Engineering Notes

The following section includes engineering and technical data, guidelines and system explanations related to air moving and control devices. Fan laws and system descriptions are consistent with industry standards, definitions and accepted practices. It is provided to assist system designers in sizing, selecting and defining their air moving and control systems as well as explaining variables inherent in system design.

Flow and Static Pressure

For any change in static pressure (SP), a squared relationship is applied to the flow ratio. This is expressed by the formula:

$$\frac{P_2}{P_1} = \left(\frac{CFM_2}{CFM_1}\right)^2$$

Where P_1 is the original static pressure,
P_2 is the desired static pressure,
CFM_1 is the original flow rate in cu.ft. per minute, and
CFM_2 is the desired flow rate in cu.ft. per minute.

This formula is based upon performance of a fan at one point on a system. This data can be used to calculate a fan performance curve indicative of all points from 0" SP (maximum flow) to maximum SP (0 flow).

$$\frac{CFM_2}{CFM_1} = \frac{RPM_2}{RPM_1} \qquad \frac{P_2}{P_1} = \left(\frac{CFM_2}{CFM_1}\right)^2 \qquad \frac{HP_2}{HP_1} = \left(\frac{CFM_2}{CFM_1}\right)^3$$

$$\frac{CFM_2}{CFM_1} = \frac{RPM_2}{RPM_1} \qquad \frac{P_2}{P_1} = \left(\frac{RPM_2}{RPM_1}\right)^2 \qquad \frac{HP_2}{HP_1} = \left(\frac{RPM_2}{RPM_1}\right)^3$$

Air Systems

An air system may consist simply of fan with ducting connected to either the inlet or discharge or to both. A more complicated system may include a fan, duct-work, air control dampers, cooling coils, heating coils, filters, diffusers, noise attenuators, turning vanes, etc. The fan is the component in the system which provides energy to the airstream to overcome the resistance to flow of the other components.

Component Losses

Every system has a combined resistance to flow which is usually different from every other system and is dependent upon the individual components in the system.

The determination of the “pressure loss” or “resistance to flow,” for the individual components can be obtained from the component manufacturers. The determination of pressure losses for ductwork and branch piping design is well documented in standard handbooks such as the ASHRAE Handbook of Fundamentals and SMACNA Duct Design Manual.

The System Curve

At a fixed volume flow rate (CFM) through a given air system a corresponding pressure loss, or resistance to this flow, will exist. If the flow rate is changed, the resulting pressure loss, or resistance to flow, will also change. The relationship governing this change for most systems is:

PRESSURE C/ PRESSURE = (QC/Q)2

Q = CFM   C = change

Interaction of the System Curve and the Fan Performance Curve

If the system characteristic curve, composed of the resistance to flow of the system and the appropriate System Effect Factors have been accurately determined, then the fan selected will develop the equivalent and necessary pressure to meet the system requirements; i.e., the fan will deliver the designated flow rate when installed in the system.

The point of intersection of the system curve and the fan performance curve determines the actual flow volume. If the system resistance has been accurately determined and the fan properly selected, their performance curves will intersect at the design flow rate. Refer to Figure 1. The normalized Duct System A from Figure 1 has been plotted with a normalized fan performance curve.

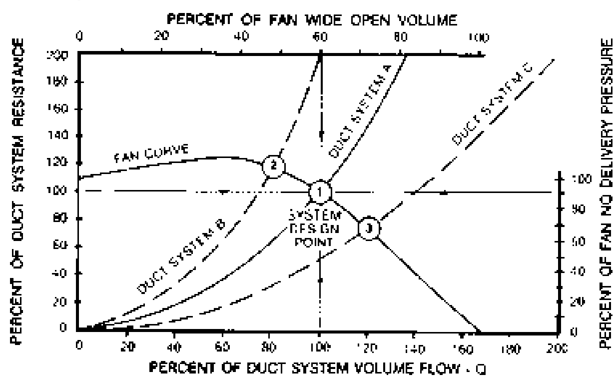


Figure 1. Interaction of System Curves and Fan Curve

The volume flow rate through the system in a given installation may be varied by changing the system resistance. This is usually accomplished by using fan dampers, duct dampers, mixing boxes, terminal units, etc. Figure 1 shows the volume flow rate may be varied from 100% design Q (Point 1, Duct System A), to approximately 80% of the design Q by increasing the resistance to flow, thus changing the system curve characteristics to Duct System B. This results in fan operation at Point 2 (the intersection of the fan curve and the new Duct System B). Similarly, the volume flow rate can be increased to approximately 120% of the design Q by decreasing the resistance to flow, thus changing the system curve characteristic to Duct System C. This results in fan operation at Point 3 (the intersection of the fan curve and the new Duct System C).

Engineering Notes

Effect of Changes in Speed

Increases or decreases in fan speed will alter the volume flow rate through a system. Figure 2 illustrates the increase in flow rate when the fan speed increases 10% to Point 2. The 10% increase in flow rate, however, extracts a severe power penalty. According to the fan laws (see below), the power increase is 33%. This fact is often startling to the system designer who finds a flow deficiency. Only 10% more air is needed but the connected motor horsepower is not capable of a 33% increase in load. (Note that the increased power requirements are the result of increased work done).

The greater air volume flow rate moved by the fan against the resulting higher system resistance to the flow is a measure of the increased work done. In the same system, the power increases as the cube of the speed ratio; the fan efficiency remains the same at all points on the same system curve.

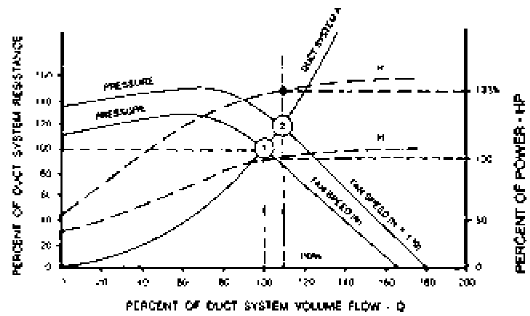


Figure 2. Effect of 10% Increase in Fan Speed

Effect of Change in Speed (Fan Size and Gas Density Remaining Constant)

For the same size fan, D_c = D and, therefore, $(D_c/D)=1$. When the density does not vary, $p_c = p$ and the density $(p_c /p) = 1$.

$$Q_c = Q\ (RPM_c / RPM)$$
$$P_{tc} = P_t\ (RPM_c / RPM)2$$
$$P_{sc} = P_s\ (RPM_c / RPM)2$$
$$P_{vc} = P_v\ (RPM_c / RPM)2$$
$$HP_c = HP_c\ (RPM_c / RPM)3$$

D = diameter	c = change
HP = horsepower	t = total
Q = CFM	s = static
P = static pressure	v = velocity
p = density	

Effect of Density on System Resistance

The resistance of a duct system is dependent upon the density of the gas flowing through the system. A gas density of 0.075 lb/ft^3 is standard in the fan industry. Figure 3 illustrates the effect on the fan performance of a density variation from the standard value.

The pressure and horsepower vary directly as the ratio of the gas density at the fan inlet to standard density. This density ratio must always be considered when selecting fans from manufacturers' catalogs or curves.

Effect of Change on Density (Fan Size and Speed Remaining Constant)

When the speed of the fan does not change, RPM_c = RPM and, therefore, $(RPM_c / RPM) = 1$. The fan size is also fixed, D_c = D and therefore $(D_c / D) = 1$.

$$Q_c = Q$$
$$P_{tc} = P_t\ (p_c / p)$$
$$HP = HP\ (p_c / p)$$
$$P_{sc} = P_s\ (p_c / p)$$
$$P_{vc} = P_v\ (p_c / p)$$

D = diameter	c = change
HP = horsepower	t = total
Q = CFM	s = static
P = static pressure	v = velocity
p = density	

NOTE: PennBarry's Fansizer software can make density corrections for you.

Table 1: Air Density Ratios

AIR TEMP. °F	ALTITUDE IN FEET ABOVE SEA LEVEL												
	0	1000	2000	3000	4000	5000	6000	7000	8000	9000	10000	15000	20000
	BAROMETRIC PRESSURE IN INCHES OF MERCURY												
	29.92	28.86	27.82	26.82	25.84	24.9	23.98	23.09	22.22	21.39	20.58	16.89	13.75
70	1.000	0.964	0.930	0.896	0.864	0.832	0.801	0.772	0.743	0.714	0.688	0.564	0.460
100	0.946	0.912	0.880	0.848	0.818	0.787	0.758	0.730	0.703	0.676	0.651	0.534	0.435
150	0.869	0.838	0.808	0.770	0.751	0.723	0.696	0.671	0.646	0.620	0.598	0.490	0.400
200	0.803	0.774	0.747	0.720	0.694	0.668	0.643	0.620	0.596	0.573	0.552	0.453	0.369
250	0.747	0.720	0.694	0.669	0.645	0.622	0.598	0.576	0.555	0.533	0.514	0.421	0.344

Note: This table provides air density adjustment factors, so fans can be selected to account for non-standard density. Unity Basis = Standard Air Density of .075 lb/ft^3. At sea level (29.92 in. HG barometric pressure) this is equivalent to dry air at 70°F.

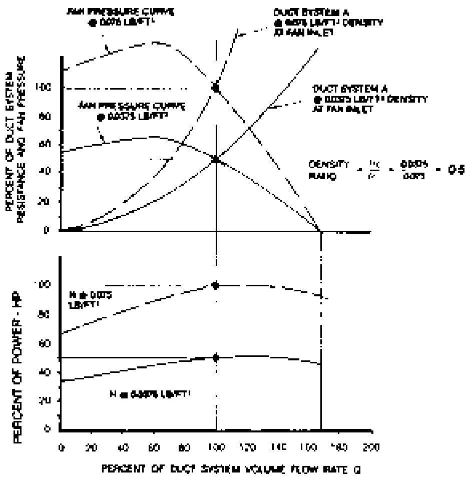


Figure 3. Density Effect

Effects of Errors in Estimating System Resistance

Higher System Resistance. Actual Duct System B in Figure 4 shows a situation where an actual system has more resistance to flow than was calculated. This condition is generally the result of an inaccurate estimate of system resistance to flow. All losses must be considered when calculating system pressure losses or the final system will be more restrictive than designed; the actual flow rate will be less than expected, (Point 2). If the actual duct system pressure loss is greater than design, an increase in fan speed may be necessary to achieve Point 5, the design volume flow rate. Before increasing fan speed, check with the fan manufacturer to determine if the speed can be safely increased. Also determine the expected increase in horsepower: power will increase as the cube of the speed and it is very easy to exceed the capacity of the connected motor and even the available electrical source.

Lower System Resistance. Actual duct system C in Figure 4 shows a situation where a system has less resistance to flow than was expected; the actual flow rate will be more than expected, (Point 3).

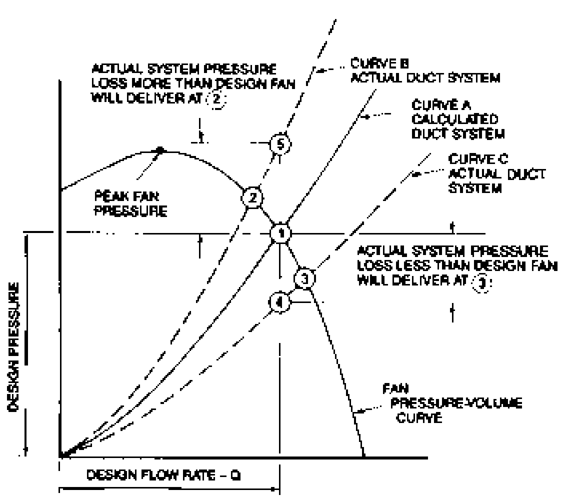


Figure 4. Fan/Duct System Curve not at Design Point

System Effect

Figure 5 illustrates deficient fan/system performance resulting from one or more undesirable flow conditions. It is assumed that the system pressure losses, shown in system curve A, have been accurately determined, and a suitable fan selected for operation at Point 1. However, no allowance has been made for the effect of the system connections on the fan's performance. To compensate for this System Effect it will be necessary to add a System Effect Factor (SEF) to the calculated system pressure losses to determine the actual system curve. The SEF for any given configuration is velocity dependent and will, therefore, vary across the range of flow volumes for the fan.

In Figure 5 the point of intersection between the fan performance curve and the actual system curve B is Point 4. The actual flow volume will, therefore, be deficient by the difference form 1-4. To achieve design flow volume a SEF equal to the pressure difference between Point 1 and 2 should have been added to the calculated system pressure losses and the fan selected to operate at Point 2. Note that because the System Effect is velocity rated, the difference represented between Points 1 and 2 is greater than the difference between Points 3 and 4.

The SEF includes only the effect of the system configuration on the fan's performance.

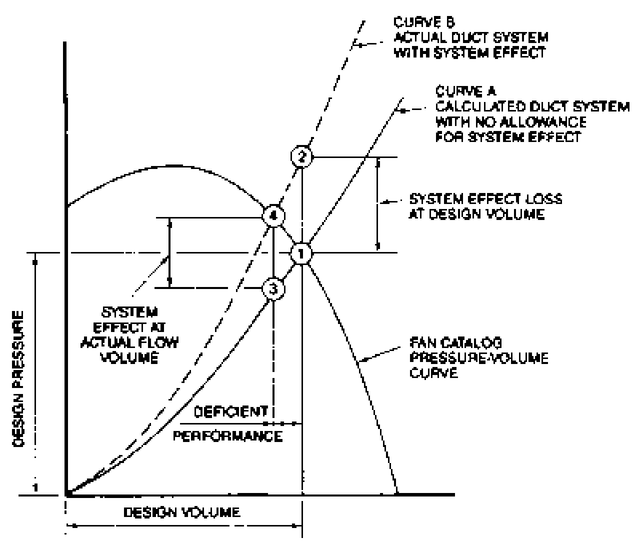


Figure 5. Deficient Fan/Duct System Performance, System Effect Ignored

System Effect Factor

A System Effect Factor is a pressure loss which recognizes the effect of fan inlet restrictions, fan outlet restrictions, or other conditions influencing fan performance when installed in the system.

Engineering Notes

Outlet System Effect Factors

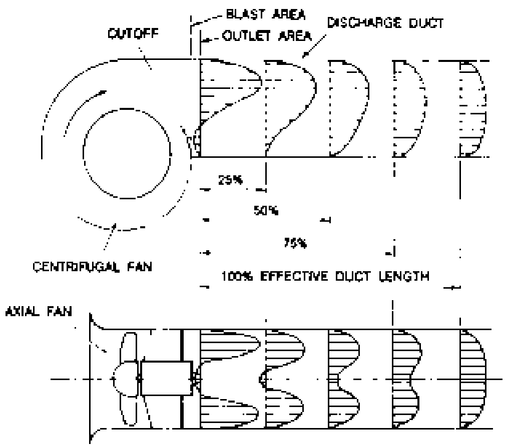


Figure 6. Fan Outlet Velocity Profiles

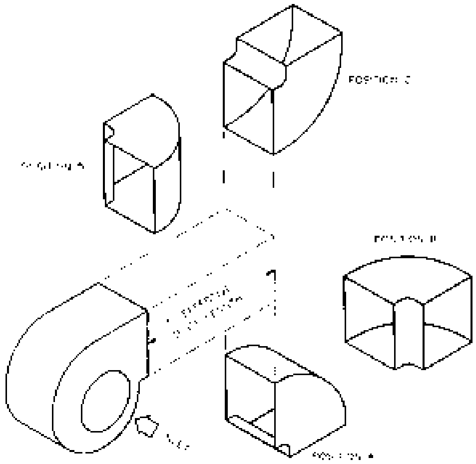
Outlet Ducts

To calculate 100% effective duct length, assume a minimum of 21/2 duct diameters for 2500 FPM or less. Add 1 duct diameter for each additional 1000 fpm.

Example: 5000 FPM=5 equivalent duct diameters. If the duct is rectangular with side dimensions a and b, the equivalent duct diameter is equal to (4ab/p)0.5

Controlled diffusion and establishment of a uniform velocity profile in a straight length of outlet duct

Centrifugal Fans Outlet Duct Elbows. The outlet velocity of centrifugal fans is generally higher toward one or adjacent sides of the rectangular duct. If an elbow must be located near the fan outlet it should have a minimum radius to duct diameter ratio of 1.5, and should be arranged to give the most uniform airflow possible. Refer to Figure 7.



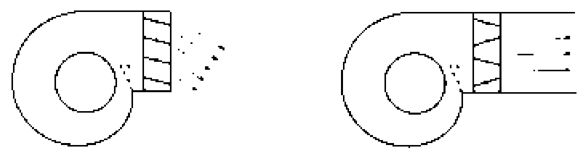
SWSI CENTRIFUGAL FAN SHOWN

Figure 7. Outlet Elbows on SWSI Centrifugal Fans

Turning Vanes. Turning vanes will usually reduce the pressure loss through an elbow. However, where a non-uniform approach velocity profile exists, such as at a fan outlet, the vanes may actually serve to continue the non-uniform profile beyond the elbow. This may result in increased losses in other system components downstream of the elbow.

Volume Control Dampers. Volume Control Dampers are manufactured with either "opposed" blades or "parallel" blades. When partially closed, the parallel bladed damper diverts the airstream to the side of the duct. This results in a non-uniform velocity profile beyond the damper and flow to branch ducts close to the downstream side may be seriously affected.

The use of an opposed blade damper is recommended when volume control is required at the fan outlet and there are other system components, such as coils or branch takeoffs downstream of the fan. When the fan discharges into a large plenum or to free space a parallel blade damper may be satisfactory. Refer to Figure 8.



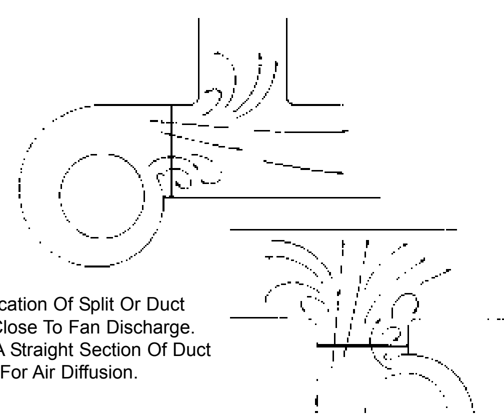
Parallel Bladed Damper Illustrating Diverted Flow

Opposed Bladed Damper Illustrating Non-Diverted Flow

Figure 8. Parallel Blade vs. Opposed Dampers

Duct Branches. Standard procedures for the design of duct systems are all based on the assumption of uniform flow profiles in the system.

In Figure 9 branch takeoffs or splits are located close to the fan outlet. Non-uniform flow conditions will exist and pressure loss and airflow may vary widely from the design intent. Wherever possible a length of straight duct should be installed between the fan outlet and any split or branch takeoff.



Avoid Location Of Split Or Duct Branch Close To Fan Discharge. Provide A Straight Section Of Duct To Allow For Air Diffusion.

Figure 9. Branches Located Too Close to Fan

Inlet System Effect Factors

Fan inlet and non-uniform inlet flow can often be corrected by inlet straightening vanes or guide vanes. Restricted fan inlets located too close to walls, obstructions or restrictions caused by a plenum or cabinet will decrease the useable performance of a fan. Cabinet clearance effect or plenum effect is considered a component part of the entire system; the pressure losses through the cabinet or plenum must be considered as a System Effect when determining system characteristics.

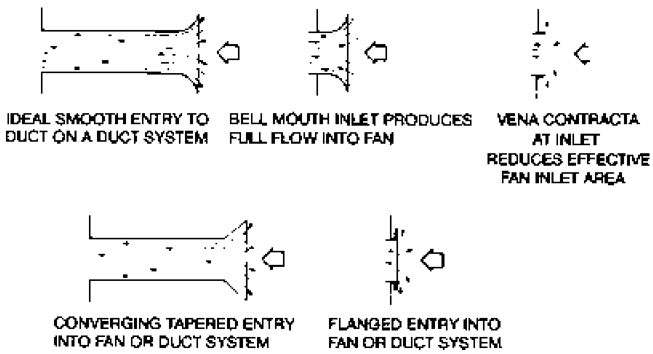


Figure 10. Typical Inlet Connections for Centrifugal and Axial Fans

Inlet Duct Elbows

Non-uniform flow into a fan inlet is the most common cause of deficient fan performance. An elbow located at, or in close proximity to the fan inlet will not allow the air to enter the impeller uniformly. The result is less than catalogued air performance.

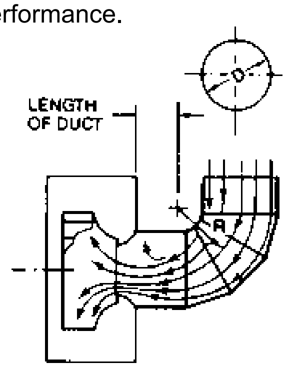


Figure 11a. Non-Uniform Flow into a Fan Inlet Induced by a 90°, 3-Piece Section Elbow – No Turning Vanes

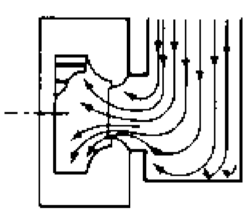


Figure 11b. Non-Uniform Flow Induced Into Fan Inlet by a Rectangular Inlet Duct

Inlet Vortex (Spin or Swirl)

Another cause of reduced performance is an inlet duct condition that produces a vortex or spin in the airstream entering a fan inlet. An example of this condition is illustrated in Figure 12.

The ideal inlet condition is one which allows the air to enter axially and uniformly without spin in either direction. A spin in the same direction as the impeller rotation (pre-rotation) reduces the pressure volume curve by an amount dependent upon the intensity of the vortex. The effect is similar to the change in the pressure volume curve achieved by inlet vanes installed in a fan inlet; the vanes induce a controlled spin the direction of impeller rotation reducing the volume flow rate.

A counter-rotating vortex at the inlet may result in a slight increase in the pressure-volume curve but the horsepower will increase substantially.

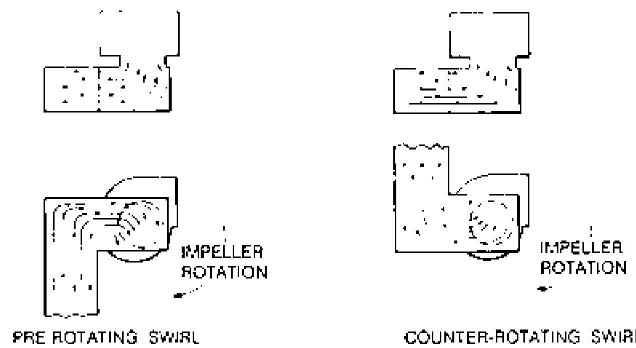


Figure 12. Inlet Duct Connections Causing Inlet Spin

INLET TURNING VANES

Where space limitations prevent the use of optimum fan inlet connections, more uniform flow can be achieved by the use of turning vanes in the inlet elbow (see Figure 13).

Numerous variations of turning vanes are available from a single curved sheet metal vane to multi-bladed "airfoil" vanes.

The pressure drop through these devices must be added to the system pressure losses.

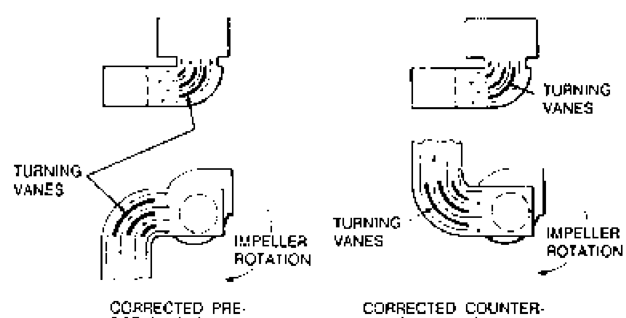


Figure 13. Inlet Turning Vanes