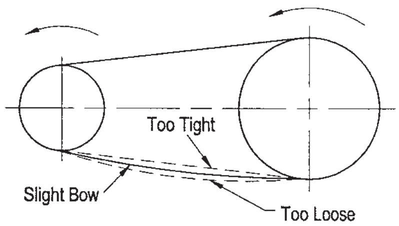


Figure 48. Fan Belt Shapes

27.3 Vibrating Fan

A fan running in an unbalanced condition gets steadily worse and causes serious trouble such as, bearing damage, weakening the motor supports, loosening of piping and surrounding equipment, springing the motor shaft and possibly damaging the impeller itself.

Commonest causes of vibration are wear, dirt build up on blades or hub, bent shaft and misalignment of components.

The amount of unbalance is measured by vibration detector. No hard and fast rule has been made on what vibration tolerances should be.

Table 10 below gives the suggested vibration tolerances.

	Vibration Velocity mm/sec	Comments
Smooth	0 to 2.0	
Fair	2.0 to 4.5	O.K. to run but should be rebalanced in due course.
Rough	4.5 to 11.0	Should be balanced as soon as possible.
Very Rough	Above 11.0	Should not be operated until balanced.

Table 10 : Vibration Tolerances

If the impeller balance requires adjustment, it is preferable to use weights bolted to the impeller, but if electrically welded balance weights are used, it is essential to earth the impeller immediately adjacent to the weld area and not to the fan casing. Failure to adopt this procedure may result in the welding current passing to earth through the bearing which could cause bearing pitting or burning.

27.4 Periodic Inspection
The fan will require periodic inspection and records should be kept of conditions, such as the amount of wear, balance, lubrication and paint work. When the fan is 'shut down' check and clean all components. Special attention should be paid to parts in the airstream, especially the impellers since build up on these sections could adversely affect balance and bearing life. Check all parts for wear and alignment, repair or replace as necessary.

27.5 Spare Parts
Replacement parts should be kept on hand for all wearing surfaces such as impellers, shafts, bearing races, seals and vee-belts. Complete item such as shaft-impeller assembly should be kept on hand if shut down time is critical.

27.6 Start Up and Stopping

27.6.1 Pre-Start Up Checklist
All securing bolts for motor, bearing assemblies, rotor, foundations, guards and connecting ductwork must be tight. Access doors should be tight and sealed. Check bearing alignment and make certain they are properly locked to the shaft and lubricated. Turn the fan impeller by hand to ensure that the impeller turns freely and does not strike or bind the fan housing or inlet bell. Also check the impeller and inlet cone alignment. The wheel could have shifted during shipment or somebody could have taken the fan apart and put it back together in the wrong relationship. In some fans, the relationship of the impeller to cone is very critical. Too much or not enough overlap can cause a loss of airflow.

Check drive alignment of vee-drive. Check electrical wiring to motor. All motors should be connected as shown on motor nameplate. Be sure power supply (voltage, frequency and current carrying capacity of wires) is in accordance with the motor nameplate. Always check to make sure motor bearings are lubricated.

27.6.2 Starting For The First Time
For high temperature fans, if the fan is run on cold gas initially, then measures should be taken, eg. restriction of flow through dampers, speed control etc., to ensure that the motor power limit is not exceeded.

Apply power to the motor for a short period and check for irregularities such as unusual noises, rubbing, high temperature or vibration. Also check correct impeller rotation.

If no problems are experienced fan may now be brought up to full speed.

Check vibration and bearing temperature for normal conditions.

27.6.3 Stopping The Fan
Please note that the impeller may continue rotating several minutes after stopping the motor. Make sure that the fan is at a complete stand still before attempting to service it.

27.7 Bearing Lubrication And Temperatures

27.7.1 Stored Fans
If fans are stored for more than 60 days, the wheel must be rotated every 30 to 45 days to prevent false brinelling of the fan bearings. False brinelling is caused by vibration of the balls or rollers between the races in a stationary bearing. This vibration may be either axial or oscillating. As the ball or rollers vibrates between the races, the lubricant is forced out of the contact area between the ball and race, causing metal-to-metal contact and localized wear of ball and races, which result in a rough and noisy bearing operation.

If the fans are not put in service prior to one year after shipment, the lubricant should be changed. Grease has a tendency to become hard and deteriorate, losing its lubricating qualities.

It is only necessary to pack about 50% of the bearing housing with grease. Excessive greasing will cause an immediate high temperature of the bearing on start up.

Clean grease fitting and gun before lubricating to avoid forcing foreign particles into the bearings. Add grease slowly until a slight bead is noticed around the bearing seal. Avoid storage in temperature below -20˚C which could cause a breakdown of the lubricant.

27.7.2 Bearing Temperatures
Most greases used are suitable for temperatures up to 100˚C. The appropriate operating temperatures using a standard grease is 75 to 80˚C during the initial start up, returning to a normal temperature of 45 to 60˚C, dependent on ambient temperature and application.

Bearing temperatures from 30˚C to 50˚C above ambient may be considered normal. Although higher temperatures may also be considered normal for certain applications, generally having temperatures in excess of 60˚C may be worthy of investigation.

It must be pointed out that whilst re-greasing bearings a sudden rise in temperature may be experienced. However, this is a temporary condition and the temperature will drop as the quantity of the grease inside the bearing stabilizes. It is preferable to re-lubricate the bearings while the shaft is rotating, as this assists in the eviction of any degenerated grease and its ultimate discharge from the bearings.

27.7.3 Initial Running Of Bearings
Bearings typically heat rapidly and return to a normal running temperature within the first four hours of installation start-up. On initial start-up, listen to the bearing for abnormal noises and monitor the temperature rise. If temperature rise is excessive, stop fan and remove some of the grease from the bearing and try again. It cannot be emphasized that the most common cause of rapid temperature rise on start-up of fans is excessive greasing of bearings.

If high temperature is still encountered, another thing which can be tried and which sometimes is quite successful, is to just untighten the bearing cap bolts about a quarter of a turn and re-tighten whilst the fan is running. This can relieve minor misalignment of the outer track to housing when it was initially tightened.

27.8 Storage Of Equipment

27.8.1 Prior To Installation
Short-term storage prior to installation requires no special attention.
Long term storage prior to installation requires the following attention:

a) Coat the shaft with an easily removable rust preventative.
b) Cover and seal all bearings, components and ancillaries including motor, filters etc. to prevent entrance of contaminants.
c) Block the wheel to prevent any unscheduled rotation.
d) It is important that the wheel be rotated at least once a month to circulate lubricant to the bearings.
e) Do not allow material of any kind to be stored on the fan.

27.8.2 For Extended Shut Down
Units that have been installed, but not operated for several months, should either have duct work disconnected or the duct openings covered. All drains, pipe connections and conduit must be covered with plastic caps or tape. Openings in the fan housing or unit casing should be covered an sealed to keep out dust, dirt and moisture. Check that all doors are closed.

Motor storage per the manufacturer's instructions, should include care in keeping motor dry (space heater should be used if necessary).

Oil or grease the control mechanisms and damper linkages periodically. Operate control mechanisms through its full range at least once a month. Inlet box damper and bearings generally do not require oil or grease.

Relieve tension on V-Belt drives by adjusting the motor base.

28. SAFETY

28.1 Guards
Rotating components of fans such as V-belt drives, shafts, coupling, cooling fins should be protected by guards. Exposed fan inlets and outlets should also be guarded.

Intakes to ductwork should, whenever possible, be screened to prevent the accidental or deliberate entrance of solid objects. For example, on a sawdust handling system an intake screen should be provided which will allow the entry of sawdust but prevent the entry of chunks of wood.

28.2 Repairs By Authorised Personnel
Testing, adjusting and maintenance of fan equipment exposes personnel to potential safety hazards. Only experienced mechanical person who are aware of mechanical safety hazards created by moving or rotating parts, should be authorised to work on fan equipment.

Many simple ventilating systems are the low pressure, low velocity type that seem relatively harmless. However, some fan systems have the added danger of high velocity air movement, which can blow or suck people off balance or blow dust or dirt into their eyes.

Solid objects can pass through the fan and be discharged by the impeller as potentially dangerous projectiles, as well as causing serious damage to the fan itself.

People have lost arms and hands, and have been killed when pulled into the rotating blades. Always lock out the electrical system before inspecting or working on a fan. Watch for pinch points on pulleys.

On material handling and high temperature fan systems, severe injury can be caused by people exposed to air blasts.

28.3 Initial Start Up of Fan
When a fan is being started up for the first time, a complete inspection should be made of all of the ductwork and the interior of the fan to make certain there is no foreign material which can be sucked into or blown through the ductwork.

1. Do not open access doors while the system is running (on the outlet, the door could virtually explode open as soon as it is unlatched). Where quick release handles are provided on access doors at least one positive bolt should be installed to prevent accidental opening.

2. All access doors should be closed before starting the system (an open access door on the inlet of the fan could

cause motor overload, as well as creating a suction hazard).

3. Do not run the system with duct work disconnected unless you are familiar with the entire system and are fully experienced in fan service work.

4. Whenever the system must be run with access doors open, or duct work disconnected, clear the area of all unauthorised personnel and establish sufficient guards and/or warning signs to keep unwary bystanders out of danger.

29. TROUBLE SHOOTING AND PROBLEM SOLVING

Table 11 provides a guide for fan trouble shooting and problem solving

Symptom	Possible Cause
1. Fan will not start	Blown fuses Broken belts Loose Pulleys Impeller touching housing Wrong voltage
2. Excessive noise and vibration	Misalignment of bearings, coupling, wheel or drive Unstable foundation Foreign material in fan causing unbalance Worn bearings Worn coupling Damaged impeller or motor Broken or loose bolts Bent shaft Fan wheel or driver unbalanced Fan delivering more than rated capacity Speed too high or fan rotating in wrong direction
3. Air volume too small	Wrong fan rotation Fan speed too slow Dampers closed too much Coils and filters dirty Inlet or outlet obstructions Fan too small for application Improperly designed turning vanes Air leaks in system Damaged impeller Wheel mounted backwards on impeller System resistance higher than design
4. Air volume too large	Wrong fan rotation Fan speed too high Dampers not installed Access door open Fan too large for application Oversized ductwork System resistance lower than design
5. Overload on motor	Speed too high Discharging over capacity due to existing system resistance being lower than original rating. Specific gravity or density of gas above design value. Wrong direction of rotation Bent shaft Poor alignment Wheel wedging or binding on inlet belt Bearings improperly lubricated Motor improperly wired Incorrect motor selection
6. Overheated bearings	Too much grease in bearings Poor alignment Damaged impeller or drive Dirt in bearings Abnormal end thrust Bent shaft Incorrect lubricant

Table 11: Fan Trouble Shooting and Problem Solving Guide

30. DIMENSIONAL UNITS

30.1 SI System Basic Units

The SI (International System) system of units comprises the basic units

m (meter) for length
kg (kilogram) for mass
s (second) for time
k (Kelvin) for thermodynamic temperature

30.2 Prefixes And Prefix Symbols

Name	Power to Ten	Prefix	Prefix symbol
Decimal multiples			
tenfold	10^{1}	deca	da
hundredfold	10^{2}	hecto	h
thousandfold	10^{3}	kilo	k
millionfold	10^{6}	mega	M
milliardfold	10^{9}	giga	G
billionfold	10^{12}	tera	T
Decimal fractions			
tenth	10^{-1}	deci	d
hundredth	10^{-2}	centi	c
thousandth	10^{-3}	milli	m
millionth	10^{-6}	micro	µ
milliardth	10^{-9}	nano	n
billionth	10^{-12}	pico	p
billiardth	10^{-15}	femto	f
trillionth	10^{-18}	atto	a

Table 12 : Prefixes And Symbols

The prefix symbols shown in Table 12 above should be placed directly before the unit symbol, without any space between them.

30.3 Conversion Factors

Acceleration

1 m/s^2	= ft/s^2 x 0.3048	ft/s^2	= m/s^2 x 3.281

Gravitational acceleration = 9.81 m/s^2 = 32.2 ft/s^2

Angle

radians	= degree x 0.0174533	degree	= radians x 57.29575
minute	= degree x 0.0166667	degree	= minute x 60
second	= degree x 0.0002778	degree	= second x 3600

Area

m^2	= ft^2 x 0.0929	ft^2	= m^2 x 10.7639
mm^2	= in^2 x 645.16	in^2	= mm^2 x 0.00155

Bending Moment

Nm	= lbf ft x 1.35582	lbf ft	= Nm x 0.7376

Density

kg/m^3	= lb/ft^3 x 16.02	lb/ft^3	= kg/m^3 x 0.0624
kg/m^3	= $tonne/m^3$ x 1000	$tonne/m^3$	= kg/m^3 x 0.001

Density of dry air = 1.2 kg/m^3 = 0.075 lb/ft^3 at 20°C
Density of water = 1 kg/litre = 1000 kg/m^3 = 1 $tonne/m^3$

Energy

MJ	= kwhr x 3.6	kwhr	= MJ x 0.2778
kJ	= BTU x 1.0551	BTU	= kJ x 0.9478
J	= cal x 4.187	cal	= J x 0.2388

Enthalpy

kJ/kg	= BTU/lb x 2.326	BTU/lb	= kJ/kg x 0.43
kJ/kg	= kcal/kg x 4.1868	kcal/kg	= kJ/kg x 0.239

Force

N (Newton) = kgm/s^2

1 Newton is equal to the force required to impart an acceleration of 1m/s^2 to body of mass 1 kg.

N	= lb (force) x 4.4483	lb (force)	= N x 0.2248
N	= kg (force) x 9.807	kg (force)	= N x 0.102

Length

1 micron = 10^{-3}mm = 10^{-6}m = 1 mm

mm	= in. x 25.4	in	= mm x 0.03937
mm	= ft x 304.8	ft	= mm x 0.00328
m	= ft x 0.3048	ft	= m x 3.2808
km	= mile x 1.609	mile	= km x 0.6215
m	= yd x 0.9144	yd	= m x 1.0936

Mass

kg	= lb x 0.4536	lb	= kg x 2.2046
tonne	= ton x 1.01605	ton	= tonne x 0.9842
tonne	= kg x 0.001	kg	= tonne x 1000
lb	= oz x 0.0625	oz	= lb x 16

Mass per unit length

kg/m	= lb/ft x 1.4882	lb/ft	= kg/m x 0.672

Modulus of Elasticity & Stress

MPa	= tonf/in^2 x 15.444	tonf/in^2	= MPa x 0.06475
MPa	= psi x 0.0069	psi	= MPa x 145
MPa	= N/mm^2		

Moment of Inertia

kg m^2	= lb.ft^2 x 0.04215	lb.ft^2	= kgm^2 x 23.7248

Power

kw	= hp x 0.7457	hp	= kw x 1.341

Fan absorbed power = (0.1 x m^3/s x Pa) 4 Fan efficiency in %

Pressure

$$1\text{Pa (Pascal)} = 1 \frac{N}{m^2} = 1 \frac{kg}{ms^2}$$

Absolute pressure is the pressure compared with zero pressure in empty space.

Atmospheric pressure can be expressed in various units and is equal to 1.013 bar, 1013.25 mbar, 101325 Pa, 760 Torr, 29.92 in.Hg, 14.696 psi, 407 in.w.g.

in w.g	= Pa x 0.004	Pa	= in w.g. x 249
in w.g.	= kPa x 4.016	kPa	= in w.g. x 0.249
in w.g.	= psi x 27.7	psi	= in w.g. x 0.0361
in w.g.	= mbar x 0.402	mbar	= in w.g. x 2.49
in w.g.	= mm w.g. x 0.03937	mm w.g.	= in w.g. x 25.4
kPa	= psi x 6.895	psi	= kPa x 0.145
kPa	= mm w.g. x 0.00981	mm w.g.	= kPa x 101.94
mm w.g.	= psi x 704.2	psi	= mm w.g. x 0.00142
mm Hg (Torr)	= Pa x 0.0075	Pa	= mm Hg x 133.33
mm Hg	= in w.g. x 13.6	in w.g.	= mm Hg x 0.07353
in Hg	= kPa x 0.2953	kPa	= in Hg x 3.3864
in Hg	= psi x 2.0367	psi	= in Hg x 0.491
Pa	= mbar x 100	mbar	= Pa x 0.01
Pa	= mm w.g. x 9.81	mm w.g.	= Pa x 0.102

Rotational Speed

rpm	= rev/s x 60	rev/s	= rpm x 0.01667
rpm	= rad/s x 9.551	rad/s	= rpm x 0.1047

Specific Volume

m^3/kg	= ft^3/lb x 0.06243	ft^3/lb	= m^3/kg x 16.0185

Specific volume for dry air = 0.833 m^3/kg @ 20˚C
Specific volume for water = 0.001 m^3/kg

Temperature

˚C	= (˚F - 32) x $^5/_9$	˚F	= (˚C x $^9/_5$) + 32
˚K (Kelvin)	= ˚C + 273.15	˚C	= ˚K - 273.15
˚R (Rankine)	= ˚F + 459.6	˚F	= ˚R – 459.6
˚K	= ˚R x $^5/_9$	˚R	= ˚K x $^9/_5$

˚K (Kelvin) and ˚R (Rankine) are absolute temperatures. At the absolute zero temperature of –459.6˚F or –273.15˚C, there is no molecular movement within a body.

Water boils at 100˚C (212˚F) and freezes at 0˚C (32˚F)

Did you know that -40˚C = -40˚F?

Torque

Nm	= lb.ft x 1.35582	lb.ft	= Nm x 0.7376
Nm	= lb.in x 0.113	lb.in	= Nm x 8.85

Torque in Nm = $\dfrac{\text{kw(output) x 9560}}{\text{rpm}}$ where 9560 = $\dfrac{60 \text{ x } 1000}{2p}$

Velocity

ft/s	= m/s x 3.281	m/s	= ft/s x 0.3048
ft/min	= m/s x 196.85	m/s	= ft/min x 0.00508
m/min	= m/s x 60	m/s	= m/min x 0.01667
km/hr	= m/s x 3.6	m/s	= km/hr x 0.2778
m/s	= mph x 0.44704	mph	= m/s x 2.2369
km/hr	= mph x 1.609	mph	= km/hr x 0.621

Velocity Pressure
Velocity pressure in Pa = $0.5rv^2$, where r = density, kg/m^3 ; v = velocity, m/s

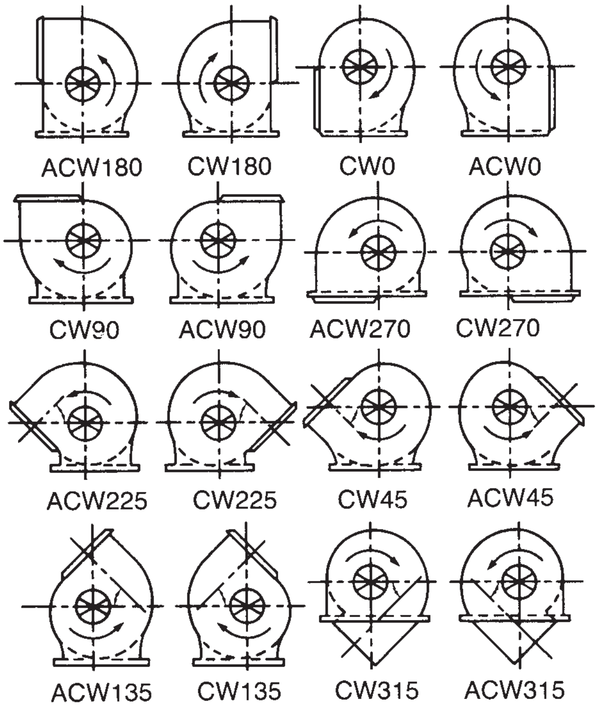
Volume

m^3	= ft^3 x 0.02832	ft^3	= m^3 x 35.3147

Volume Flowrate

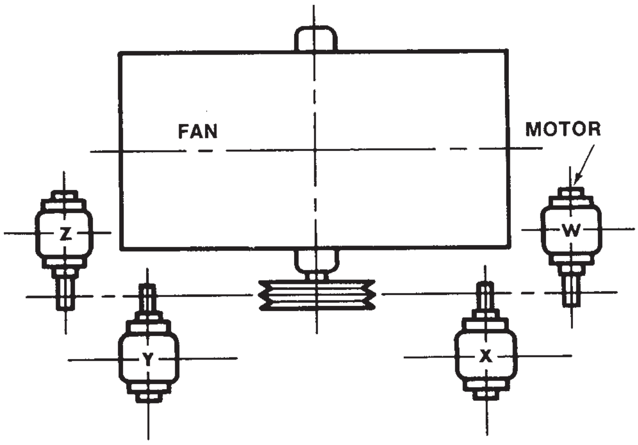
cfm	= m^3/hr x 0.5886	m^3/hr	= cfm x 1.699
cfm	= m^3/min x 35.31	m^3/min	= cfm x 0.02832
cfm	= m^3/s x 2118.6	m^3/s	= cfm x 0.000472
cfm	= l/s x 2.1186	l/s	= cfm x 0.472
cfm	= l/min x 0.0353	l/min	= cfm x 28.32

Direction of Rotation and Discharge



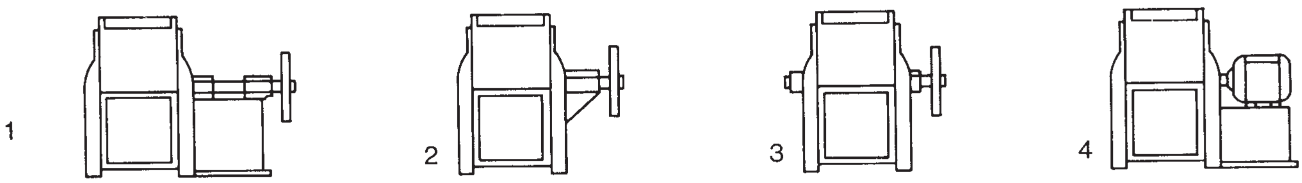
Direction of rotation is determined from the drive side. On single inlet fans, drive side is considered as opposite inlet, regardless of actual drive location.

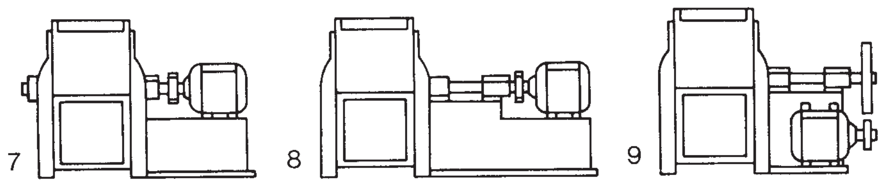
STANDARD MOTOR POSITIONS



The location of motor is determined from plan view of the blower, designating the motor position by letters W, X, Y and Z as the case may be.

ARRANGEMENTS OF DRIVE





ARRANGEMENT No. 7, SWSI AND DWDI
For belt drive or direct connection.
Arrangement No. 3 plus base for motor.

ARRANGEMENT No. 1, SWSI
For belt drive or direct connection. Wheel overhung. Two bearings on base.

ARRANGEMENT No. 3, SWSI AND DWDI
For belt drive or direct connection. One bearing on each side and supported by fan housing.

ARRANGEMENT No. 8, SWSI
For belt drive or direct connection.
Arrangement No. 1 plus base for motor.

ARRANGEMENT No. 2, SWSI
For belt drive or direct connection. Wheel overhung. Bearings in bracket supported by fan housing.

ARRANGEMENT No. 4, SWSI
For direct drive. Wheel overhung on motor shaft. No bearings on fan. Base mounted or an integrally direct connected motor.

ARRANGEMENT No. 9, SWSI
For belt drive Arrangement No. 1 designed for mounting prime mover on side of base.

Table2: Fan Rotation, Arrangment and Motor Position