Naval Facilities Engineering Command
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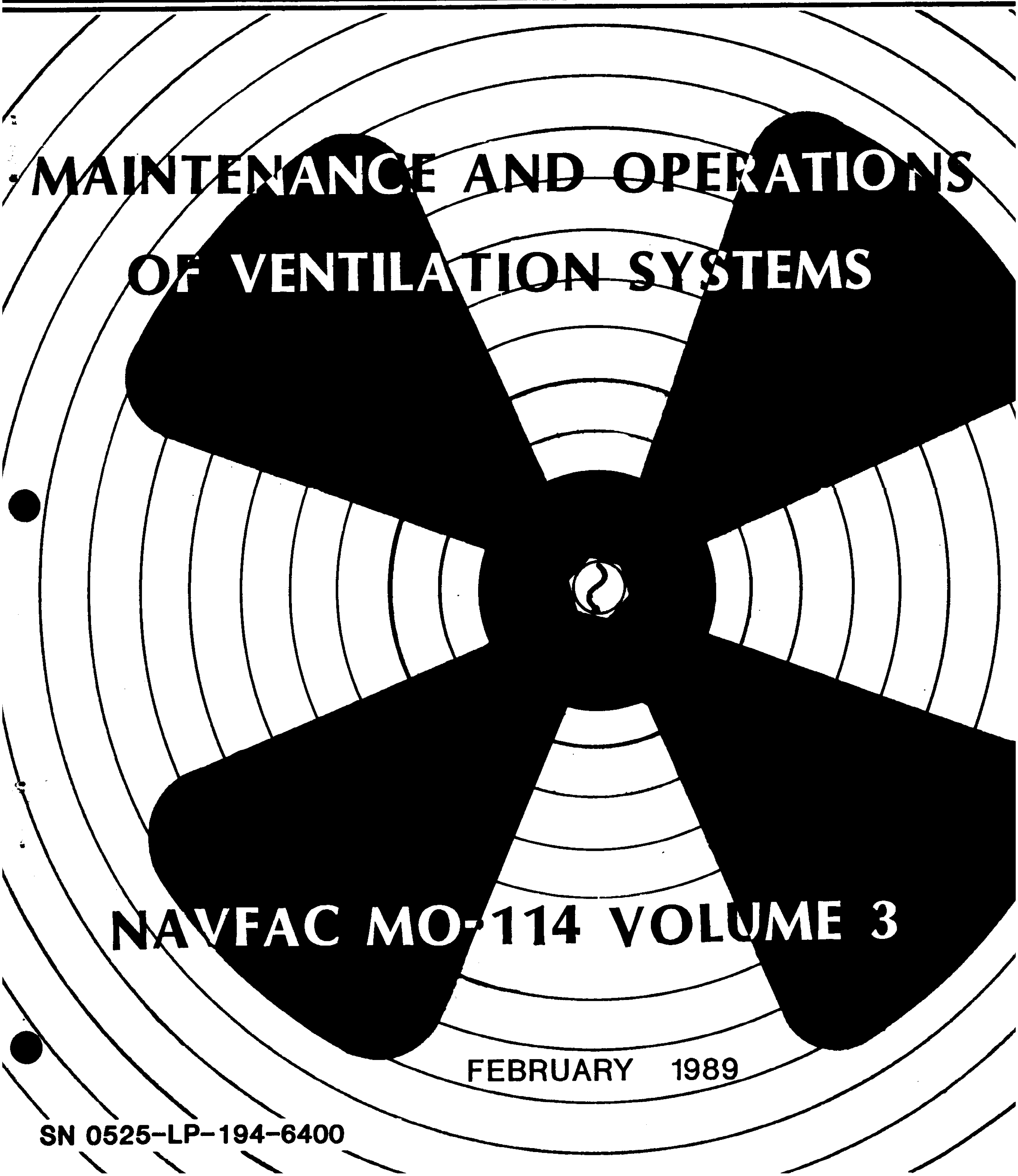
MAINTENANCE AND OPERATIONS OF VENTILATION SYSTEMS



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ABSTRACT

NAVFAC MO-114, Volume 3, provides practical information on the operation and maintenance of mechanical, natural or gravity and industrial ventilation systems. Mechanical ventilation cooling systems and maintenance instructions are covered as well. It is primarily directed to the personnel in the field who actually supervise and perform the operations and maintenance work. The data contained in the manual originated at the activity level and represents actual experiences of field personnel.

FOREWORD

This publication provides practical information on the operation and maintenance of ventilating systems with special emphasis on economy consistent with safety. It is primarily directed to the personnel in the field who actually supervise and perform the operations and the maintenance work. Although the general subject of ventilating systems is highly technical, this publication has been written in nontechnical language, brief and direct, so that the reader will have the basic information required for the intelligent handling of field situations.

This publication has been written not only to be read, but more importantly, to be used. To obtain maximum benefit, it should be consulted together with the equipment manufacturers' instruction manuals, parts lists, and drawings. Much of the data contained herein has originated at activity level and represents actual experience of field personnel. In addition, the latest information on operation and maintenance procedures has been collected and included.

This publication supersedes MO-114, "Plumbing, Heating and Ventilation," Chapter 3, dated April 1964. Recommendations or suggestions for modification or addition that will improve the publication and motivate or extend its use are invited and should be submitted through appropriate channels to the Commander, Naval Facilities Engineering Command (Attention: Code 163, Alexandria, VA 22332-2300.

This publication is certified as an official publication of the Naval Facilities Engineering Command and has been reviewed and approved in accordance with SECNAVINST 5600.16, Procedures Covering Review of the Department of the Navy Publications.

ROBERT E. HAMMOND
Deputy Assistant Commander for
Public Works Centers and Departments

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CHAPTER 1. GENERAL

1.1 INTRODUCTION. Preventive maintenance service and inspection frequencies for various components are shown in appendix A.

1.2 MANUFACTURER INSTRUCTIONS. Maintenance personnel should have available copies of manufacturers' instructions for each separate type of equipment. When new equipment is installed make sure you file the manufacturers instructions in the maintenance department. It is extremely important, for example, to use the correct lubricants on fans and motors.

1.3 CODES. Motors for fans and blowers must meet requirements of the National Board of Fire Underwriters. All reliable manufacturers of such equipment test their products in accordance with the standard test code for fans.

1.4 TOOL REQUIREMENT. The job of the maintenance mechanic consists of performing basic mechanical operations using common tools and materials of the trade. Three principles should be followed in performing the operation:

1. Know what needs to be done.
2. Select the proper tools and materials.
3. Follow approved safety procedures.

The following standard tools should be available to the maintenance mechanic:

1. Standard and Phillips head screwdrivers - various sizes
2. Pliers: vise grip, slipjoint, needlenose, diagonal, cutting pliers, side cutters
3. Ball peen hammer
4. Hacksaw and spare blades
5. 3/8-inch drive socket set and ratchet
6. Small set of Allen wrenches
7. Assorted center punches, drift punches, steel chisel
8. 12-foot measuring tape
9. Crescent wrenches 4 to 14 inches
10. Open end and box end wrenches l/4 to 3/4 inch
11. File
12. Pipe wrenches to 24 inches
13. Small level and square
14. Pocket knife
15. Flashlights
16. Grease guns and oilers
17. Wire brush

18. Extension cord and inspection lights
19. Various cleaning tools - brushes, scrapers, etc.
20. Emery cloth
21. Thermometer
22. Tubing and pipe cutters
23. Flaring tool
24. Small acetylene outfit
25. Package kit and packing
26. 1/2-inch drive socket set
27. Strap wrench
28. Tachometer
29. Amp meter - clamp on
30. Vacuum - cleaning equipment and materials
31. Velocity measuring equipment
32. Respirator
33. Eye Protection

1.5 SAFETY. Maintenance personnel have responsibility for the safety of others in keeping ventilating equipment in operating condition. During temporary inoperative periods of ventilating systems, you may be exposed to the effects of noxious or explosive gases and fumes without realizing their danger. All concerned should be warned of the hazards involved. Equipment should be put back into operation as promptly as possible.

The following general safety precautions should be observed by the maintenance mechanic working with tools or equipment:

1. Dropping heavy objects on feet or toes is a hazard. This can be avoided by using proper trucking and hoisting equipment. The operator should wear safety shoes with metal tips to protect toes.

2. Back injuries may be caused by attempting to lift heavy objects and by not using arm and leg muscles correctly.

3. Good housekeeping is very important. Keep work area clean. Keep oil and water off the floor.

4. "Mushroom" heads should be ground from chisels and punches as these particles may fly when struck with the hammer causing serious injury to the operator or a bystander.

5. Files should never be used without handles. The tang may injure the hands.

6. Wear goggles when drilling because chips may fly. Eyes should always be protected.

7. Never use a hammer to pound on a screwdriver or use a screwdriver as a punch or chisel.

8. Never stroke a hacksaw over 60 strokes per minute.

9. Never use pliers on parts designed to be used with wrenches.

10. Always pull on a wrench instead of pushing.

11. Avoid pounding on a wrench or using "cheater bars" to obtain greater turning torque.

12. Wrenches should always fit snugly. Poorly fitted wrenches will ruin nut and bolt heads. They may slip and cause injury to the service mechanic.

13. Always "crack" valves before opening.

14. Always have good lighting and good ventilation.

15. Never use gasoline or other flammable materials when cleaning.

16. Never use carbon tetrachloride for cleaning. Its effects are cumulative in the body.

17. Many components are quite fragile. Parts may be ruined by over-tightening nuts and bolts, not tightening them in the correct order, or using the wrong size wrench.

18. Always disconnect the electrical circuit or make sure that electrical devices are safe before starting on a job. An electrical short across a ring or wrist watch can cause a severe burn. It is best to remove rings and wrist watches when working on electrical equipment.

19. Many electrical shocks occur when the service technician comes in contact with an electrical current and a ground. Avoid working on any electrical circuit if standing on a damp floor or if one hand is touching a water pipe.

20. Do not wear gloves, ties, or loose clothing around moving machinery.

21. Do not service parts while the equipment is operating, unless guards or other adequate protection are provided.

22. Before repairing controls and motors, open the electrical switches to them. Tag or lock-open the switches to prevent accidents or short circuits. Wear goggles during grinding operations.

In addition to the aforementioned safety precautions, specific safety requirements for equipment or system conditions will be identified throughout the text where appropriate.

Safety precautions, safe maintenance practices, and safety policy are covered in detail in NAVFAC Instruction 5100.11. All maintenance personnel should be familiar with the contents of this instruction.

CHAPTER 1. SELF-STUDY QUESTIONS

Q1-1 Safety and careful working habits are best carried out by:

() a. Oneself
() b. The supervisor
() c. Following your co-workers example
() d. Deep contemplation

Q1-2 Sharp pointed tools should be carried;

() a. With the point up
() b. In the hip pocket
() c. With the point down and away from you
() d. With the point straight ahead

Q1-3 When striking or cutting, be sure chips fly away from you and others because:

() a. No one likes chips in their clothing
() b. There is danger of personal injury from the chips
() c. Of the loss of metal being carried away
() d. Of the financial cost of the lost metal

Q1-4 Spilled grease or oil must be cleaned up:

() a. When it happens
() b. At quitting time
() c. When the foreman finds it
() d. At your convenience

Q1-5 Oily, greasy, and paint-filled rags must be stored for disposal in:

() a. Any old box
() b. Self-closing metal containers
() c. Your personal locker

Ql-6 The safety guards placed on machinery for the protection of the operator are to be removed:

() a. When the operator thinks it is unnecessary
() b. When it gets in the way
() c. When efficiency is greater without it
() d. At no time

Q1-7 The use of goggles when operating certain types of equipment is necessary:

() a. To protect the eyes
() b. To make vision more difficult
() c. To increase sales of goggles
() d. To reduce the safety factor

Ql-8 When a fire occurs, it is very important to:

() a. Keep your head and attempt to extinguish it
() b. Protect the safety of all individuals
() c. Notify the fire department
() d. Do all of the above

Ql-9 Keep all combustibles from the immediate areas where fires can be started such as:

() a. Welding areas
() b. Trash-burning areas
() c. Boiler rooms
() d. All of the above

Ql-10 All flammables, such as gasoline, paints, oils, kerosene and cleaning fluids, should be kept in:

() a. Closed labeled containers in a special metal locker or building
() b. A container closest to the areas to be used
() c. A place under the building

Q1-11 The best way to prevent fires is to:

() a. Be safety conscious and observe the rules at all times
() b. Keep all valves and switches open at all times
() c. Never use noninflammable liquids
() d. Call the fire department each time you use a flammable material

Q1-12 Hand tools should never be used which are:

() a. Oily and greasy
() b. Heavy and cumbersome
() c. Checked out to another person
() d. Usable only one time

Q1-13 A screwdriver should never be used as a

() a. Chisel
() b. Punch
() c. Pry bar
() d. All of the above

Ql-14 When using a screwdriver, the stock or material worked on should never be:

() a. Hand held
() b. Case hardened
() c. Vise held
() d. Oiled with penetrating oil

Ql-15 Avoid jamming or locking the blade of a hacksaw in the work piece to prevent possible injury by:

() a. Shattering the material
() b. Fracturing the handle
() c. Flying pieces of broken blade
() d. Separation of the locking device

Q1-16 Small pieces of material to be ground or buffed should be:

() a. Hand held
() b. Held by the foreman
() c. Mechanically held by vise grips or similar devices

CHAPTER 2. MECHANICAL VENTILATION SYSTEMS

2.1 GENERAL. Mechanical ventilation systems at Navy installations supply fresh air and remove heat, dust, toxic gases, fumes, and odors. They usually consist of an opening for air intake, and one or more outlets for discharging unsuitable air. Air intakes and outlets are louvered for controlling air during cold weather, and screened to keep out birds, insects, and rodents. Mechanical systems are usually provided with ducts for distribution of fresh air and for exhausting air from rooms or other spaces, or they may be roof- or wall-type ventilators without ducts. Fans may be located at any point or points in the system, depending on accessibility and economy of space. Sometimes they are placed near the fresh air entry, and at other times, they are located in attics or in roof monitors to pull undesirable air from the building.

This chapter describes the movement of air through duct systems. It will review the types of fans used, the application of each type, ductwork, the relationship between fans and ducts, and room air distribution.

2.2 FAN AND DUCT SYSTEMS. Figure 2-l shows a simple fan and duct system, and the pressures that might be found within various parts of that system. The room, at point A, is at atmospheric or neutral pressure. The fan creates a suction in the return air duct which draws the air out of the room. The lowest suction pressure exists at point B, just at the inlet to the fan.

As the air passes through the fan, the fan imparts velocity and pressure to the air. At point C (the fan discharge), the air has its highest total pressure of any place in the system. Because the pressure is higher at point C than in the room, the air flows from the fan discharge into the room, However, on its way, the air is constantly losing some energy due to friction against the duct walls, This results in a constantly falling static pressure.

The amount of air delivered through the duct system depends upon the performance characteristics of the fan used, Conversely, the amount of air delivered by a given fan will depend upon the size and the configuration of the duct system and its components, The following sections will describe these relationships.

2.3 FAN CURVE. The fan curve is a graph that describes how the performance of a fan will vary when the fan pushes air through any system. The apparatus shown in Figure 2-2 may be used by the fan manufacturer to create a fan curve for any fan.

With the damper in the duct wide open, the fan will deliver its maximum cubic feet per minute (cfm). The static pressure at the fan outlet will be quite low due to the increase in the velocity flow of the air. The manufacturer measures the airflow and the increase in static pressure being produced across the fan. This will be used as one data point on the fan curve.

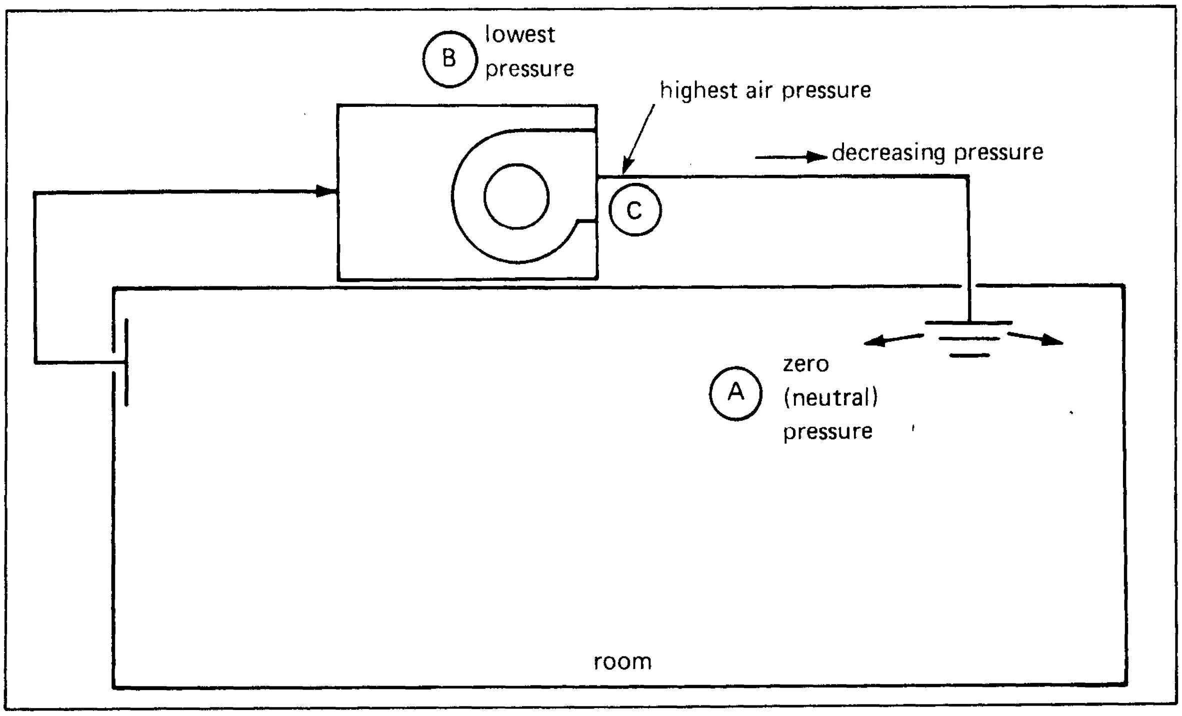


FIGURE 2-1
Typical Air Pressures in an Air Distribution System

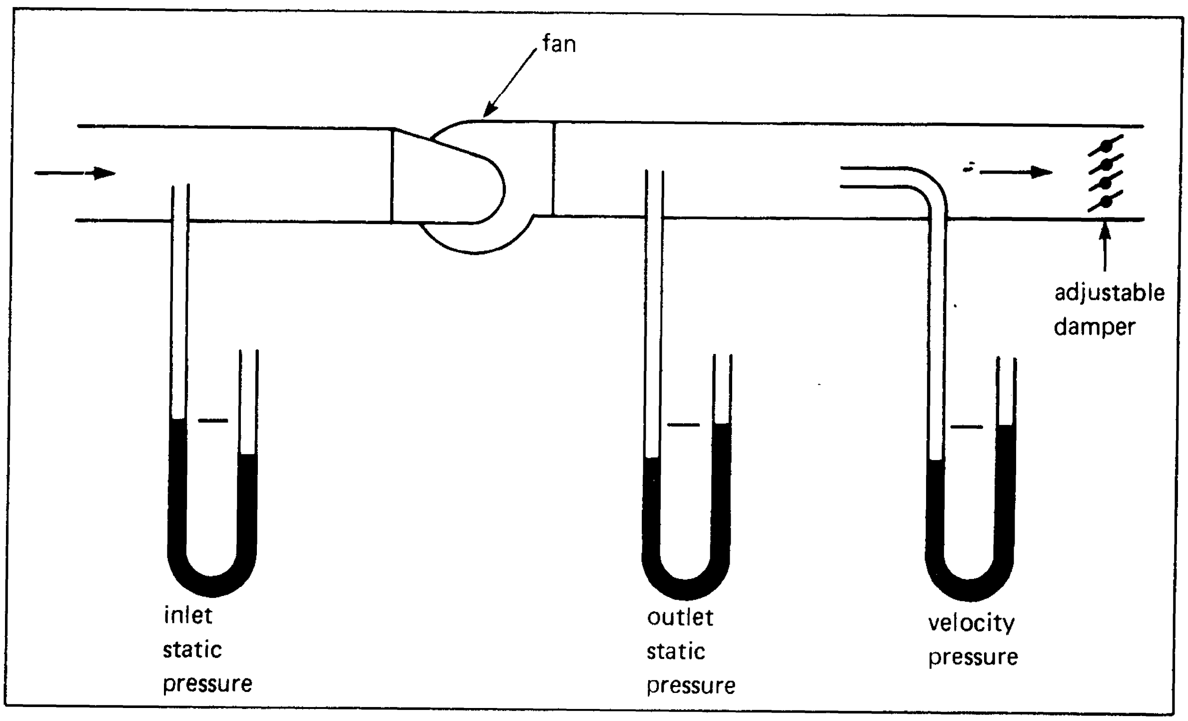


FIGURE 2-2
Apparatus for Testing Fan Performance

Next, the duct damper is moved to a slightly closed position. The additional friction through the damper tends to decrease the air flow somewhat, causing the static pressure at the fan discharge to go up slightly. In developing the fan curve data, for each different setting of the duct damper, the static pressure across the fan and the airflow are measured. When this data is plotted on a graph, we have a fan curve (Figure 2-3). The service technician may use the fan curve in order to determine the airflow in an operating system. By measuring the static pressure at the fan inlet and the fan outlet and calculating the change in static pressure (Δ sp), the fan curve may be used to read the cfm.

We can create another fan curve for the same fan, but this time we change the motor or the drive so that the fan operates at a slower velocity, revolutions per minute (rpm). When we plot these results on the same fan curve as the original, we find that the new curve is similar in shape to the original, but the flow in cfm and static pressure are lower. Figure 2-4 shows a family of fan curves for one fan at many different rpm settings.

Table 2-1 shows a multirating table for a fan. It presents the same information as that contained in a fan curve, but some prefer the tabulated form instead of the graph presentation.

2.4 SYSTEM CURVES. For any duct system, the flow of air through the ductwork will experience a pressure drop caused by the friction with the duct walls. Elbows, changes in duct size or direction, fittings, coils, filters, dampers, and any other devices which the air must pass through will cause additional pressure drop.

The amount of friction depends upon the velocity of the air. For a given system, more air being pushed through the ductwork will result in a larger pressure drop, and will require higher fan capacity to accomplish the task. Figure 2-5 shows the relationship between the air flow through one specific duct system and the pressure drop that the air will experience. This is called a system curve. The pressure drop of the air due to friction is proportional to the square of the velocity.

In practical terms, consider a duct system which is carrying some 1,200 cfm, and has a system pressure drop of 1.5 inches water column. If you want to push twice as much air through the system, it will experience a pressure drop of four times as much, or 6.0 (i.e. 4 X 1.5) inches water column.

Example: A duct is carrying 2,000 cfm. The fan moving the air through the system is overcoming a duct friction loss of 1.2 inches water column. How much static pressure loss will the fan need to overcome if you want to increase the airflow to 2,300 cfm?

Solution: The ratio of the new cfm to the old cfm is (2,300/2,000) which equals 1.15. The ratio of the new duct velocity to the old duct velocity will also be 1.15. The static pressure loss through the duct will increase with the square of the velocity. It will increase by a factor of $(1.15)^2 = 1.32$. The old pressure drop must be multiplied by a factor of 1.32. The new pressure drop is:

SP loss = (1.2) x (1.32) = 1.58 inches water column

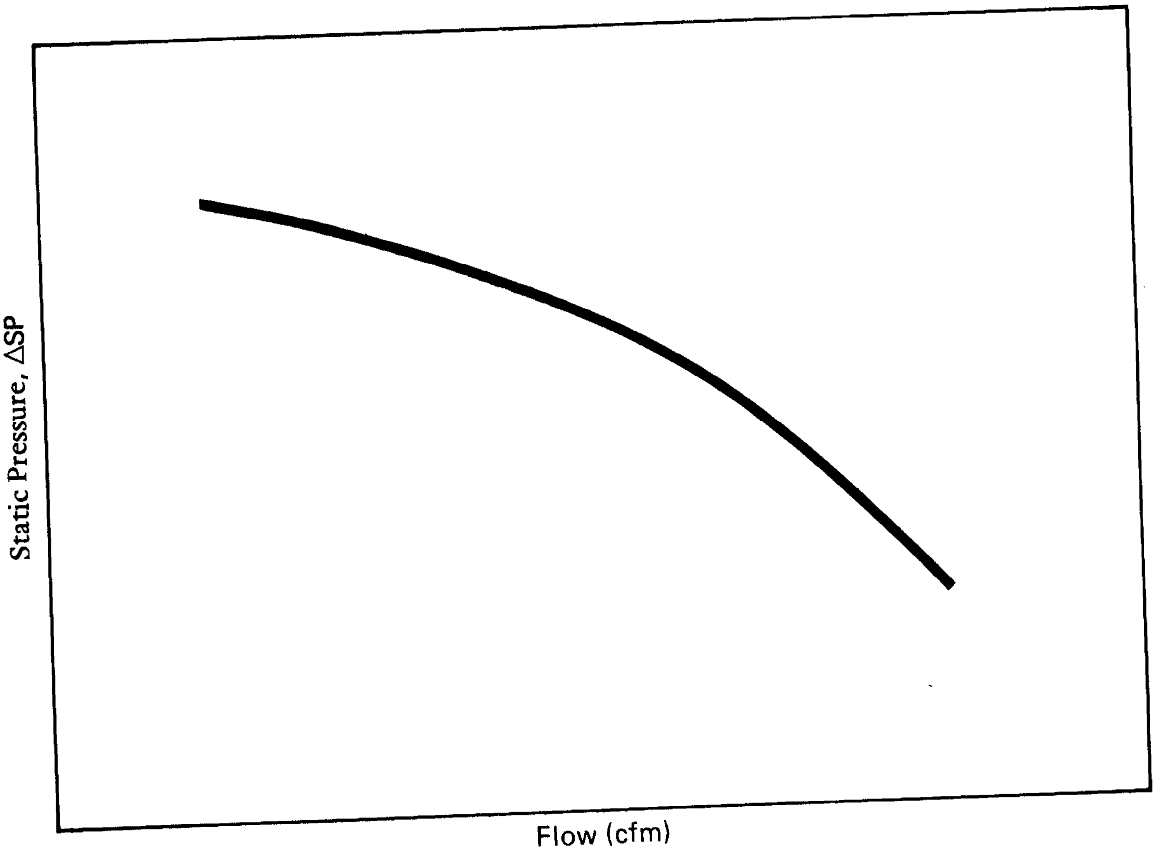


FIGURE 2-3
Sample Fan Curve

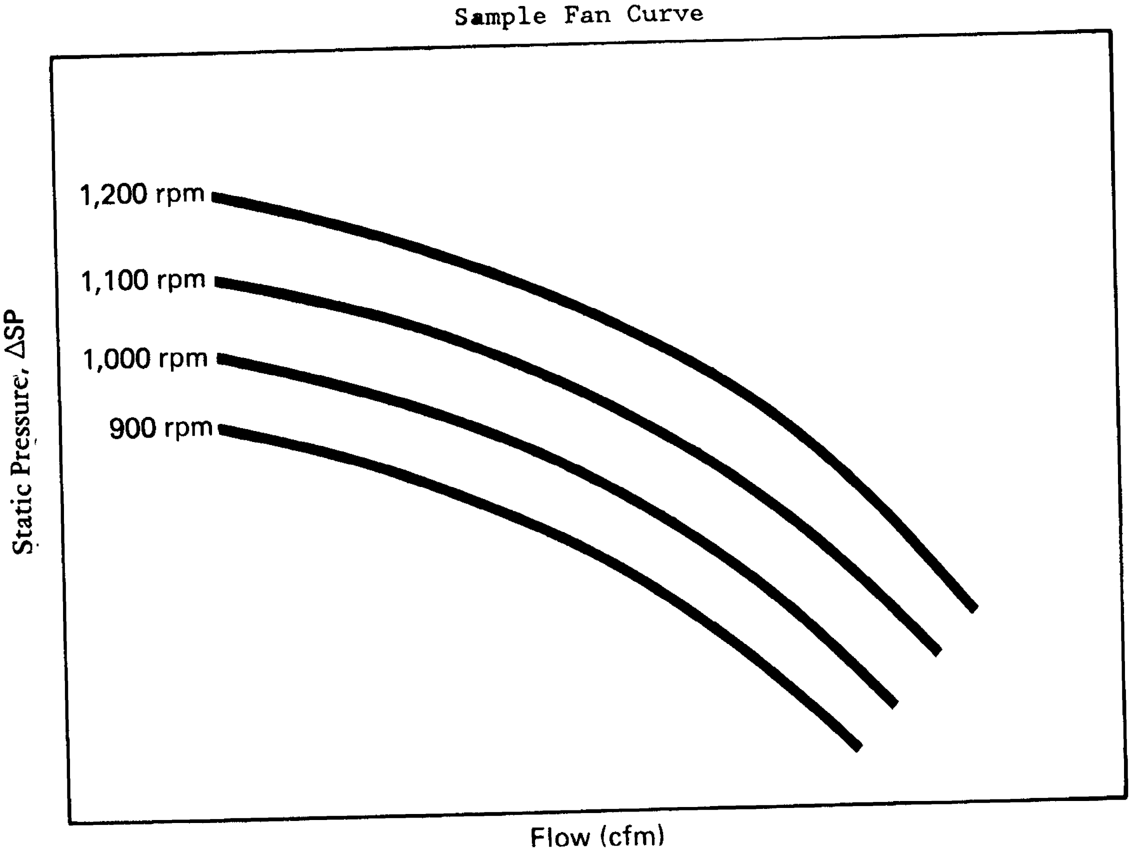


FIGURE 2-4
A Family of Fan Curves

TABLE 2-1
Portion of a Fan Multirating Table

	2" SP		2 1/2" SP		3" SP		3 1/2" SP		4" SP	
cfm	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp	rpm	bhp
13,048	583	5.16	632	6.40	675	7.71	720	9.11	765	10.6
13,980	596	5.61	643	6.89	687	8.22	728	9.65	772	11.2
14,912	611	6.08	657	7.41	700	8.78	738	10.2	779	11.8
15,844	627	6.59	671	7.97	713	9.38	750	10.9	787	12.4
16,776	643	7.12	686	8.59	726	10.0	764	11.6	800	13.1

sp -- static pressure
bhp -- brake horsepower

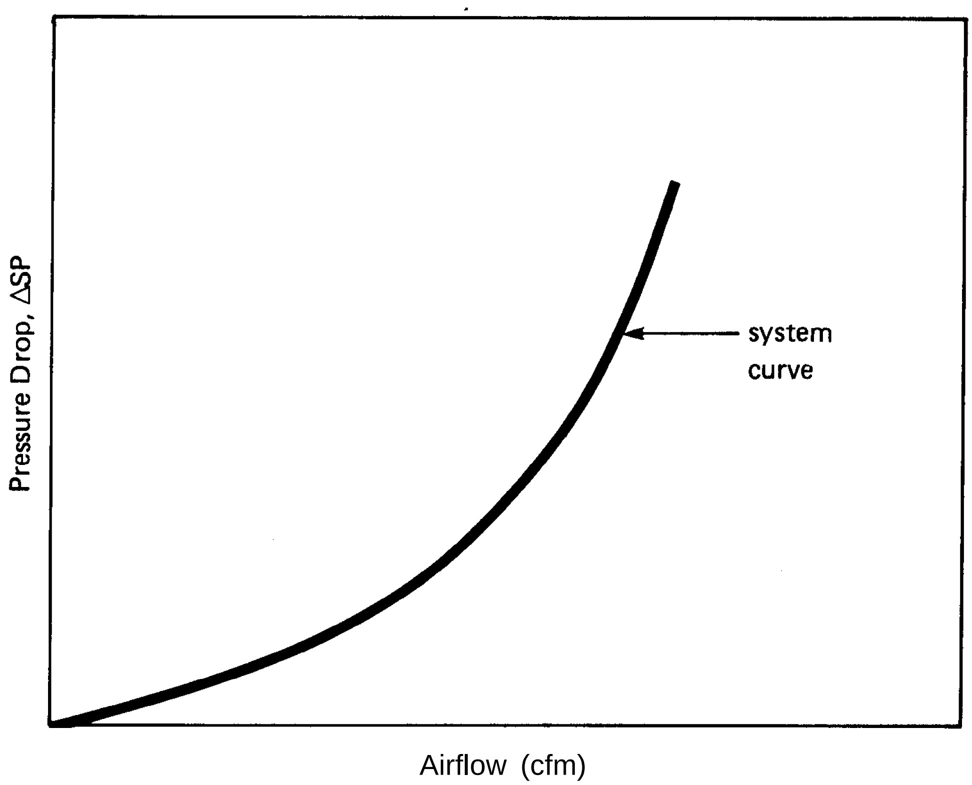


FIGURE 2-5
Sample System Curve

Table 2-2 may be used to solve the same problem. Simply divide the new desired cfm by the existing cfm to determine the ratio in the first column. Read in the second column, the appropriate factor to be multiplied by the original pressure drop.

Once the pressure drop is known for a system at any given airflow, the pressure drop for any other airflow may be calculated using the method above. When the pressure drop and airflow are plotted on a graph for a number of different airflows, we have created a system curve. The original pressure drop for a given airflow may be arrived at by measurement of the existing system. For a system yet to be installed, the designer must calculate what the friction losses will be in order to determine the total pressure loss. Calculation of these pressure losses will be explained later in this chapter.

TABLE 2-2
Static Pressure
Increase/Decrease

Ratio of (new cfm/old cfm)	New Pressure Drop = (Old Pressure Drop) x (This Factor)
0.50	0.25
0.55	0.30
0.60	0.36
0.65	0.42
0.70	0.49
0.75	0.56
0.80	0.64
0.85	0.72
0.90	0.81
0.95	0.90
1.00	1.00
1.05	1.10
1.10	1.21
1.15	1.32
1.20	1.44
1.25	1.56
1.30	1.69
1.35	1.85
1.40	1.96
1.45	2.10
1.50	2.25

2.5 FAN TYPES. The two general categories of fans are:

a. Centrifugal
b. Axial

The centrifugal fan (Figure 2-6) consists of an impeller and a housing. The overall direction of airflow is perpendicular to the direction of the fan shaft. Air is drawn into the center of the impeller and discharged through the housing. The fan may have the inlet on just one side; a double fan width may be used with inlets on both sides. These two arrangements are commonly called single width single inlet (SWSI) or double width double inlet (DWDI).

The impeller is driven through a direct drive or belt drive arrangement. Direct drive fans, with the motor housing inside the impeller (Figure 2-7), are sometimes referred to as squirrel cage fans.

There are several different designs in use for the centrifugal fan impeller. They are:

1. Forward curved (FC)
2. Backward inclined (BI)
3. Airfoil (AF)
4. Radial

The blade configuration for each of the impeller types is shown in Figure 2-8. Of all the different designs, the FC fan is by far the most commonly used in residential and commercial applications. For a given cfm requirement, the FC fan will be the smallest wheel, and will operate at the lowest rpm, The FC wheel is used where the static pressure requirements are moderately low (up to 2 to 3 inches water column). The forward looking cup of the FC wheel can become filled with dirt, and reduce the fan's effectiveness. This is not a problem in clean applications.

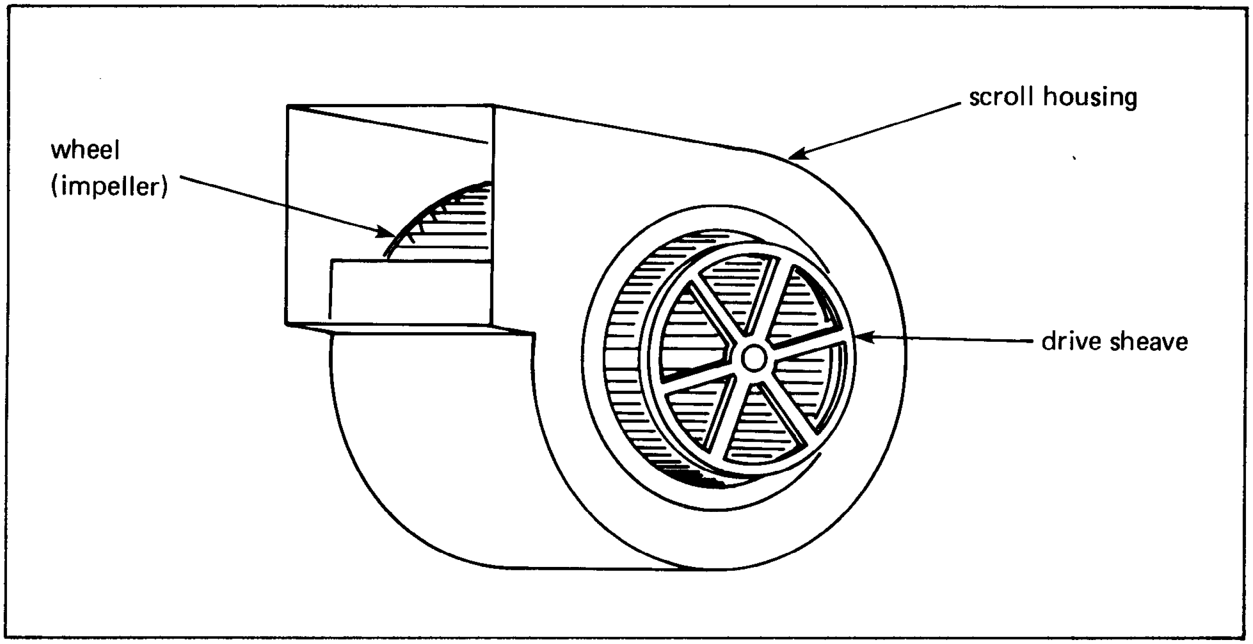


FIGURE 2-6
Centrifugal Fan

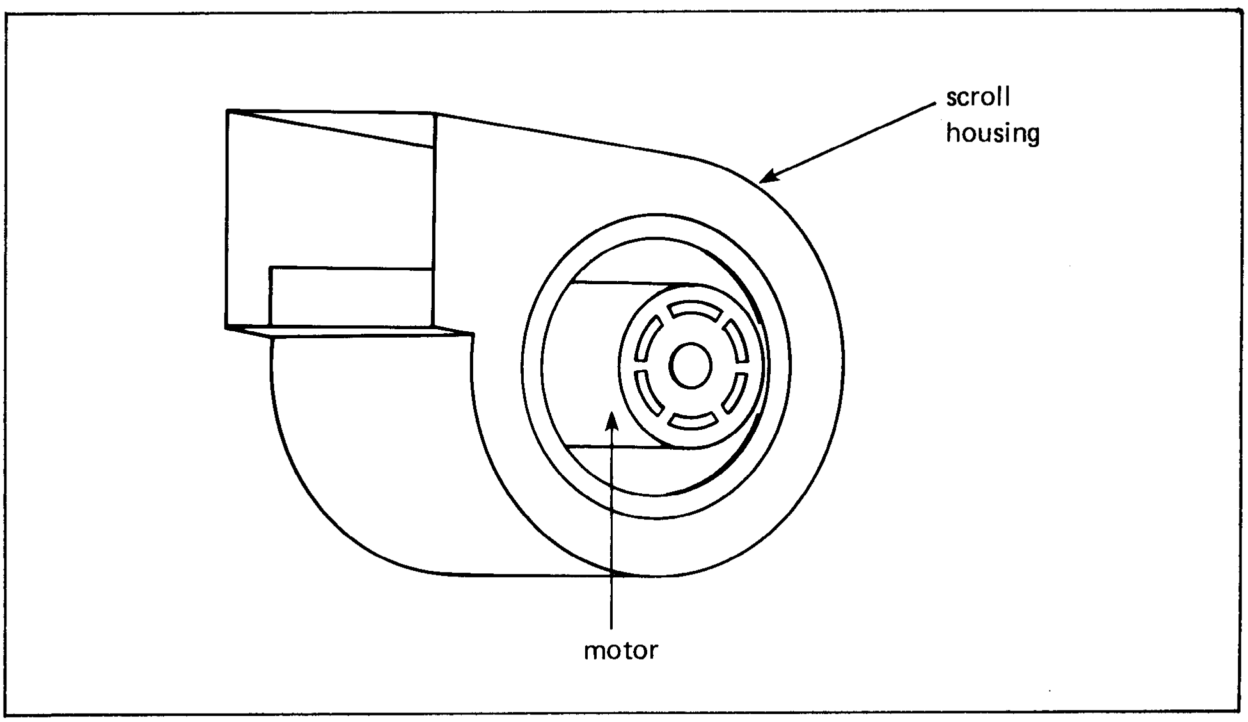


FIGURE 2-7
Direct Drive "Squirrel Cage" Fan

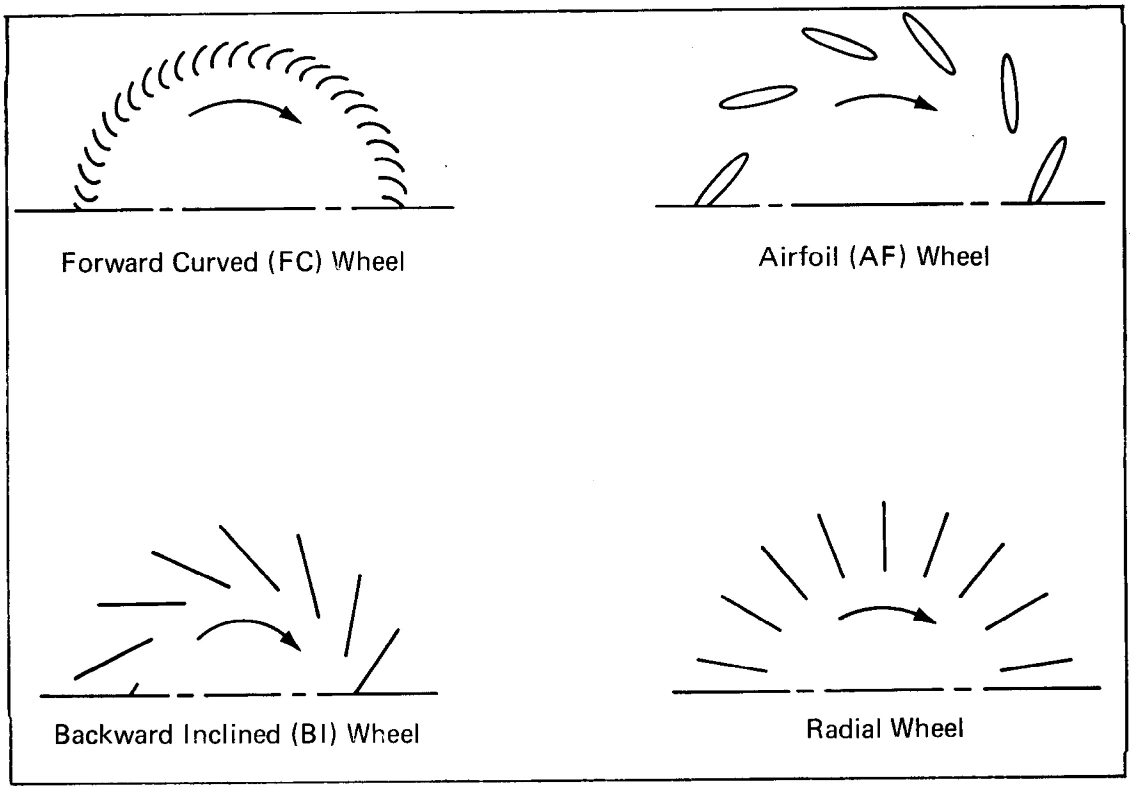


FIGURE 2-8
Impeller Types Used in Centrifugal Fans

The BI and the AF fans are more efficient, and more expensive than the FC wheel. The rpm for these fans is much higher than for the FC fans. Note the direction of rotation indicated. It is opposite from the usual direction. You must be able to recognize the correct rotation direction to be backwards when troubleshooting a new installation. A fan that is rotating backwards will not move air in a backwards direction. It will move air in the proper direction through the ductwork, but at a drastically reduced flow rate.

The radial design fan blade is not generally used in normal air conditioning applications. It is designed for high-strength, rugged construction. It is more commonly used in material handling applications.

Axial fan airflow is in the same direction as the fan shaft. Common types of axial flow fans are:

1. Propeller
2. Tubeaxial
3. Vaneaxial

The propeller fan has relatively low efficiency, inexpensive construction, and is not capable of producing static pressures in excess of a few tenths of an inch. Propeller fans can move lots of air with a free discharge but they have a very flat fan curve. As the static pressure requirement increases, the cfm decreases very dramatically (Figure 2-9).

The tubeaxial fan (Figure 2-10) consists of a propeller-type fan blade inside a cylindrical tube. The clearance between the blades and the tube is close, increasing the efficiency. The tubeaxial is capable of developing enough static pressure to move air through low resistance duct systems.

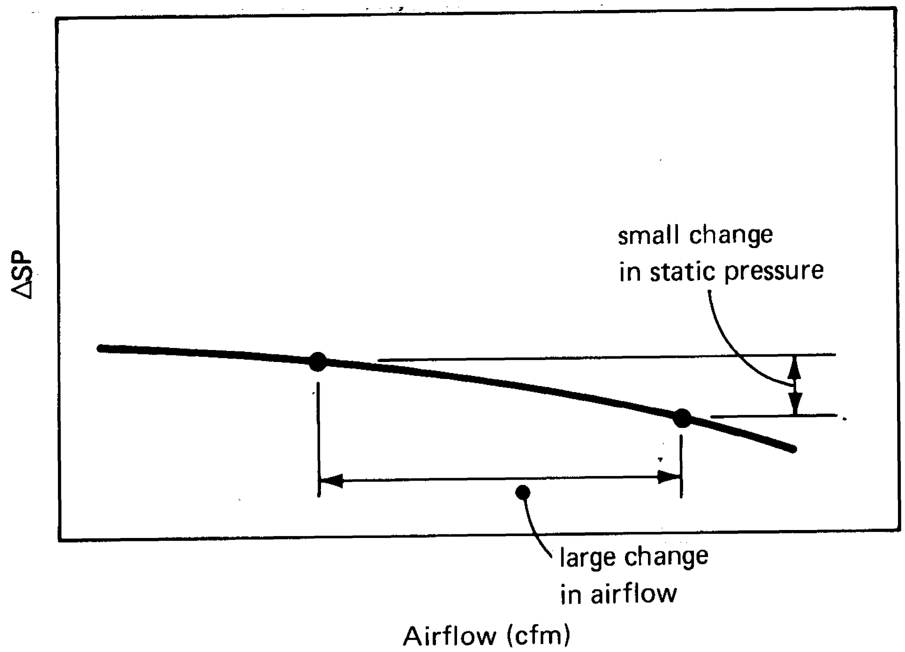


FIGURE 2-9
Flat Fan Curve for a Propeller Fan

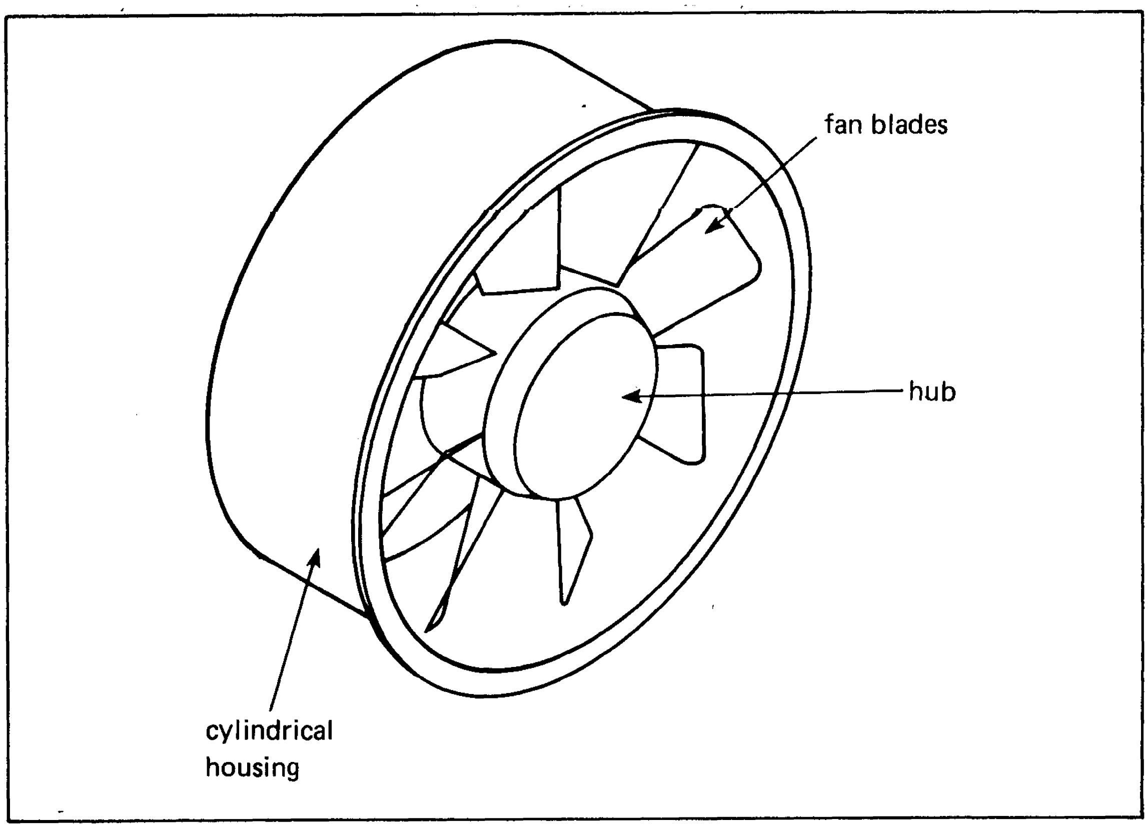


FIGURE 2-10
Tube - Axial Fan

The vaneaxial fan has a set of stationary vanes placed ahead of or after the propeller. These vanes help straighten the air and convert the rotation energy to static pressure energy. The fan blades may be fixed or adjustable to meet the specific application. Vaneaxial fans can develop static pressures comparable to that produced by the centrifugal fans. They have the advantage of a straight-through design, and are well suited to installation in a straight run of ductwork. Some vaneaxial fans have higher noise characteristics than comparable capacity centrifugal fans.

2.6 FAN-TO-SHAFT CONNECTION. The fan is mechanically connected to its shaft by one of three methods:

a. Set screw
b. Set screw and key
C. Expanding hub

The same methods are used for fastening pulleys (sheaves) to shafts. Figure 2-11 shows the simplest type of fan-to-shaft attachment. The set screw threads through the fan hub, and bottoms out on the fan shaft. When installing a fan with a set screw, there will be one flat side on the shaft. It is imperative that the set screw be lined up with this flat spot. If it is not, the fan may be secure to the shaft, but it will soon work its way loose.

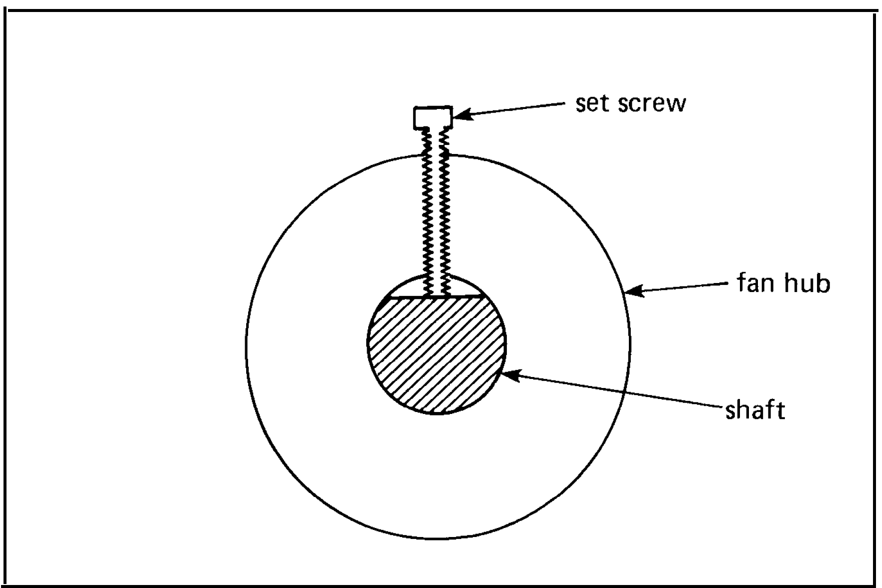


FIGURE 2-11
Fan Hub Fastened to Shaft With a Set Screw

Figure 2-12 shows a key fastening arrangement. This is much stronger than the simple set screw method. It is used for more severe fan duty. When removing a fan that is fastened in this manner, take care not to lose the key.

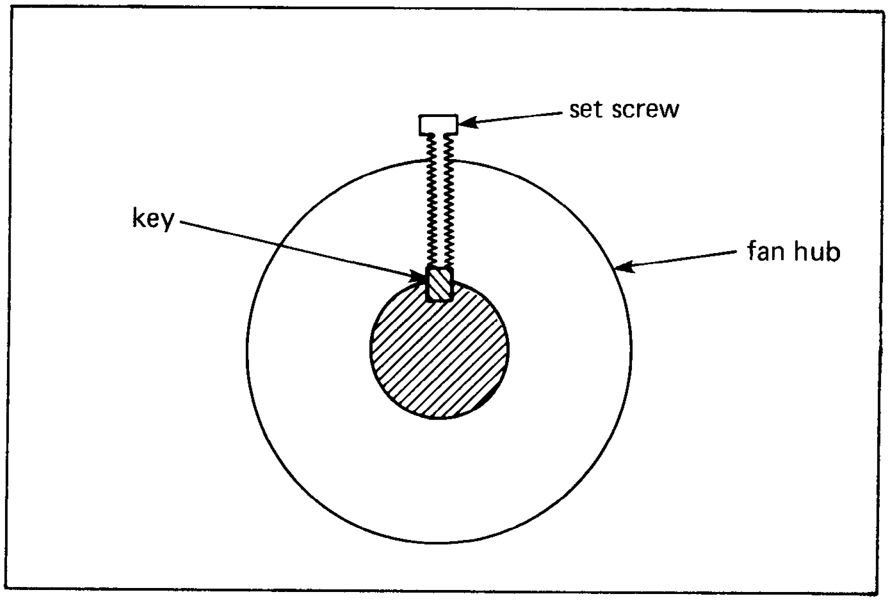


FIGURE 2-12
Fan Keyed to a Fan Shaft

Figure 2-13 shows a fan with a separate tapered fan hub. The hub is tightened against the side of the fan, forcing the tapered section in between the fan hub and the shaft.

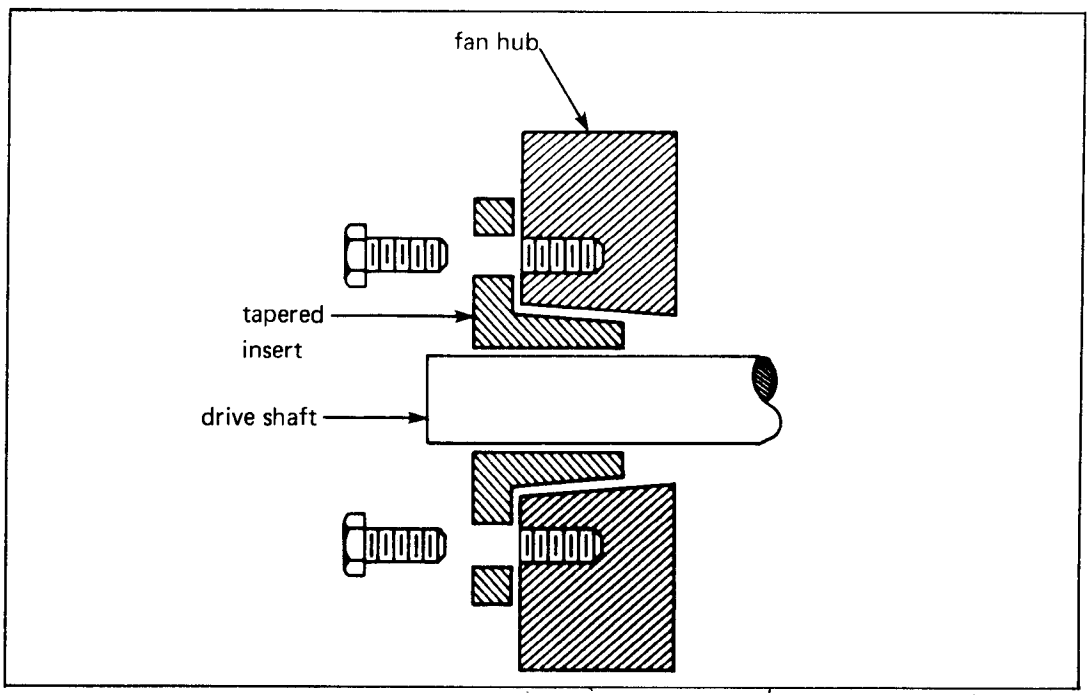


FIGURE 2-13
Tapered Fan Hub

2.7 BELT-DRIVE FANS. The general arrangement for a belt-drive fan is shown in Figure 2-14. Generally, the motor speed is 1,750 rpm and the fan speed is somewhat lower. The relationship between the fan rpm and the motor is:

$$\text{Fan rpm} = (\text{PD motor}/\text{PD fan}) \times \text{motor rpm}$$

where: PD motor = the pitch diameter of the pulley on the motor

PD fan = the pitch diameter of the pulley on the fan

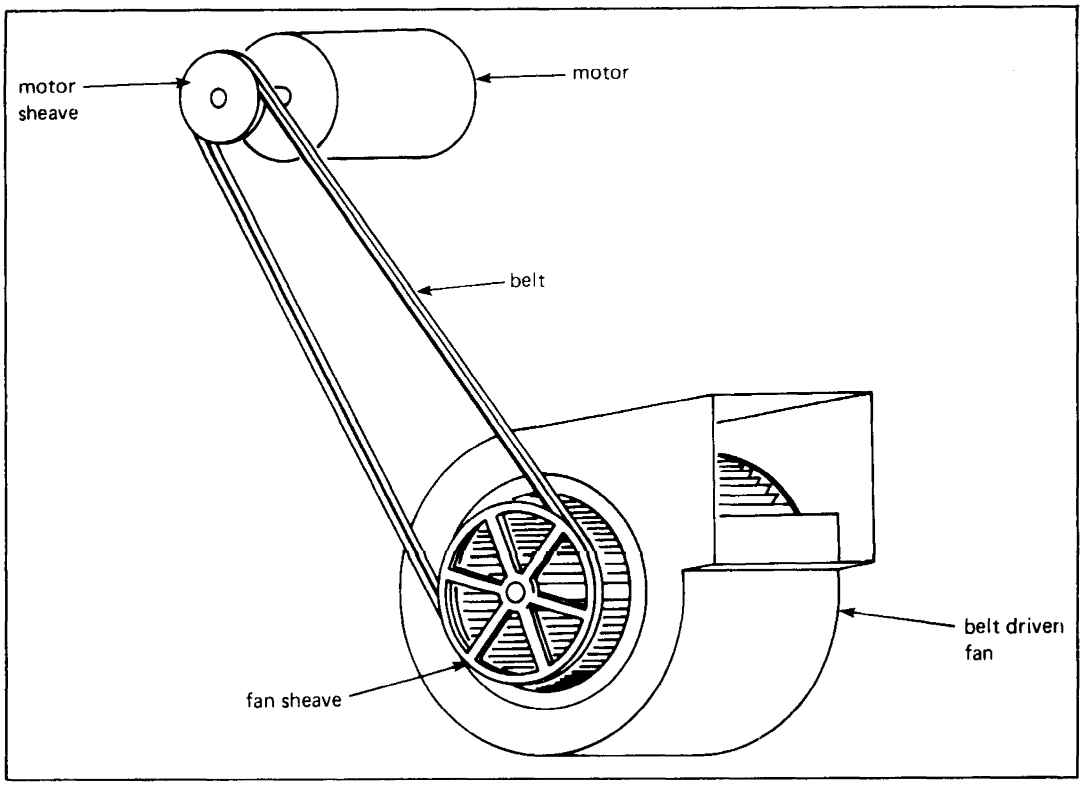


FIGURE 2-14
Typical Belt Drive Fan

Pitch diameter is a term used to describe the diameter dimension in between the outside diameter of the belt and the inside diameter of the belt (Figure 2-15).

Example: A 1,750-rpm motor with a 3-inch pitch diameter pulley needs to drive a fan at 950 rpm. What size fan pulley should be used?

Solution:

$$950 \text{ rpm} = (3/\text{PD fan}) \times 1{,}750$$

$$\text{PD fan} = (3) \times (1{,}750/950)$$

$$= 5.53 \text{ inches}$$

When a fan speed needs to be changed in order to change the airflow, the motor pulley size is increased to increase the fan flow, and decreased to decrease the fan flow. Because the motor pulley is usually smaller than the fan pulley, it is less expensive to change the motor pulley than the fan pulley. For a change in cfm, the new pulley setting ($P\ d_2$) is calculated as follows:

$$P\ d_2 = P\ d_1 \times (cfm_2/cfm_1)$$

where:
- $P\ d_1$ = original pitch diameter
- $P\ d_2$ = new pitch diameter
- cfm_1 = air quantity at the fan, original
- cfm_2 = air quantity at the fan, new

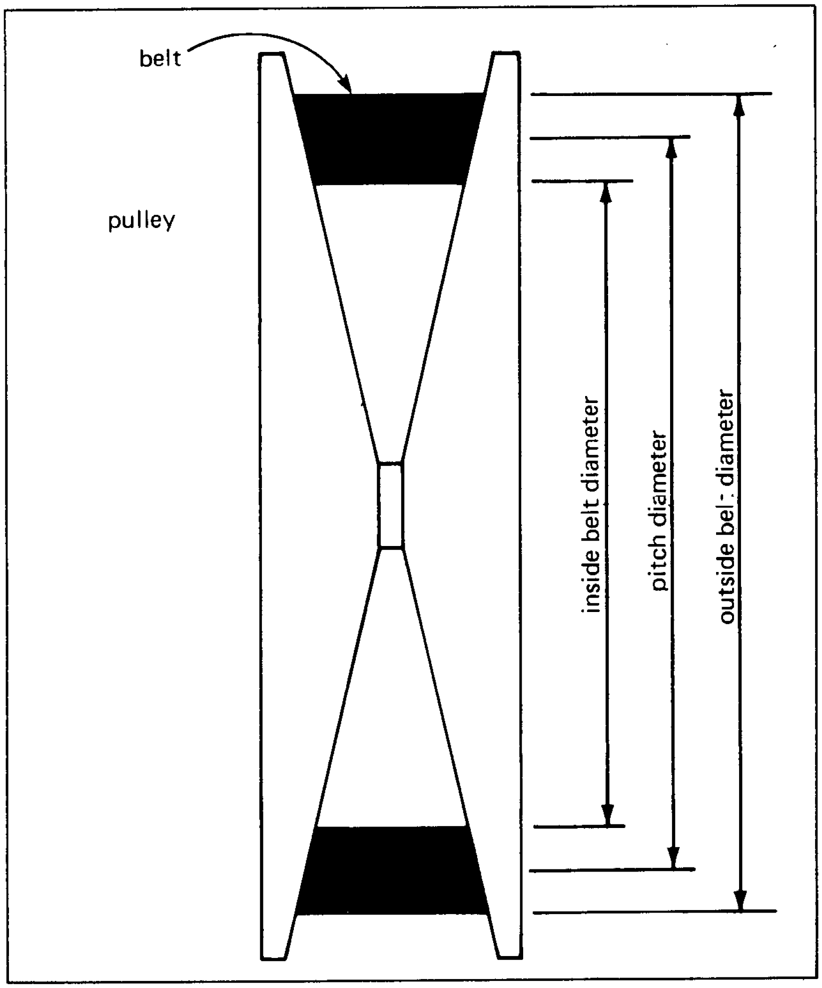


FIGURE 2-15
V-Belt Pully Standard Groove Dimensions For Sheaves

Figure 2-16 shows a variable pitch sheave. The two halves of the sheave are two different pieces, with one threaded onto the shaft of the other. Depending how closely the halves are positioned to each other determines the actual pitch diameter of the sheave. Variations of ±30 percent from the nominal pitch diameter are commonly available with this type of sheave. The variable pitch sheave is only slightly more expensive than the fixed pitch sheave in the smaller sizes. However, in the larger sizes, or for multiple groove sheaves, the variable pitch sheave can be two or three times the price of a fixed sheave.

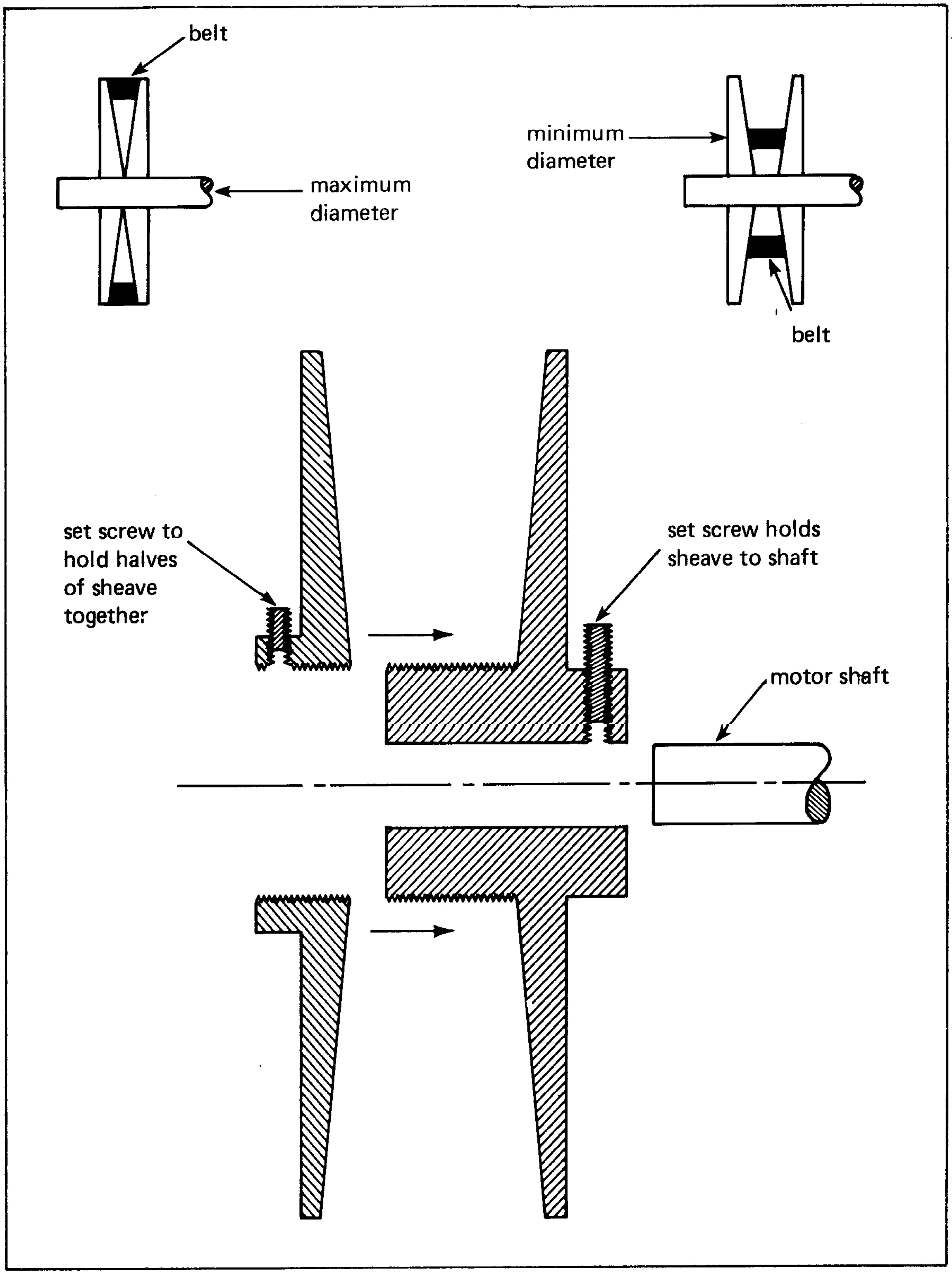


FIGURE 2-16
Variable Pitch Sheave

2.8 DRIVE BELTS. Drive belts, or V-belts, are designated by their cross-sectional dimension and their length, in inches. The standard cross-section dimensions of a belt width and depth are given in Table 2-3. A model for a belt might be, for example, B48. This would mean that it has a cross section of 0.66 inches width, 0.41 inches depth, and 48 inches length. The length is at its pitch diameter. The actual outside dimension of the belt will be an inch or two longer. When replacing a fan belt, the new belt must be of the same cross section as the original. If it is different, it will ride higher or lower in the groove of the sheave, and change the effective pitch diameter of the sheave. New belts will normally ride much higher in the sheave groove than an old belt. When the new belt wears, its "w" dimension can become smaller, causing the belt to sink deeper into the groove of the sheave.

TABLE 2-3
Nominal V-Belt Cross Sections

Belt Designation	W (in.)	d (in.)
A	0.50	0.31
B	0.66	0.41
C	0.88	0.53
D	1.25	0.91
E	1.50	0.91

When replacing a drive belt, it is necessary to move the motor to relieve the belt tension. After the belt is slipped over the sheaves, the belts are retensioned. One guideline is to make it tight enough so that the center of the span will deflect 1 inch when moderate finger pressure is applied (Figure 2-17). This is not very exact. There are tools available for measuring belt tension, but the required belt tension is not always known. A simple guideline is to make the belt as loose as possible, without it slipping on the sheave or flopping around when it is operating. A belt that is tightened too much puts extra force on the fan and motor bearings, and can cause bearing failure.

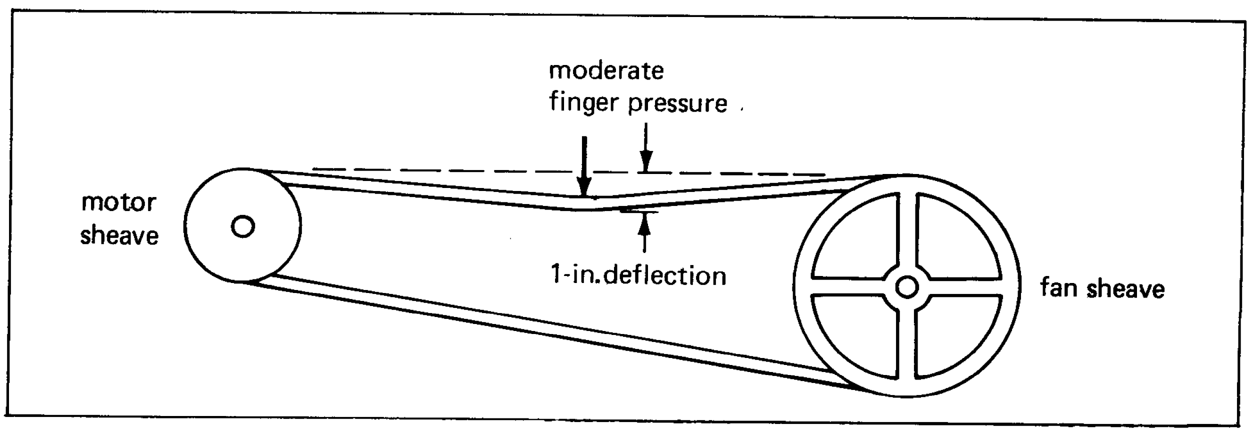


FIGURE 2-17
Setting Belt Tension

2.9 FAN LAWS. Fan laws are a set of relationships that predict the changes in a fan's operating characteristics when its speed is changed. When the speed of a fan is increased, the flow (cfm) increases, the static pressure (sp) produced by the fan (and used up in the system ductwork friction) increases, and the horsepower (hp) required to drive the fan increases. The following formulas predict how much each of these variables will increase:

$$cfm_2 = cfm_1 \times (rpm_2/rpm_1)$$

$$sp_2 = sp_2 \times (rpm_2/rpm_1)^2$$

$$hp_2 = hp_1 \times (rpm_2/rpm_1)^3$$

In other words, the flow (cfm) increases in direct proportion with the rpm, the static pressure increases with the square of the wpm, and the horsepower consumed by the fan varies with the cube of the rpm.

Example: A fan is delivering 4,000 cfm against a static pressure of 1.6 inches water column. It is driven by a motor that has a 3.5-inch sheave. The fan horsepower is 1.4 hp. If the motor sheave were changed to 4.5 inches, what would be the new cfm, static pressure, and horsepower?

Solution: The diameter of the motor sheave has been increased by a factor of (4.5/3.5) = 1.29. Remember when you increase the motor pulley you increase the flow, static pressure and horsepower requirement.

Therefore, the ratio of the new fan rpm to the old fan rpm is:

$$(rpm_2/rpm_1) = 1.29$$

The new cfm being delivered will be:

$$cfm_2 = (4{,}000) \times (1.29) = 5{,}160 \text{ cfm}$$

The new static pressure will be:

$$sp_2 = (1.6) \times (1.29)^2 = 2.66 \text{ inches water column}$$

and the new horsepower will be:

$$hp_2 = (1.4) \times (1.29)^3 = 3.0 \text{ hp}$$

The most important facet of the fan laws to the technician is to understand how fast the horsepower draw increases with a relatively small increase in fan speed. In the example above, the airflow and fan rpm were increased by 29 percent, the static pressure was increased by 166 percent and the horsepower requirement more than doubled to a 214 percent increase.

When changing a sheave size, or increasing the setting on a variable pitch sheave, be careful not to overload the motor. With reasonable accuracy, you can substitute "motor amps" in the above equation to replace horsepower.

Example: A motor with a 9.0-amp nameplate rating is using a 5-inch sheave to drive a fan. The motor is slightly oversized, and the actual amperage draw is 6.2 amps. If the motor sheave size is increased to 6 inches, will the motor be able to handle the increase.

Solution: The increase in fan rpm is the same proportion as the increase in the motor sheave diameter, so

$$(rpm_2/rpm_1) = 6.0/5.0 = 1.2$$

The new amperage draw on the motor will be:

$$\begin{aligned} amps_2 &= 6.2 \times (1.2)^3 \\ &= 10.7 \text{ amps} \end{aligned}$$

The increase in motor sheave diameter will require that the motor be changed to a larger horsepower size, which provides at least 10.7 amps.

Table 2-4 is provided to simplify the process of figuring the rpm ratio squared and cubed.

TABLE 2-4
Tabulated Data Representing the Result of Application of the Fan Laws

If the new fan rpm divided by the original fan rpm is:	multiply the original fan cfm by:	multiply the original fan sp by:	multiply the original hp and amps by:
1.05	1.05	1.10	1.16
1.10	1.10	1.21	1.33
1.15	1.15	1.32	1.52
1.20	1.20	1.44	1.73
1.25	1.25	1.57	1.95
1.30	1.30	1.69	2.20
1.35	1.35	1.82	2.46
1.40	1.40	1.96	2.74
1.45	1.45	2.10	3.05
1.50	1.50	2.25	3.37

2.10 VELOCITY AND CFM. When we talk about airflow quantities, we are talking about cubic feet per minute (cfm). When we talk about velocity, we are describing how fast a quantity of air is moving, in feet per minute (fpm). The general relationship between velocity and air quantity is:

$$Q = A \times V$$

where: Q = flow in cfm

V = velocity in fpm

A = cross-section area, in ft^2

In almost all cases, airflow is not measured directly with a meter that reads out in cfm. Rather, we usually use a meter that measures velocity, and we use the above formula to calculate the actual air quantity.

2.11 THE VELOCITY TRAVERSE. The velocity of air flowing inside a duct is rarely uniform across the entire cross section. In order to arrive at an accurate average velocity through the duct, it is necessary to measure the velocity at a number of different points in the duct. This is called taking a velocity traverse.

For air flowing through a duct, the velocity distribution might look something like Figure 2-18. If the length of the arrows represent the actual air velocity, you can see that the air velocity is highest in the center of the duct, and drops off very quickly near the sides of the duct. In order to arrive at an average, the technician mentally divides the duct cross section into equal areas, and measures the velocity at the center of each of the equal areas (see Figure 2-19). In a straight duct, some service technicians get rough estimates of the average velocity by measuring just the maximum velocity in the center of the duct, and multiplying that value by 0.85.

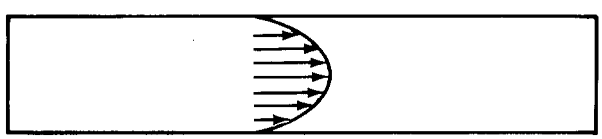


FIGURE 2-18
Velocity Distribution of Air
Flowing Inside a Duct

When selecting a location to take a duct traverse, a location preceded by a long straight duct is ideal. If measurements are made close to a fan or an elbow, a very odd and nonuniform set of readings will result (see Figure 2-20). When measuring velocities from a supply air or return air grille, the face area is mentally divided and individual velocity readings are taken in the center of each area and averaged.

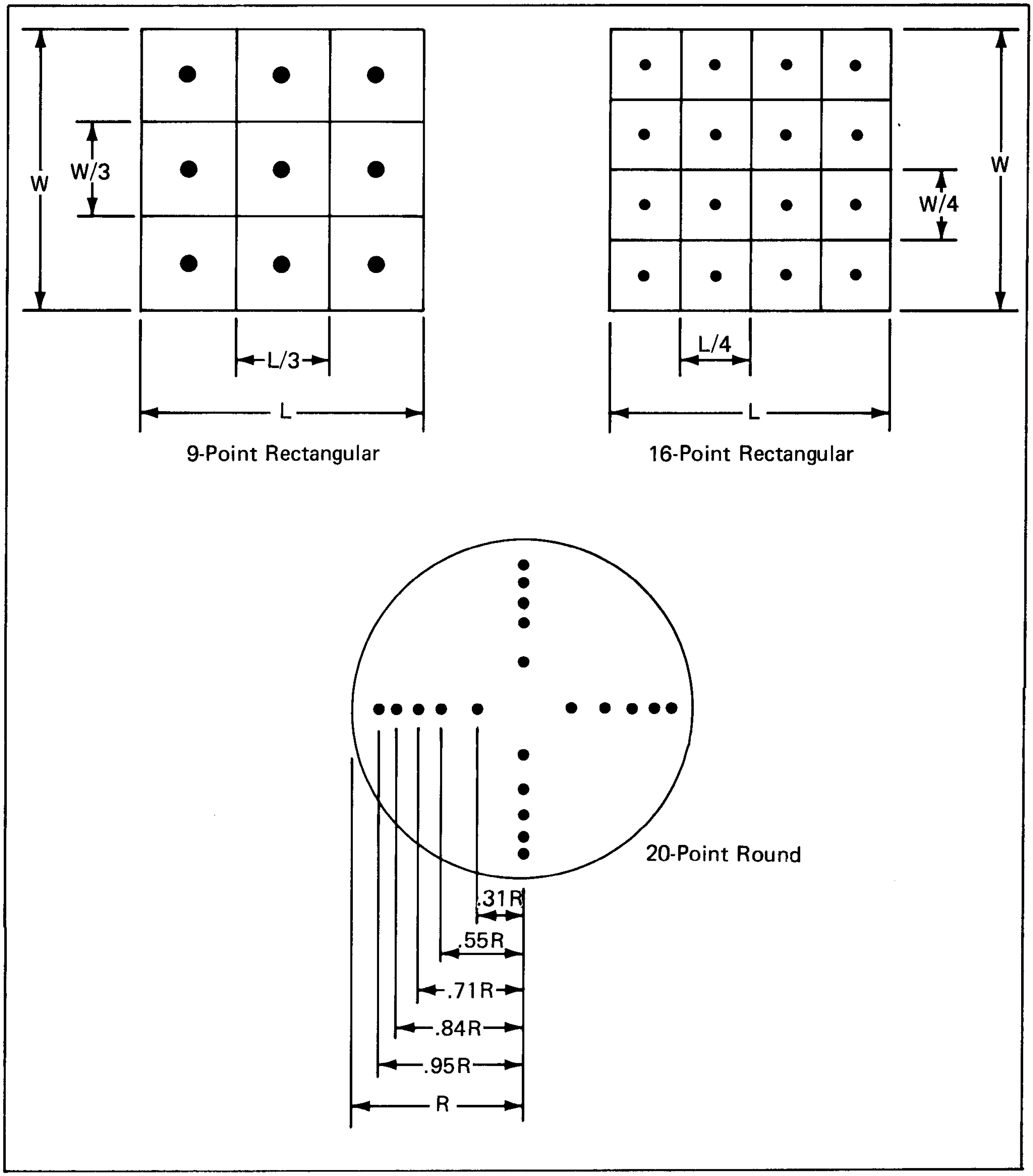


FIGURE 2-19
Measurement Locations for a Velocity Traverse

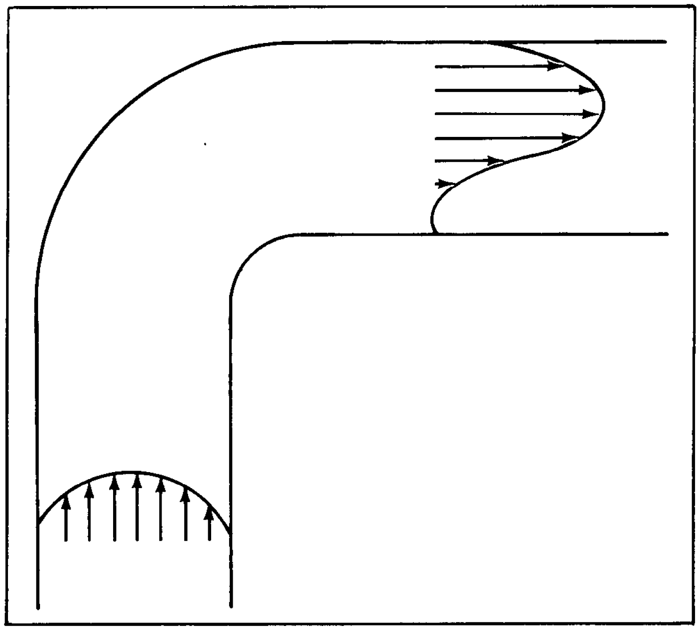


FIGURE 2-20
Uneven Velocity Distribution Downstream
From a Turn in a Duct

Example: A velocity traverse is taken across the face of a 16 by 32-inch return air register as shown in Figure 2-21. How much air is passing through this return register?

1,260	1,270	1,250
1,250	1,260	1,240
1,210	1,230	1,190

FIGURE 2-21
Sample Velocity Traverse
(fpm)

Solution: $Q = AxV$

The average velocity is calculated by averaging all the individual readings.

Average velocity = (11,160/9) = 1,240 fpm

and then:

$Q = (16 \text{ in} \times 32 \text{ in})/144 \text{ in}^2/\text{ft}^2 \times 1,240 \text{ fpm}$

$= 4,409 \text{ cfm}$

2.12 ROTATING VANE ANEMOMETER. An anemometer is any device used to measure the velocity of air. A rotating vane anemometer is shown in Figure 2-22. When placed in a moving airstream, the air causes the propeller to rotate, just as a child's pinwheel spins in the wind. The rotating vane is geared to the dial which indicates how many feet of air have passed through the vane. If the rotating vane is placed in an airstream for 1 minute, then the "feet" reading will actually be feet per minute. Or, a reading may be taken for one half of a minute and then doubled to arrive at velocity in feet per minute.

The rotating vane anemometer cannot be used to measure velocity inside a duct. It is used to measure velocities in exposed areas such as grilles and registers, openings such as doorways, or across coils where there is physical access. The range of accurate measurement is in the range of 100 to 5,000 fpm. Below 100 fpm, the friction in the bearings and gears introduce inaccuracies.

In order to use the rotating vane anemometer, it is placed in a position perpendicular to the airflow. When the rotating vane gets up to a stable speed, the start lever is turned, engaging the vane to the counting mechanism. After 30 seconds or 1 minute, the counting mechanism is disengaged, and the velocity is calculated.

With this instrument, as with all air measuring devices, make sure that you are not blocking the airflow from the instrument with your hand or your body. An interesting feature of a continuously measuring and recording instrument such as this is that the need to take separate readings across the face of a grille can be eliminated. The anemometer may be continuously moved across the face of the grille while the velocity is being measured. The overall reading will then be an actual average, as recorded.

The angular settings of the blades of the rotating vane are crucial to the accuracy of the instrument. This is not a device to be subjected to rough treatment if accuracy is to be maintained. Periodically, this instrument needs to be recalibrated.

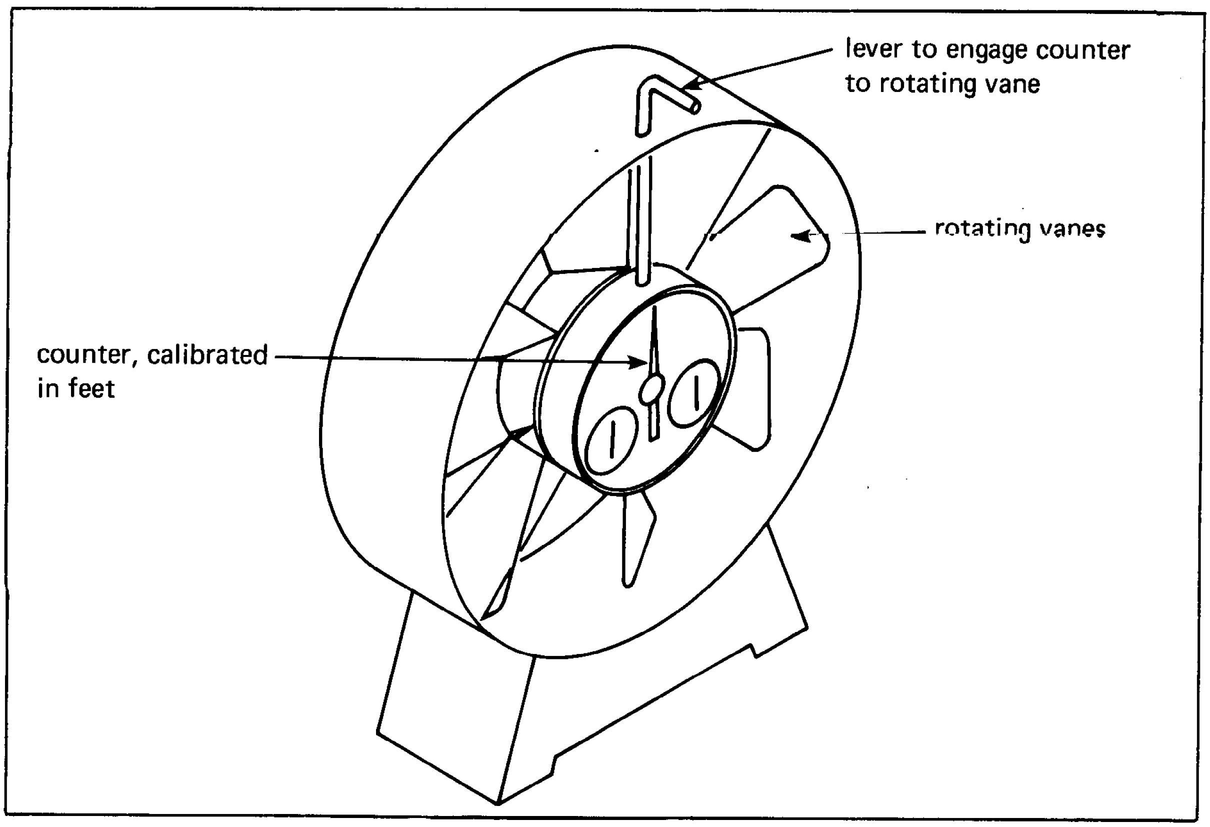


FIGURE 2-22
Rotating Vane Anemometer

2.13 DEFLECTING VANE VELOMETER. Figure 2-23 shows another velocity measuring instrument that can be similar in appearance to the rotating vane anemometer. The major difference is that the vane is connected to a spring that only allows the vane to deflect rather than rotate. The deflection of the vane is geared to a dial that reads out in fpm. The applications for this instrument are the same as for the rotating vane.

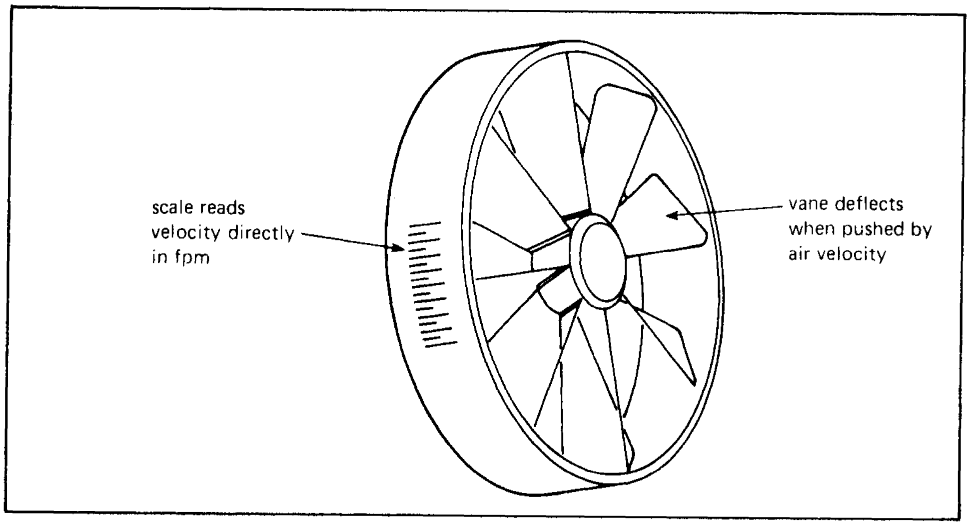


FIGURE 2-23
Deflecting Vane Anemometer

2.14 HOT WIRE ANEMOMETER. Figure 2-24 shows an instrument with a readout box and a remote probe. The probe has a thin wire that is exposed to the airflow. A current is passed through the wire causing it to heat. The wire is cooled by the velocity of the air. Electronic circuitry in the box translates the velocity into a dial readout.

The hot wire anemometer is quite accurate, even at velocities down to 30 or 40 fpm. Also, it can be used to measure velocity inside small ducts, as only the end of the probe needs to be inserted into a small hole in the side of the duct. The current for the wire can be provided either by batteries in the readout box, or from a 120-volt power source.

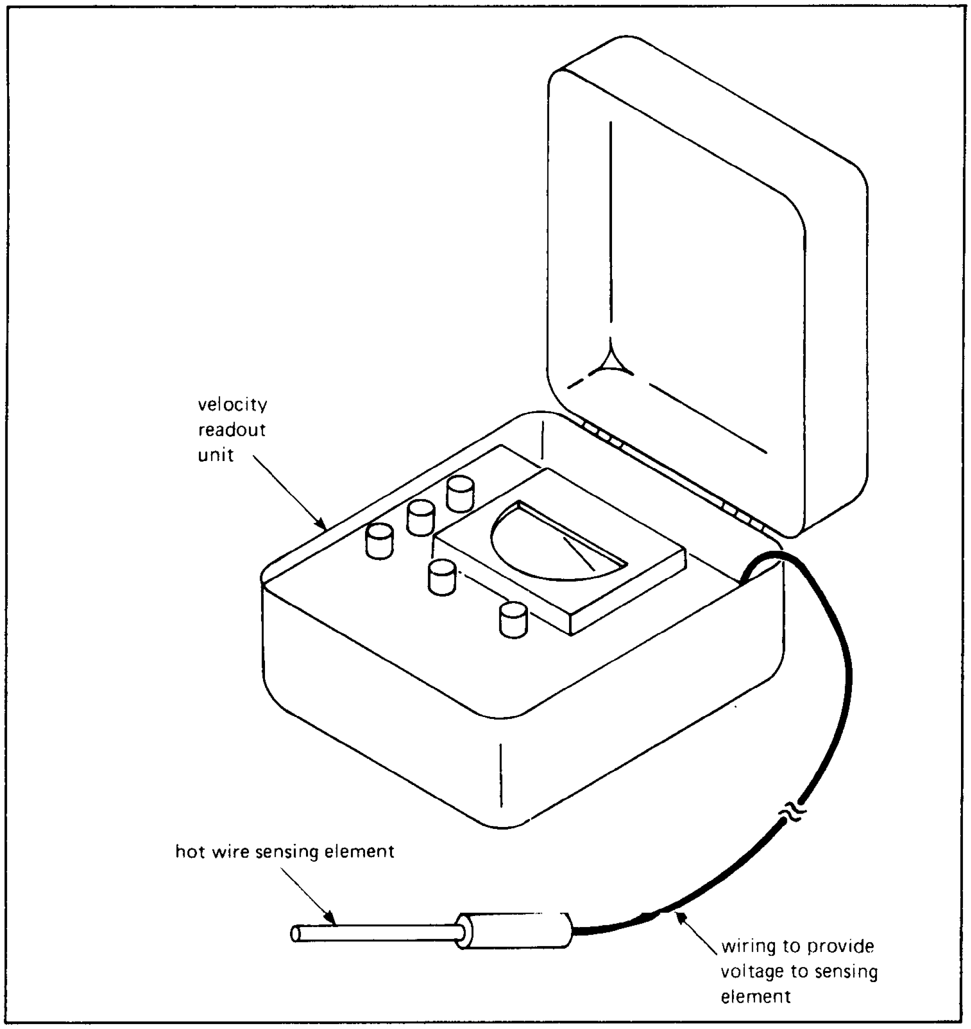


FIGURE 2-24
Hot Wire Anemometer

2.15 VELOCITY PRESSURE AND STATIC PRESSURE. There are additional instruments to be discussed; however, we cannot proceed further without a discussion of velocity pressure and static pressure.

When you hold your hand out the window of an automobile moving at 55 mph, the force on your hand is due to the pressure of the moving air. This is called velocity pressure. The instruments described up to this point are responsive to only velocity pressure. Velocity pressure can be measured with a device called a manometer as shown in Figure 2-25.

When moving air impacts the end of the sensing tube, it creates a velocity pressure. This pressure, in turn, causes the water level in the left leg of the manometer to be lower than the level in the right leg. The greater the velocity, the greater the difference in liquid levels. Velocity pressure is measured in inches of water column (or w.c.). A velocity of just over 4,000 fpm will cause the difference in levels to be exactly 1.0 inch w.c.

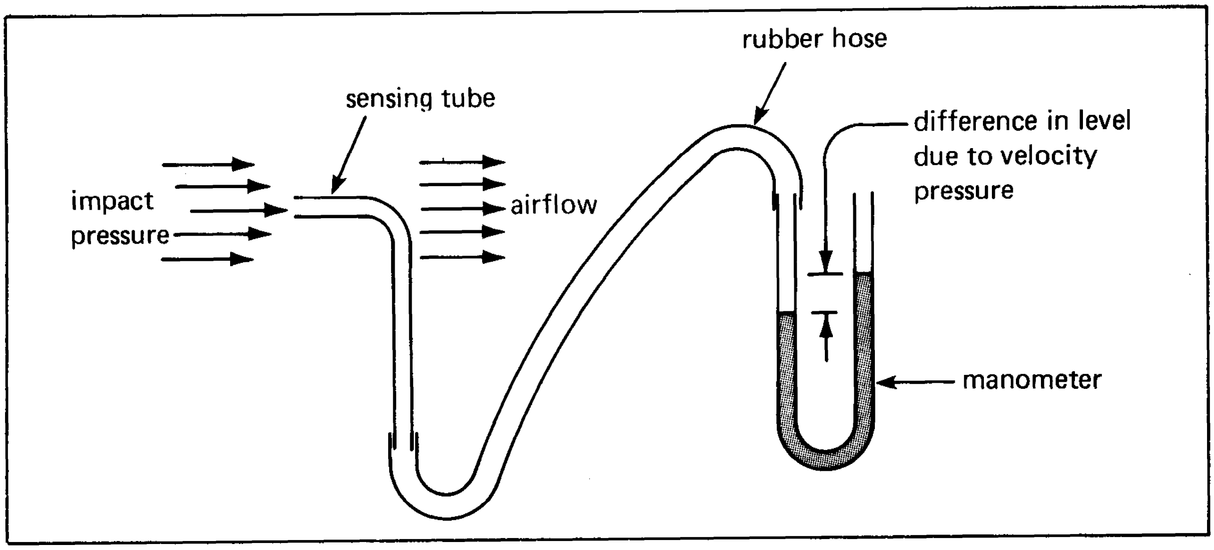


FIGURE 2-25
Using a Manometer to Measure Velocity Pressure

The relationship between velocity and the velocity pressure is:

$$V = 4{,}005\sqrt{VP}$$

where: V = velocity, fpm
VP = velocity pressure, w.c. in inches

<u>Example</u>: Using a water filled manometer, you measured a velocity pressure of 4.0 inches of w.c. What is the air velocity?

<u>Solution</u>: $V = 4{,}005\sqrt{VP}$

$V = 4{,}005\sqrt{4.0}$

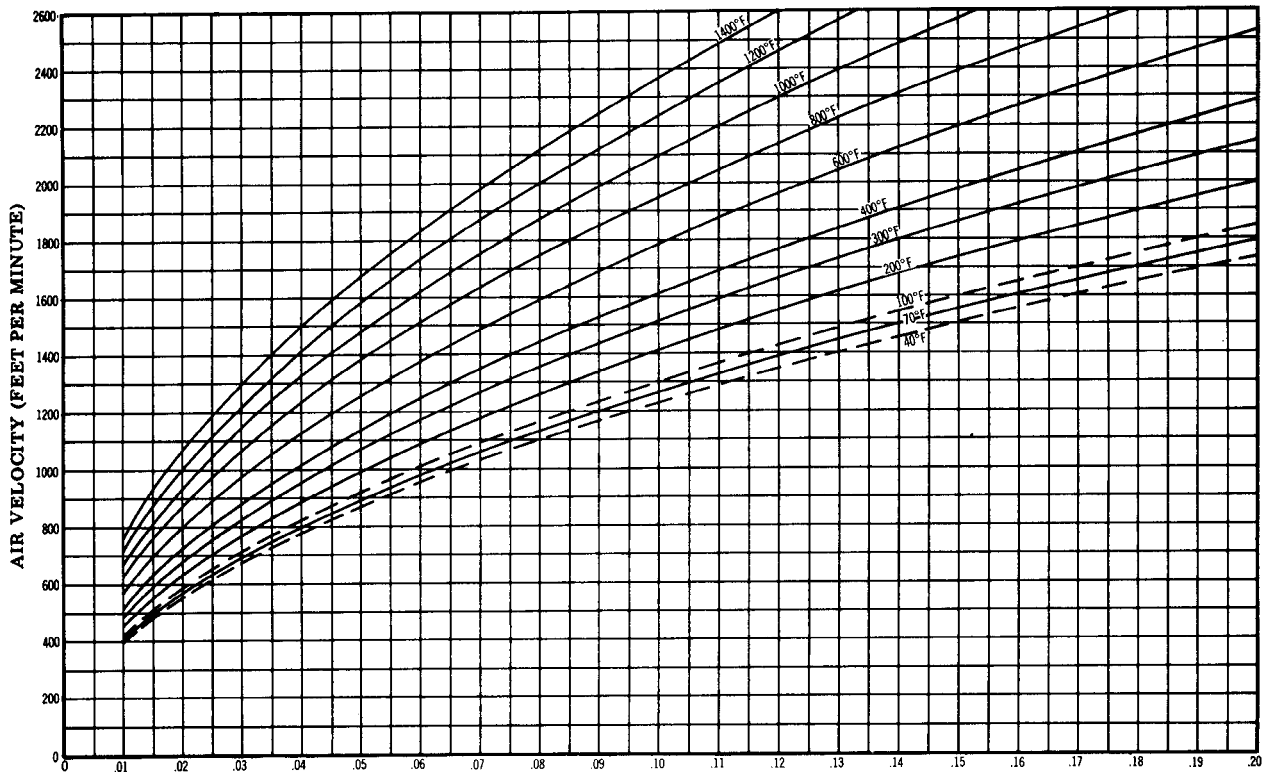
$V = 4{,}005 \times 2.0$
$V = 8{,}010$ fpm

Figure 2-26, air velocity flow charts, tabulates the velocity pressure for velocities that may be encountered in normal applications.

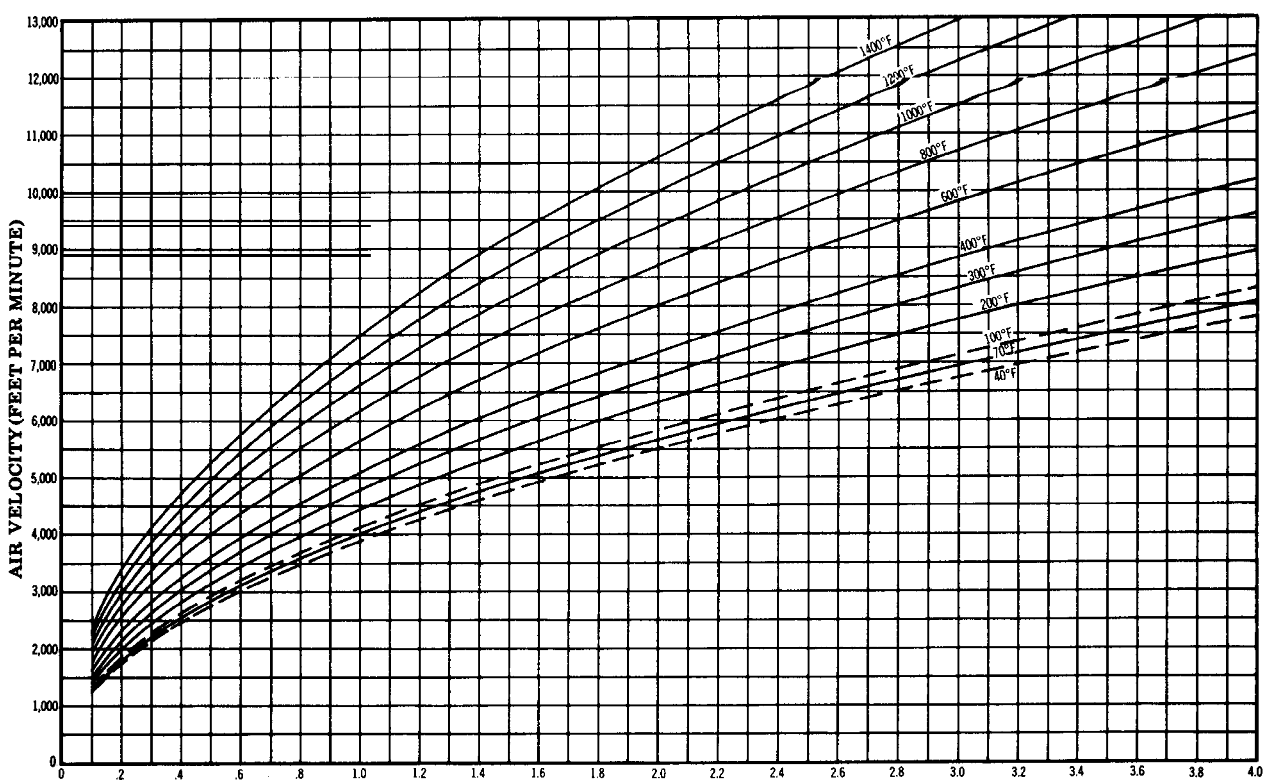
The formula for velocity pressure given above is actually a simplification based on the assumption that the air is at 70 °F and at atmospheric pressure. For higher temperature air, the same velocity would produce a lower velocity pressure due to the reduction in air density. For air at temperatures other than 70 °F, the relationship between velocity and velocity pressure is:

$$V = 1{,}097 \times \frac{VP}{d}$$

where: V = velocity, fpm
VP = velocity pressure, in. w.c.
d = density of air, lb/ft3



VELOCITY PRESSURE (INCHES OF WATER)



VELOCITY PRESSURE (INCHES OF WATER)

FIGURE 2-26
Air Velocity Flow Charts

For velocities of air at temperatures other than 70 °F, Figure 2-26 gives tabulated velocity pressures.

Instruments such as the rotating vane anemometer and deflecting vane velometer actually depend on velocity pressure to move an object against a resistance. The velocity readout is based on air at 70 °F. If the air being measured is significantly different from 70 °F, the velocity reading must be corrected by the ratio of absolute temperatures. This can be simply done by:

$$V = \frac{460 + T}{460 + 70} \times \text{instrument reading}$$

where: V = actual corrected velocity

T = actual air temperature, °F

We have seen that velocity pressure is the force exerted due to the velocity of air. Now consider the fan and duct system shown in Figure 2-27. If the fan is discharging into the duct, and the duct is closed (no air can escape), then a pressure will build up inside the duct. Even with no air velocity, this pressure will be exerted in all directions, and against the duct walls. If we place the manometer probe inside the duct, the amount of pressure will be registered. This is called static pressure.

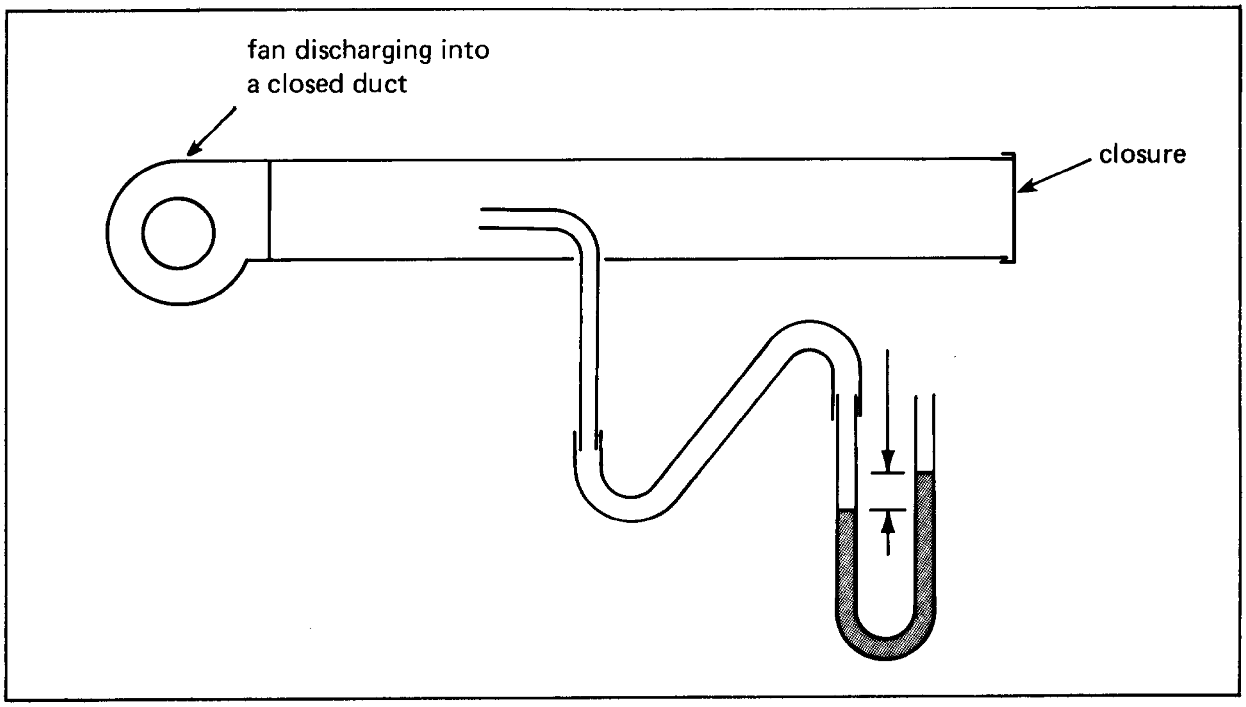


FIGURE 2-27
Velocity Probe is Affected by Static Pressure,
Even When There is no Airflow

In Figure 2-28, a duct system is shown with both velocity and static pressure. The velocity in the duct connected to the suction side of the fan is the same as the velocity in the discharge duct. Therefore, the velocity pressure would be the same everywhere in this duct system. However, the static pressure is different everywhere along the length of the duct system. Just as the air enters the suction duct, the static pressure would be very close to zero (atmospheric pressure). Just at the suction side of the fan, the static pressure would be the lowest of anyplace in the system. At the discharge side of the fan, the static pressure would be the highest, and it would gradually be reduced until just before the end of the discharge duct, the static pressure would once again be zero. When measuring airflow in ducts, we must make sure that we account for the effects of static pressure, which are independent of the effect of velocity pressure.

Consider the duct probes shown in Figure 2-28. Probe A is facing into the direction of airflow and senses both velocity pressure and static pressure. Probe B has an opening perpendicular to the direction of airflow. It is unaffected by velocity and it senses static pressure only. When we connect these two probes to the manometer as shown, the reading will correspond to the velocity pressure only. Regardless of the static pressure, the same static pressure acts on both legs on the manometer, and each cancels the effect of the other.

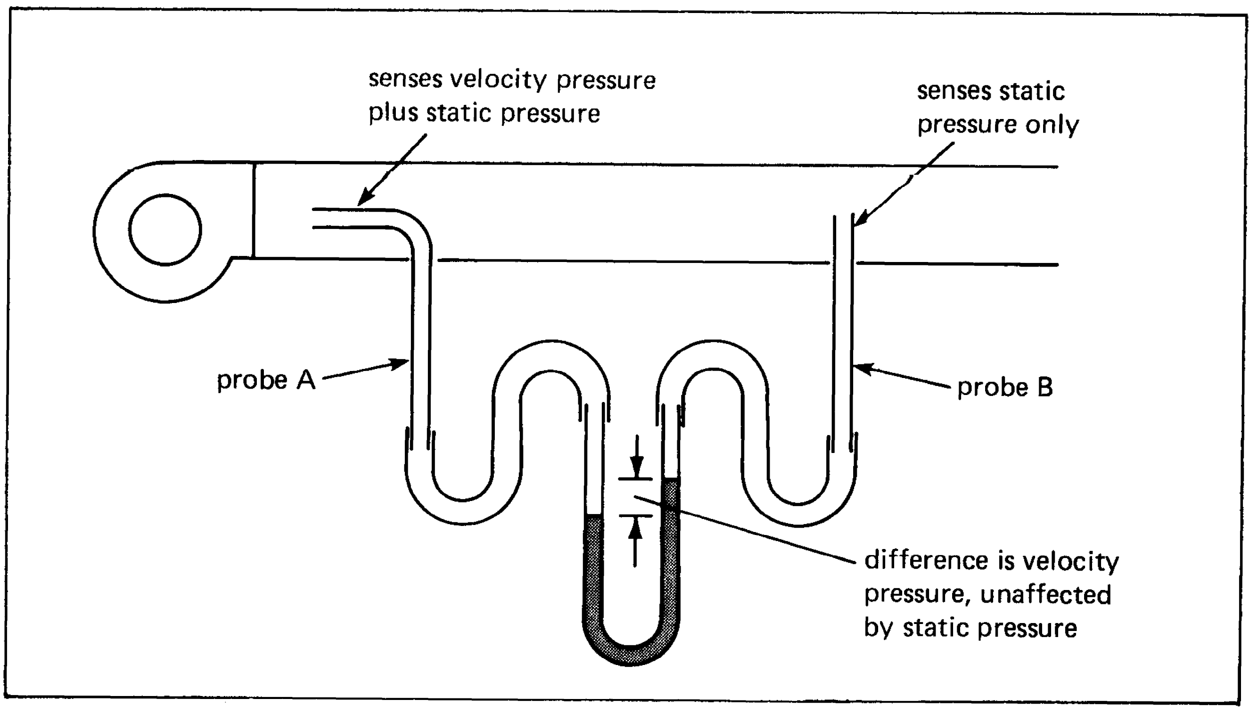


FIGURE 2-28
Velocity Flow Measurement in Duct

2.16 PITOT TUBE. The pitot (pronounced pee-toe) tube is a device that senses both velocity and static pressure. It is inserted into a moving air stream and connected to a manometer. The inside of the pitot tube is shown in Figure 2-29. It is actually a tube within a tube. The inside tube senses pressure at the end of the tube facing in the direction of airflow. It senses the total pressure within the duct; that is, it senses velocity plus static pressure. The hose connection to send this pressure to the manometer is at the bottom of the pitot tube. A second hose is connected to the fitting on the side of the pitot tube and reads pressure in the area between the two tubes. This pressure is created by a series of small holes around the circumference of the outer tube. As each of these holes is perpendicular to the direction of airflow, they will sense static pressure only.

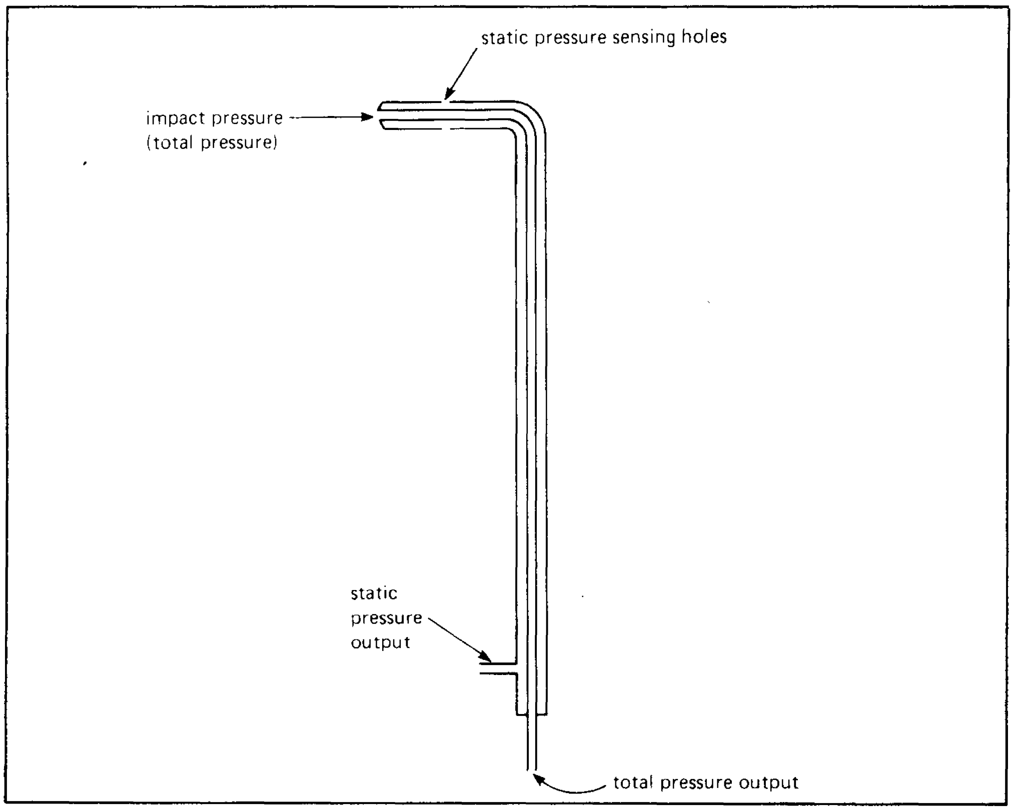


FIGURE 2-29
Pitot Tube Internals

The pitot tube may be used with a manometer to measure velocity pressure or the static pressure in a duct by hooking it up as in Figure 2-30. It does not matter if the static pressure within the duct is positive or negative.

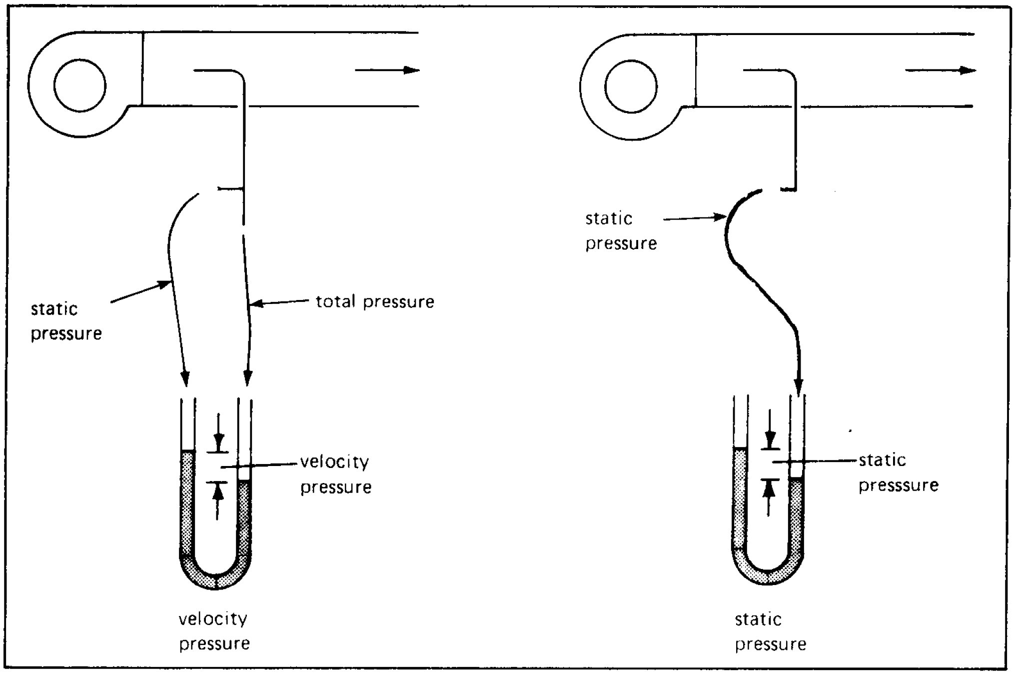


FIGURE 2-30
Using a Pitot Tube and Manometer to Measure
Velocity or Static Pressure

2.17 INCLINED MANOMETER. The manometers discussed up to this point have been U-tube manometers. That is, in order to measure a static or velocity pressure of 1.0 inches w.c., a vertical displacement of 1.0 inches in a tube must be read. However, in reading commonly encountered velocity pressures in air conditioning ducts, pressures of only a few hundredths of an inch must be read. With a U-tube manometer, it is impossible to read hundredths of an inch with any degree of accuracy. For accuracy, the inclined manometer is used (Figure 2-31). It essentially spreads out the 1 inch of vertical column into a much longer horizontal length. The actual vertical rise is still 1 inch, but it becomes much easier to read accurately when the scale has been expanded.

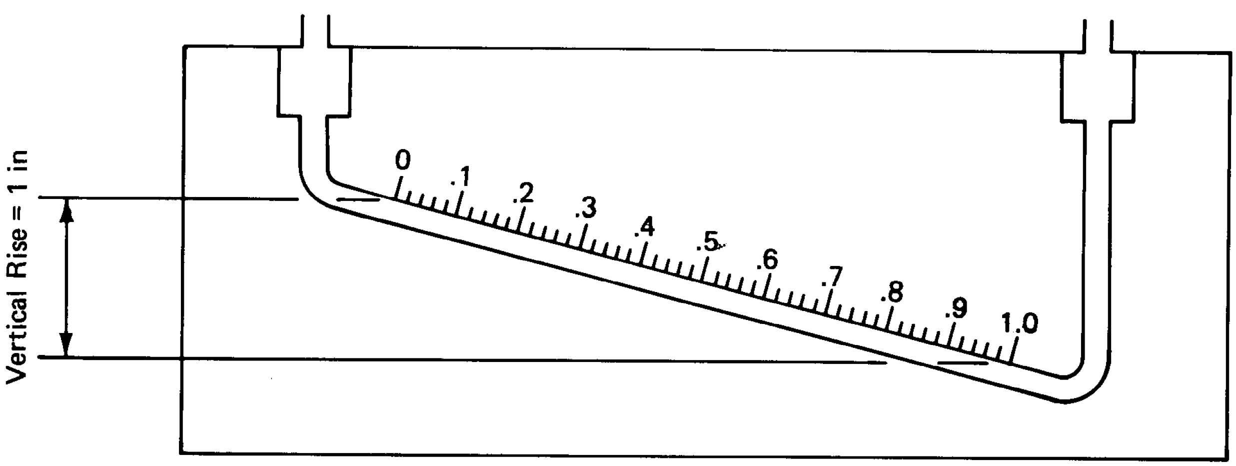


FIGURE 2-31
Inclined Manometer

Some inclined manometers do not use water as the fluid, but instead, a red gage oil is used. This has the advantage of easier readability. Also, there is a problem using water because when the water evaporates, the solids left behind can cloud the inside of the manometer tube. Manometers that use red gage oil will not have an actual vertical rise that matches the numbers on the gage. This is explained by the fact that the gage oil has a specific gravity of 0.86 compared to that of water. The exact readings from the water and red gage oil may not be used interchangeably. The correct fluid to match the actual calibration of the manometer must be used.

When using inclined manometers, it is important that they be mounted in a level position when taking readings. Many units are supplied with an integral water level, and magnetic mounts or screw-type feet for attaining a level position. Once leveled, the gage must be set to zero by adjusting the amount of fluid allowed to remain in the reservoir. When using the inclined manometer, it is important to anticipate which pressure sensing port will see the greater pressure, and always have the greater pressure pushing the fluid down the scale. Some popular ranges available in inclined manometers are 0 to 0.25 inch, 0 to 0.50 inch, and 0 to 1.0 inch. For measurements above 1.0 inch, a U-tube manometer may be used with acceptable accuracy.

NOTE: When hooked up backwards, the technician will face the embarrassing task of cleaning up all the oil which has been either blown out of the gage onto the floor, or sucked up into the duct. Also, it is important that the pressure differential being measured does not exceed the range of the manometer.

2-18 PRESSURE GAGES. Dial type pressure gages are also used in conjunction with pitot tubes in the same way as inclined manometers. Figure 2-32 shows an industry standard commonly referred to as a magnehelic gage. These gages are available in all the same ranges as inclined manometers. They each have a high pressure and a low pressure connection. When measuring a negative static pressure, for example, you would attach the negative pressure sensed by the pitot tube to the low pressure connection, and leave the high pressure connection open to the atmosphere. Pressure gages have the advantage of being easy to use, relatively rugged, and accurate. However, only the manometer can be counted upon to never get out of calibration.

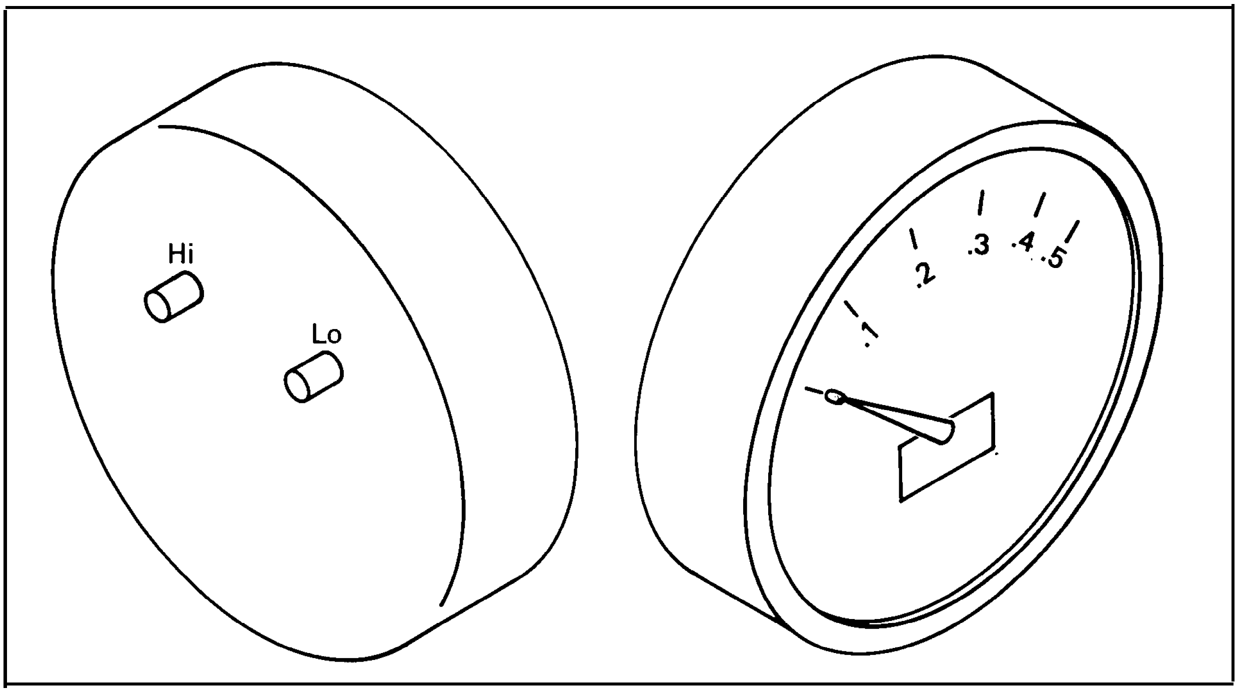


FIGURE 2-32
Dial Pressure Gage

2.19 ALNOR VELOMETER. The Alnor velometer shown in Figure 2-33 is very commonly used by service technicians to measure velocity and static pressure. The various scales on the readout unit correspond to the ranges of the various probes. Velometers are easy to use and are quite accurate when properly calibrated.

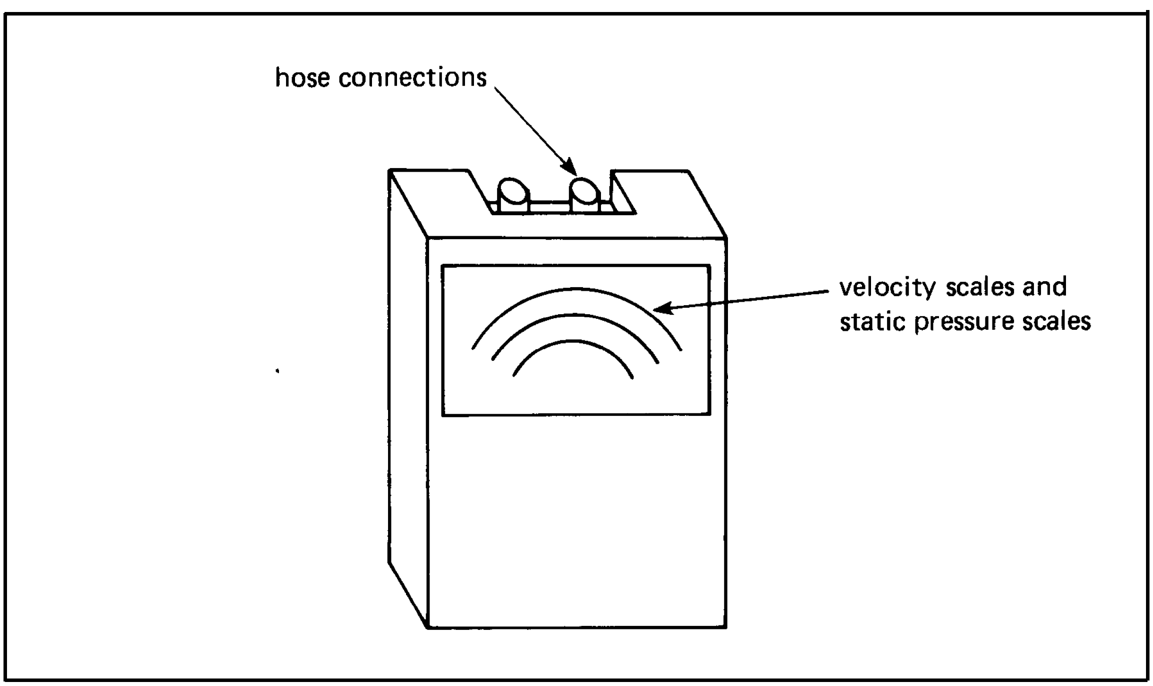


FIGURE 2-33
Alnor Velometer

2.20 DIFFUSER CFM. It is frequently necessary to measure the airflow from room-type diffusers such as that shown in Figure 2-34. While the velocity can be measured, it is difficult to know the discharge area of the diffuser in order to calculate cfm. For new construction, the air balancer will have the specifications for each diffuser. This specification will include the effective area (Ao) of that diffuser. Multiplying the velocity by the effective area will give you the actual cfm. For the technician taking measurements on an existing job, the diffuser specifications are sometimes unavailable. In this case, data from similar diffusers may be used.

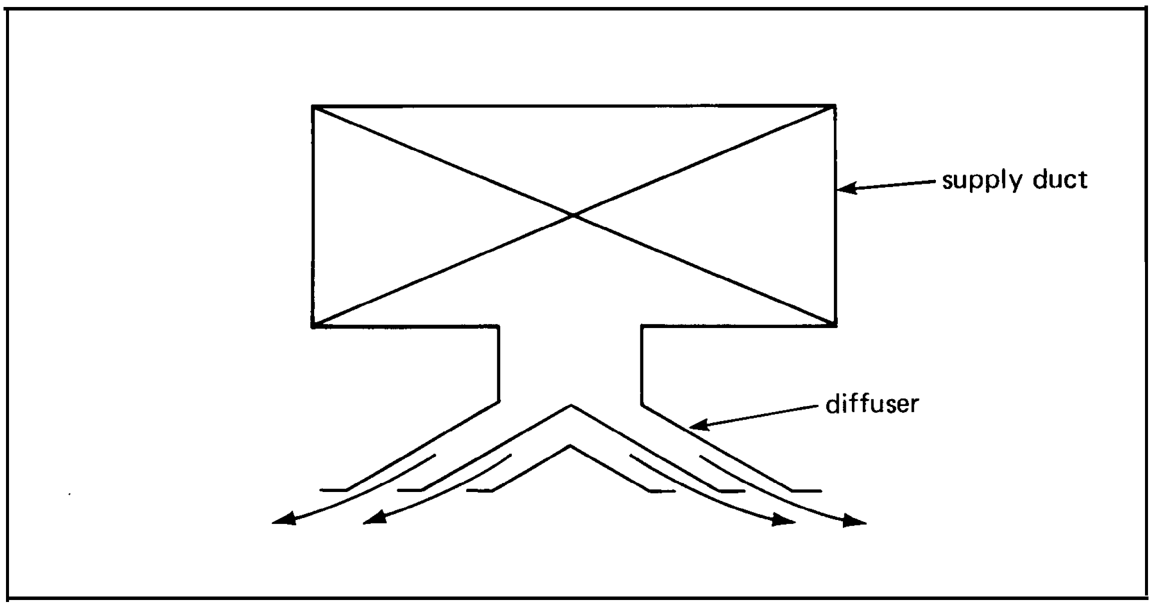


FIGURE 2-34
Ceiling Diffuser

Another option is using the type of hood shown in Figure 2-35. The top of the hood is placed against the ceiling so that all air from the diffuser enters the funnel-shaped hood. The hood then tapers down to a l- by l-foot area at the bottom. Velocity measurements taken at the bottom may then be multiplied by one square foot in order to get cfm. In addition to the obvious advantages of this hood, there is the added advantage of being able to take all measurements in rooms with normal ceiling height from the floor, without the need for a ladder.

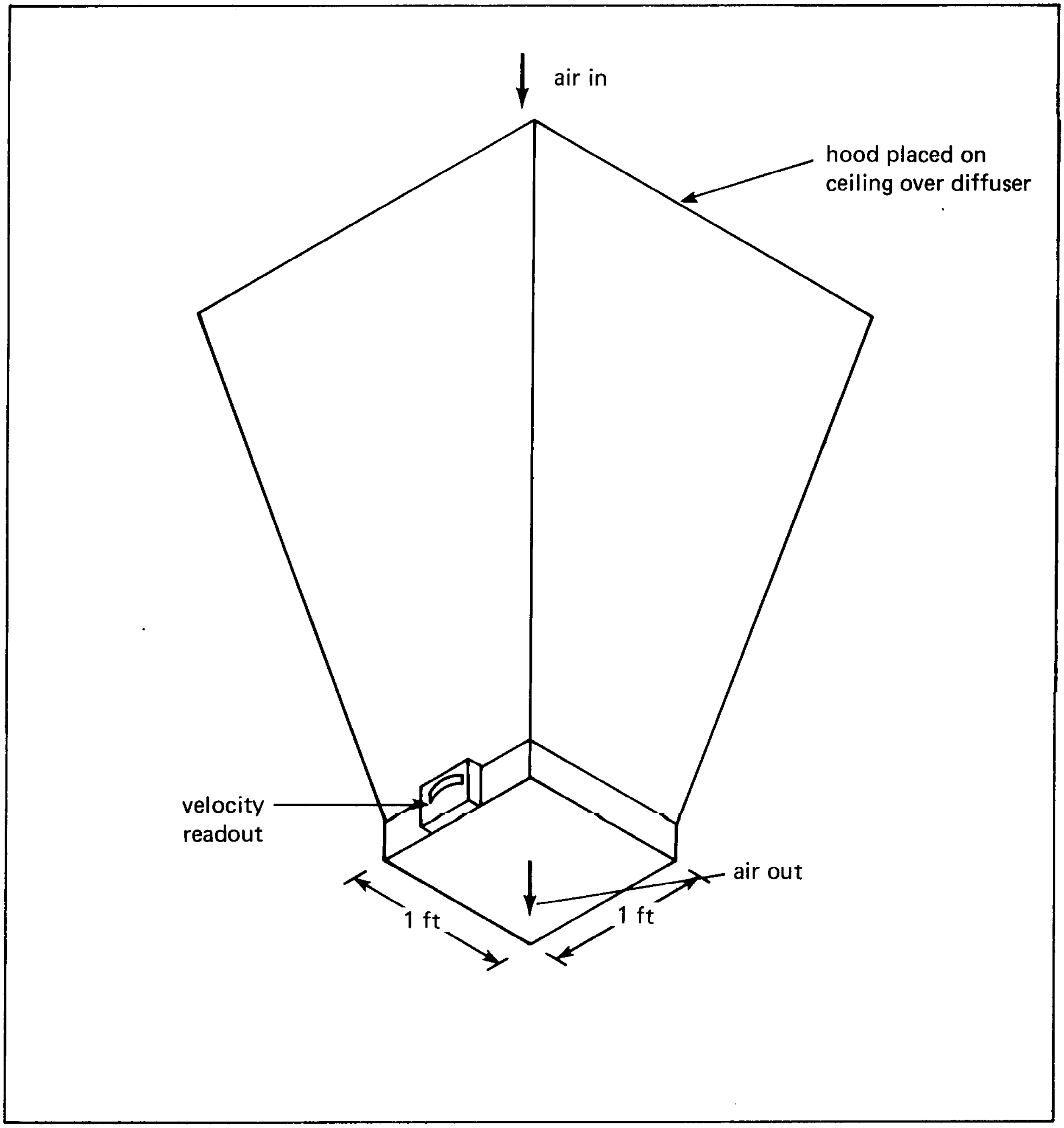


FIGURE 2-35
Flow Hood for Measuring Airflow From Ceiling Diffusers

2.21 TEST PROCEDURE. Testing and balancing of a mechanical ventilating system is necessary to insure proper operation. The most common complaints from occupants who are too hot or too cold are frequently traced to an air system which is not balanced properly.

At the time of initial installation of the duct system, the design data should be recorded. After initial start-up, balance the system so that each air outlet is adjusted to deliver its rated cfm. After the system is balanced, take static pressure measurements at the fan inlet and outlet, and at the inlet and outlet of each coil or filter. This data may be used as reference data for future troubleshooting. By comparing measurements at a future date to the original measurements, the maintenance technician may determine if problems are due to equipment or if occupants in the building have taken it upon themselves to rebalance the system.

CHAPTER 2. SELF-STUDY QUESTIONS

Q2-l What three pressures are measured by the pitot tube?

Q2-2 What is the average velocity (fpm) and flow rate (cfm) for the following nine-point traverse of an 18-inch square duct?

300	450	300	
450	600	450	(18 inches)
300	450	300	

Q2-3 A duct measures 18 by 20 inches. The area of the duct in square feet is:

() a. 2.0 ft^2
() b. 2.5 ft^2
() c. 3.0 ft^2
() d. 3.5 ft^2

Q2-4 If the average velocity in question 3 above is 400 ft/min, the quantity of air flowing through the duct in ft^3/min (cfm) is:

() a. 800 cfm
() b. 1,000 cfm
() c. 1,200 cfm
() d. 1,400 cfm

Q2-5 A pitot tube traverse of a 4-ft^2 duct gives a reading of 2 total pressure (TP) and 1.50 static pressure (SP). The cfm flowing through the duct is approximately: (Hint $\longrightarrow$ $V = 4005 \times \sqrt{vp}$)

() a. 12,200 cfm
() b. 11,350 cfm
() c. 10,960 cfm
() d. 8,000 cfm

Q2-6 When making a standard 16-point duct traverse for an 18- by 18-inch square duct, the first hole drilled should be how many inches away from the duct wall?

() a. 4.50 inches
() b. 12.00 inches
() c. 9.75 inches
() d. 2.25 inches

Q2-7 When making a standard 20-point duct traverse for a 20-inch diameter round duct, the third measurement should be made how many inches from the wall?

() a. 0.50 inches
() b. 1.60 inches
() c. 2.90 inches
() d. 4.50 inches

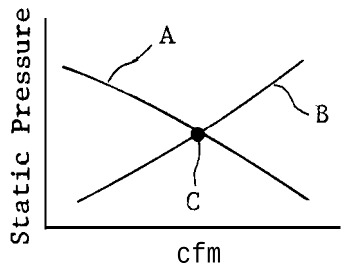
Q2-8 If the average measured velocity through the 20-inch diameter is 600 ft/min, the quantity of air flowing through the duct is:

() **a.** 1,320 cfm
() **b.** 188,496 cfm
() **c.** 5,236 cfm
() **d.** 1,667 cfm

Q2-9 A system is handling 10,000 cfm at 600 rpm, 1.1 inch of static pressure, and 6 hp. It is desired to increase the airflow by 10 percent. Calculate the new cfm, rpm, static pressure, and hp.

Q2-10 The pulley on a shaft rotating at 600 rpm is increased from 4 to 4.5 inches. What is the new rpm for the pulley?

Q2-11 For the two curves shown below, identify the system curve and the fan curve.



Q2-12 What is point C in No. 11?

CHAPTER 3. NATURAL OR GRAVITY VENTILATION SYSTEMS

3.1 GENERAL. Natural ventilation systems do not depend upon fans and motors to move air. Rather, wind velocity and the natural buoyancy of warm air provide the motive force for moving air.

3.2 WIND-OPERATED FANS. Figure 3-l shows a turbine-type exhauster with a fan attached to a turbine. As wind passes the turbine, it turns and drives the fan to remove the warm air. The driving force on the fan is minimal, and for effective operation, the bearings that allow free rotation must be clean and lubricated. The wind-powered turbine fan sitting stationary atop a roof on a windy day is a testimonial to poor maintenance of bearings.

3.3 GRAVITY VENTILATORS. Gravity ventilators have no moving parts. They merely provide the necessary openings to allow warm air to rise in a building and exit through the roof, and for cooler air to enter from a lower elevation. Figure 3-2 shows a gravity ventilator which may be located on a rooftop. The taller the building, the better a gravity ventilation system may be expected to operate.

Another type of gravity ventilator is the louvered roof monitor, shown on figure 3-3, which provides an exit path for the warm air in the building.

One of the advantages to gravity systems is the low maintenance requirements. Semiannual inspection will reveal any need for painting or repair. Dampers should be closed at the beginning of each winter season, and opened prior to the need for summer ventilation.

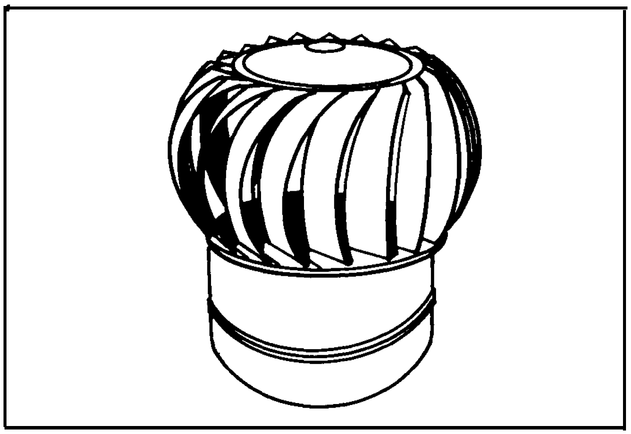


FIGURE 3-l
Wind Driven Turbine Ventilator

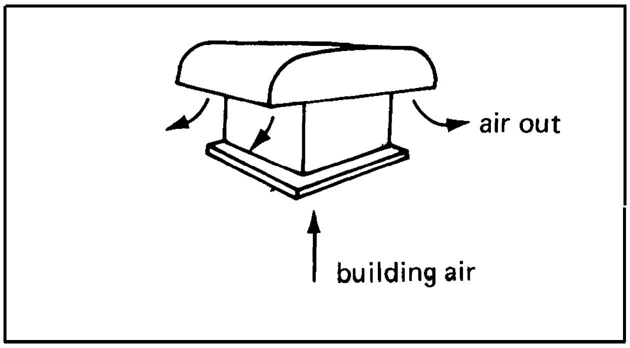


FIGURE 3-2
Gravity Type Roof Ventilator

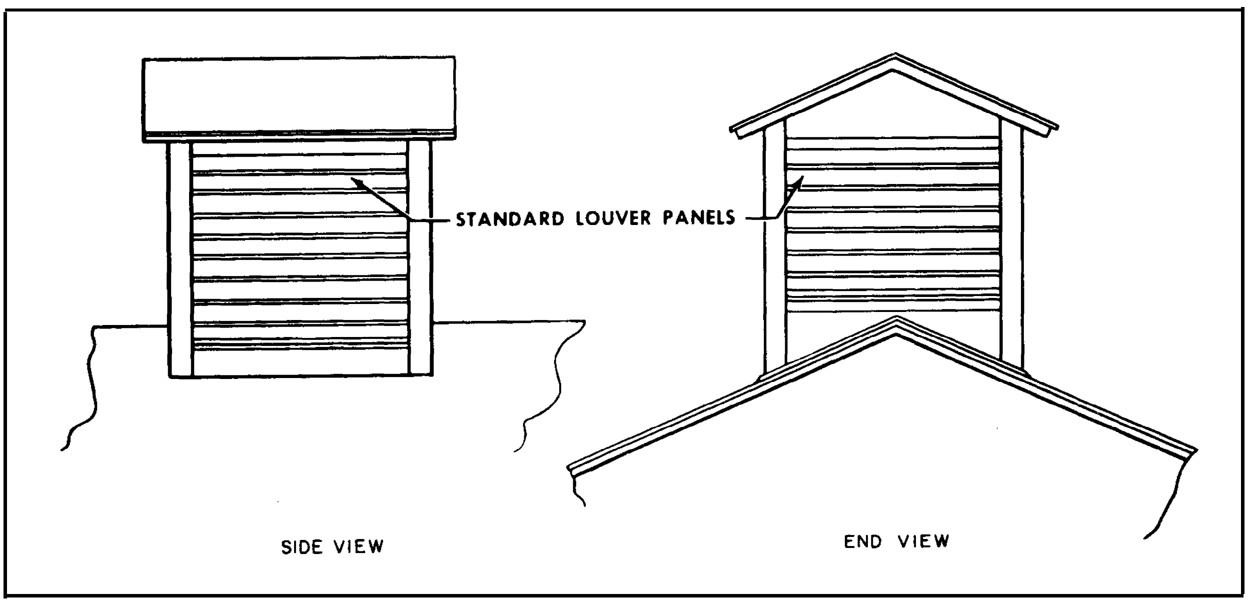


FIGURE 3-3
Louvered Roof Monitor

CHAPTER 3. SELF-STUDY QUESTIONS

Q3-l Bearing maintenance for a wind operated turbine is not critical?

() True () False

Q3-2 Natural ventilation systems require a motor as part of their operation?

() True () False

Q3-3 Gravity ventilators work the same on short buildings as on tall buildings?

() True () False

Q3-4 Natural ventilation systems do not require fans and motors to move air?

() True () False

Q3-5 As the air in the building heats up, it rises and exits out the roof ventilators.

() True () False

Q3-6 Natural and gravity ventilation systems should be inspected semiannual.

() True () False

CHAPTER 4. INDUSTRIAL VENTILATION SYSTEMS

4.1 GENERAL. Industrial ventilation is defined as "the use of supply and exhaust ventilation to control emissions, exposures, and hazards in the workplace." General ventilation systems such as heating, ventilation, and air conditioning systems (HVAC) are used for comfort control (e.g., temperature, humidity, and odor).

Design and installation of the system is usually done in accordance with the guidelines found in MIL-HDBK-1003/17. Industrial Ventilation Systems; and Industrial Ventilation, published by the American Conference of Governmental Industrial Hygienists.

The objectives of the industrial ventilation system are to: (a) control airborne concentrations, dust, and vapor to keep below explosive or flammable levels, and (b) control toxic particulates, gases, and vapors to keep below exposure levels called threshold limit values (or TLVs) harmful to workers.

There are five basic components to an industrial ventilation system. Each component (hood, duct, air cleaner, fan, and stack) is critical to the operation and success of the system.

4.2 HOOD. The hood or capture device is any device whose purpose is to receive and contain the emitted contaminant at the source of generation.

The three basic types of hoods are:

a. The enclosing hood. Examples are glove box, lab hoods, bench hoods, grinder hoods, and others with four of more sides.

b. The capture hood. These are hoods with one to three sides. A welding snorkel type hood is typical. Others include side draft, down draft, and push pull hoods.

c. The receiving hood. These hoods are designed to receive the emission source (which has some initial velocity imparted to it by the emitting source). A canopy hood is a receiving hood because it receives hot rising air and gases. A small hand held tool hood may be a receiving hood.

Figure 4-1 shows the three different types of hoods.

4.2.1 Capture and Face Velocities. The capture or face velocity is the velocity of the air into the hood that is required to remove the contamination. The velocity that is required should be obtained from the design information. However, if the design information is not available, then Table 4-l can be used.

The velocity should be measured across the face of the hood. The measurement can be made using various flow measuring devices (velometers, discussed in Chapter 2) or smoke generators. A minimum of 16 measurements should be taken for rectangular hood openings. If the measurement is taken over a round opening, then 20 measurements should be taken and averaged.

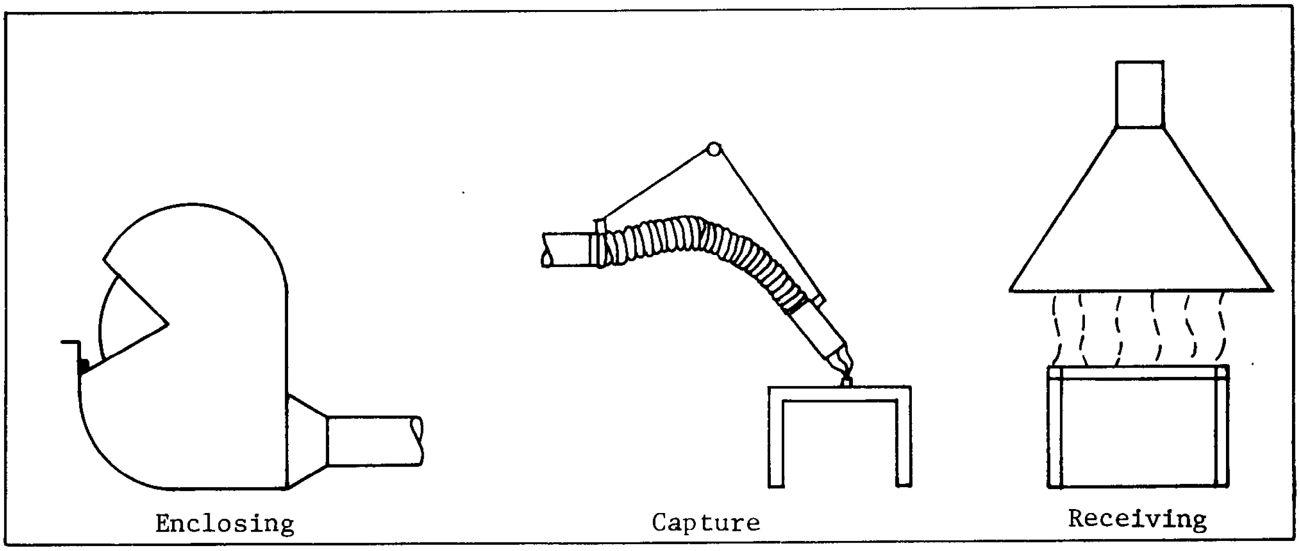


FIGURE 4-l
Hood Types

Smoke can be used to provide a rough estimate of face velocity. Squeeze off a quick burst of smoke. Time the smoke plumes travel over 2 feet and calculate the velocity in feet per minute. For example, if it takes 2 seconds to travel 2 feet, the velocity is 60 fpm.

Most hood face velocities are on the order of 100 fpm, which is the equivalent of a very light breeze. Any competing air currents introduced by open windows, fans, etc., can destroy the capture of the contaminations by the hood. Therefore, all competing air currents should be removed or reduced.

4.3 AIR CLEANERS. Air cleaning devices remove contaminants from an air or gas stream. They are available in a wide range of designs to meet variations in air cleaning requirements. There are two broad classes of cleaners: air filters (for gasses and vapors), and dust collectors.

4.3.1 Air Filters. Air filters are designed to remove dust concentrations of the order found in outside air and are employed in ventilation, air conditioning, and heating systems where dust loadings seldom exceed one grain per thousand cubic feet of air (a typical atmospheric dust concentration is 0.05 grains per thousand cubic foot). Commercially available fabric filters use fabric configured as:

a. Tubes or stockings,
b. Flat bags or
c. Pleated cartridges.

Gas and vapor filters include:

a. Wet scrubbers,
b. Adsorption and absorption collectors,
c. Incinerators,
d. Catalytic scrubbers or
e. Chemical reaction scrubbers.

Maintenance of these systems should be done in accordance with the manufacturer's instructions.

TABLE 4-1
Range of Capture Velocities

Condition of Dispersion of Contaminant	Examples	Capture Velocity (fpm)
Released with practically no velocity into quiet air.	Evaporation from tanks; degreasing, etc.	50-100
Released at low velocity into moderately still air.	Spray booths; intermittent container filling; low speed conveyor transfers; welding; plating; pickling	100-200
Active generation into zone of rapid air motion	Spray painting in shallow booths; barrel filling; conveyor loading; crushers	200-500
Released at high initial velocity into zone of very rapid air motion.	Grinding; abrasive blasting, tumbling	500-2000

In each category above, a range of capture velocity is shown. The proper choice of values depends on several factors:

Lower End of Range	Upper End of Range
1. Room air currents minimal or favorable to capture.	1. Disturbing room air currents.
2. Contaminants of low toxicity or of nuisance value only.	2. Contaminants of high toxicity.
3. Intermittent, low production.	3. High production, heavy use.
4. Large hood-large air mass in motion.	4. Small hood-local control only.

4.3.2 Dust Collectors. Dust collectors are usually designed for much heavier loads from industrial processes where the air or gas to be cleaned originates in local systems or process stack gas effluents. Loading will vary from less than 0.1 to 20 grains or more per cubic foot.

The various dust collectors include:

1. Gravity separators --> balloon flues, clean out ducts, settling chambers,

2. Impact devices --> baffled ducts,

3. Centrifugal collectors --> cyclones,

4. Filtration devices --> furnace filters, paint filters, bag houses, rolling filters,

5. Electrostatic Precipitators --> single and two stage or

6. Wet scrubbers

Figure 4-2 shows a cyclone dust separator.

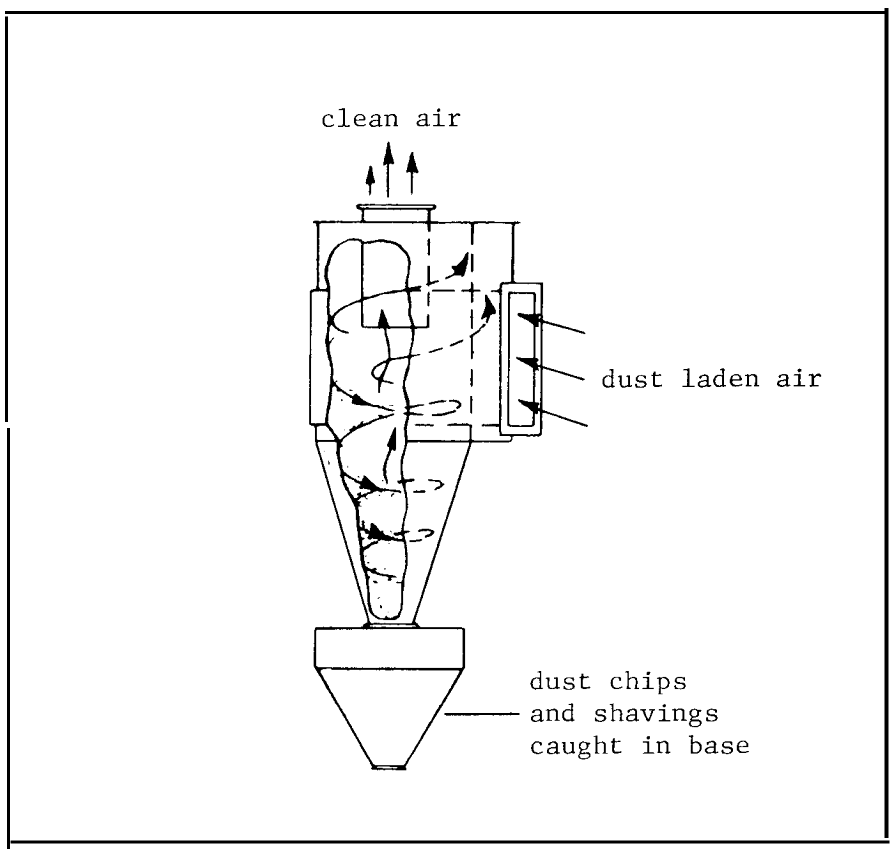


FIGURE 4-2
Cyclone Centrifugal Separator

4.4 DUCT SYSTEMS. Duct systems can be made of galvanized metal, PVC plastic, ABS plastic, and reinforced plastic. Metal duct may be spiral wound with joints and flexible duct may be made of many different materials.

Ducts that carry corrosive vapors, gases, and mists require materials that are corrosion resistant. Ducts that carry dust laden air should have cleanouts located near elbows, junctions, and vertical runs.

Air leakage into industrial ventilation system should be maintained at less than 1 percent. Any leaks that appear should be repaired immediately. In many operating systems, poor performance can be traced to cleanouts that have been opened and not properly closed.

4.5 FANS. The fans used in an industrial ventilation system depend upon the function of the system. Items considered in the selection of the fan are: volume of air required; fan static pressure; type of material being handled through the fan such as fibrous, explosive, or inflammable; space limitations; noise; operating temperature; efficiency; and corrosive applications.

The types of fans used in industrial ventilation systems may be axial, centrifugal, or special design such as an air foil type. However, the work-horse for most exhaust systems is the centrifugal straight or radial blade design as shown in Figure 2-10. This type of fan is used to reduce dirt buildup in the blades.

4.6 TESTING OF VENTILATION SYSTEMS. Airflow measurements are often needed to determine if the industrial ventilation system is operating properly. Measurements can be done following the procedures outlined in the current volume of the Industrial Ventilation Handbook by the American Conference of Governmental Industrial Hygienists. Measurements are made using the equipment and techniques outlined in Chapter 2.

CHAPTER 4. SELF-STUDY QUESTIONS

Q4-1. List two objectives of industrial ventilation.

a.

b.

Q4-2. An evaporative tank is located in an area with competing air currents. What would be the recommended capture velocity if the design information was not available?

Q4-3. If the tank hood in No. 2 above has an open area 3 ft by 2 ft, what is the air flow quantity (cfm) into the hood?

Q4-4. Name the five major components of an industrial ventilation system.

Q4-5. What is the typical atmospheric dust concentration in grains/thousand cubic foot?

(a) 1.0 grains/thousand ft^3
(b) 0.05 grains/thousand ft^3
(c) 0.005 grains/thousand ft^3

Q4-6. Fill in the blanks;

(a) The ________ or _____ velocity is the velocity of the air into the hood that is required to remove the contamination.

(b) To measure velocity across the face of the hood a minimum of ____ measurements are required for a rectangular hood and ____ measurements for a round opening hood.

CHAPTER 5. MECHANICAL VENTILATION COOLING SYSTEMS

5.1 GENERAL. Mechanical ventilation cooling systems, better known as "Whole House Fans" are ventilation systems that rely on "wind chill" or evaporative cooling of the skin for its cooling effect.

The building is normally ventilated at the rate of 30 to 60 air changes per hour for best cooling effect. Air movement velocity is normally limited to 100 feet per minute (fpm) or less in residences and office areas as higher velocities will cause light objects to flutter and move and produce annoying drafts. In industrial applications, local velocities up to 4,000 fpm may be used. However, velocities greater than 1,000 fpm may disrupt the performance of nearby local exhaust systems and care must be taken to direct air motion to prevent such interference. The current edition of Industrial Ventilation, by the American Conference of Governmental Industrial Hygienists, has a chapter on ventilation for heat control and should be consulted regarding velocities and air movement in the industrial workplace.

For dry bulb temperatures over 95 °F, ventilation cooling does not appear to work. Current practice is to limit its application in offices and residences to temperatures of 85 °F.

5.2 FANS AND INSTALLATION PRACTICES. The fans used for mechanical ventilation are generally propeller types and may be direct or belt driven, have sealed bearings or bearings requiring periodic oiling. Multiple speed arrangements are also available. The fans may be installed directly on an outside wall, in a penthouse on a flat roof, or in a hallway which vents into an attic as shown in Figures 5-l and 5-2.

The net free area of the exhaust opening must be as large or larger than the area of the fan in order for the fan to operate properly. Some installations will have shutters that automatically open and close with the operation of the fan. For proper operation, the net free area of the shutter opening should be equal to or greater than the area of the fan.

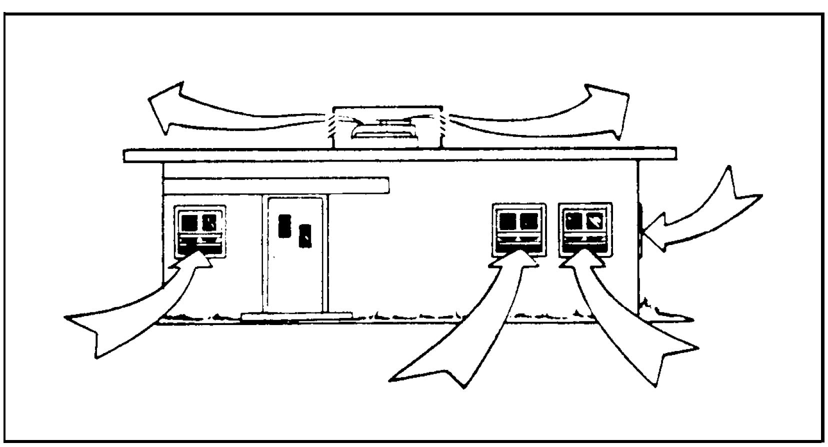


FIGURE 5-1A
Roof Mounted Fan With Penthouse

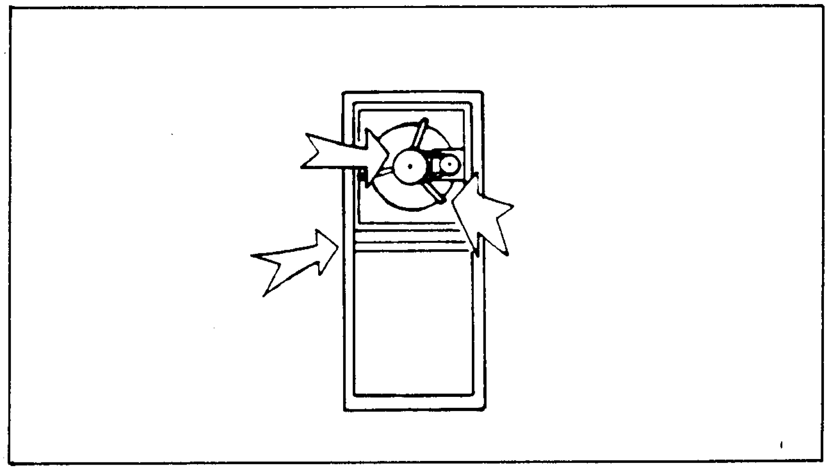


FIGURE 5-1B
Fan Installed in Upper Part of Window

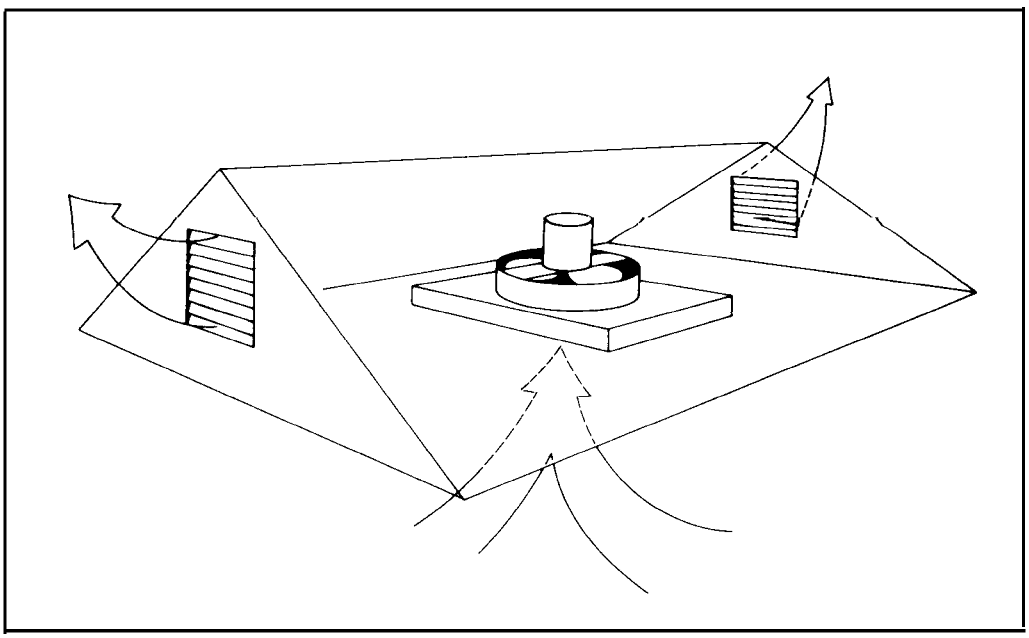


FIGURE 5-2
Typical Attic Installation of a Mechanical
Ventilation Cooling (MVC) Fan

5.3 MAINTENANCE AND OPERATION. General maintenance and inspection of mechanical ventilation cooling systems should be done in accordance with the information provided in Chapter 6, Maintenance Instructions, and Appendix A, Maintenance and Inspection Schedule.

The following are problems specific to mechanical ventilation cooling systems along with some possible causes.

Shutter slams shut when fan turns off:

- Exhaust area in attic is not large enough.
- Closing spring tension is too great.
- Inlet opening area into building is not large enough.

Fans make a deep "wooping" type sound when in operation:

- Exhaust area in attic is not large enough.

Moisture appears on walls and on floors when air conditioning is turned off and mechanical ventilation fan is turned on:

- Moisture in the air is being condensed because the surfaces in the structure have a temperature below the dew point of the air. Continued operation of the fan will warm up the surfaces and the moisture will evaporate.

Window screens tend to become plugged with dirt:

- The screens are acting as air filters and removing airborne dust and dirt particles. This is a common occurrence with these types of systems. Screens require increased cleaning or transfer grills with filters in them can be installed on the exterior walls.

Standing pilots tend to blow out when fan is in operation:

- Stoves: This may occur when excessive drafts blow across the stove. Opening different windows should resolve the problem.

- Water heaters: Excessive negative pressures will cause the outside air to move down the heater flue and blow out the pilot. Open more windows. Provide an outside source of combustion air and seal the area where the water heater is located from the area being ventilated, will eliminate the problem.

CHAPTER 5. SELF-STUDY PROBLEMS

Q5-1. A building measures 50 ft by 100 ft and the rooms are 8 ft high. What is the air flow quantity that is required to provide 60 air changes per hour?

a. 667 cfm
b. 20,000 cfm
c. 40,000 cfm
d. 20,000 fpm

Q5-2. A hallway measures 4 ft x 8 ft. If the maximum velocity rate is limited to 100 feet per minute, what is the air flow quantity?

a. 3,200 cfm
b. 1,600 cfm
c. 53.3 cfm
d. 800 cfm

Q5-3. A mechanical ventilation cooling fan has a diameter of 3 feet. What is the minimum exhaust area required?

a. 3.0 sq. ft.
b. 9.4 sq. ft.
c. 7.1 sq. ft.
d. 6.0 sq. ft.

Q5-4. List two reasons a shutter might be slamming shut when the fan turns off.

a.

b.

CHAPTER 6. MAINTENANCE INSTRUCTIONS

6.1 FANS AND BLOWERS. The maintenance frequency required for fans and blowers is influenced tremendously by the operating conditions. Fans operating under dusty or corrosive conditions require far more frequent attention than those handling clean air in a dry location.

Fan blades may be inspected for buildup of foreign material which may cause imbalance or loss of capacity. Centrifugal fans with FC wheels are especially susceptible to capacity loss due to dust packing the blades. Applications where buildup of foreign material should be anticipated are paint spray booths, kitchen exhausts, and machine shops where normal dust in the air is more likely to stick to a coated fan blade or support.

When cleaning accumulations from fan blades or wheels, use air hoses, solvents, or soft brushes. Take care not to bend propeller fan blades when cleaning, as this will cause a permanent vibration and require the replacement of the blade.

6.2 MOTORS. Information on the maintenance of motors that is generally applicable to fan motors is included in MO-116, Facilities Engineering Electrical Interior Facilities.

6.3 FAN BEARINGS. Fan bearings of three types are in general use:

a. Ball bearings
b. Roller bearings
c. Sleeve bearings

The failure of bearings may usually be attributed to overheating or overloading. The condition of bearings may be indicated by temperature during operation. Temperatures over 140 °F should be investigated further for proper lubrication and load. For fans operating in high temperature environments, special high temperature lubricants should be used. Standard lubricants lose too much viscosity above 140 °F. In all cases, the lubricant recommended by the manufacturer is the best choice.

When lubricating fan bearings, do not use large quantities of grease or oil. A few drops of oil properly distributed over the running surfaces of the bearing will provide satisfactory lubrication. Too much lubricant may cause high operating temperatures due to the working and churning of the lubricant by the rolling elements. A potential fire hazard can also be created by too much lubricant.

Overloading of bearings is most commonly caused by belt tensions that are too tight. The fan bearing and motor bearing are both in jeopardy from over-tightened belts.

6.4 VIBRATION. Vibration of fans is a symptom of another problem. Vibration may be caused by:

a. Poorly aligned bearings, couplings, or drive
b. Bent shaft
c. Weak or inadequate foundation
d. Loose set screws on fan-to-shaft connection
e. Loose bolts on bearings
f. Unbalanced wheel

If items a through f above are not apparent causes, the wheel balance must be checked. Make sure that the fan blades are clean. Check for static imbalance as follows:

1. Run the fan for 10 minutes to bring the lubricant to operating temperature. Shut the fan down.

2. Block off fan inlet or outlet to prevent fan rotation due to air currents.

3. Lock out the electrical supply.

4. Disconnect the motor (belt driven fans).

5. Give the fan wheel a vigorous rotation by hand, and mark the part of the fan which stops at the top.

6. Repeat, alternating the direction of the rotation. Continue repeating until a definite pattern becomes apparent.

7. Add a metal weight at the light point on the wheel, and test again.

If no static unbalance is apparent, or if a satisfactory balance can be produced with weights, the fan wheel may have a dynamic unbalance. A balancing contractor or the manufacturer should be contacted to correct a dynamic unbalance. Table 6-l shows a summary of fan problems and remedies.

TABLE 6-1
Summary of Fan Problems and Remedies

Problem	Causes	Remedy (Section)
Vibration	Unbalanced fan wheel or rotor	Check for dirt or foreign material. Check for water in blades. Inspect for erosion or corrosion. (6.1)
	Improper fan mounting	Make sure foundation and bearing bolts are tight.
	Loose screws on wheel	Check alignment, tighten screws.
	Bent fan shaft	Check shaft with a dial indicator. Replace if bent.
	V-belt drive misaligned	Make sure that the fan and motor shafts are parallel.
	Incorrectly sized fan	Fan will operate in the stall region if it is oversized. If fan is undersized, cleaning filters and opening dampers can lower the system resistance.
Noise	Foreign material in fan housing	Clean the fan housing of foreign material.
	Noisy belts (squealing)	Check belt tension and alignment. Replace if worn.
	Worn bearings (howling, screeching, clicking)	Change bearings. (6.3)
	Misaligned bearing seal (high pitch squeal)	Realign bearing face.
Overheated Bearings	Worn bearings	Replace bearings. Inspect shaft. (6.3)
	Improper grease	Use only a lithium base, high-speed channeling type grease.
	Overgreasing	Run fan for a few hours, then clean off excess grease.

(continued)

TABLE 6-1
Summary of Fan Problems and Remedies
(Continued)

Problem	Causes	Remedy (Section)
	Bearing exposed to heat from an oven or dryer after shutdown	Allow fan to run 15 minutes after oven or dryer heat is shut off to prevent overheating.
	Loose belts causing friction heating thickness	Tighten belts. You should be able to depress the belt about one belt.
	Excessive V-belt tension	Adjust tension.
Poor Performance	Incorrect fan rotation direction	Reverse motor leads.
	Tight turns in duct at inlet and outlet	Install turning vanes or elbow splitters in the duct.
	Improperly installed inlet volume control rotation.	Make sure that the IVC prespin is in the same direction as the fan
	Off-center wheel	Make sure fan is centered in air inlet.
	Low fan horsepower	Check air prespin, fan drive sheaves, and sources of air resistance such as a closed damper.
	High fan horsepower	Fan may be designed for hotter, less dense air. Fan may be running backwards.

CHAPTER 6. SELF-STUDY QUESTIONS

Q6-1 Vibration in a fan may be due to a poorly aligned coupling.

( ) True ( ) False

Q6-2 Overheated bearings may be caused by overgreasing.

( ) True ( ) False

Q6-3 Poor air flow in a fan system may be caused by incorrect fan rotation direction.

( ) True ( ) False

APPENDIX A

PREVENTIVE MAINTENANCE SERVICE AND INSPECTION

FANS

Check Points	D	W	M	Q	A
1. Inspect for proper operation, balance, and cleanliness. Check fan wheels for alignment and freedom of rotation. Check fan blades for cleanliness and for damage. Adjust and clean when necessary.			X		
2. Tighten bolts and nuts around duct connections, fan housings, and floor or equipment connections.				X	
3. Lubricate fan bearings.				X	
4. Disassemble fan if necessary. Clean, paint, and inspect components. Repair or replace damaged components.					X

DUCTS

Check Points	D	W	M	Q	A
1. Inspect ducts for deformation, for leakage losses, and for presence of foreign matter in the interior. Repair and clean when necessary.				X	
2. Clean and paint ducts.					X

HOODS

Check Points	D	W	M	Q	A
1. Inspect hoods for damage, for poor connections to exhaust ducts, and for presence of foreign matter. Repair and clean when necessary.				X	
2. Check relative airflow through hoods.				X	
3. Clean and paint hoods.					X

FILTERS

Check Points	D	W	M	Q
1. Inspect filters for cleanliness. Clean or replace when necessary.			X	

V-BELTS

Check Points	D	W	M	Q	A
1. Inspect V-belts for breaks, for evidence of wear, and for proper tension. Replace damaged belts.			X		
2. Check pulleys for proper alignment and for cleanliness. Adjust and clean when necessary.			X		

ROOF MONITORS AND ATTIC LOUVERS

Check Points	D	W	M	Q	A
1. Inspect roof monitors and attic louvers for damage. Repair or replace loose or defective boards.				X	
2. Paint roof monitors and attic louvers when necessary.					X

APPENDIX B

ANSWERS TO SELF-STUDY QUESTIONS

Ql-1. (a)	Ql-9. (d)
Q1-2. (c)	Ql-10. (a)
Q1-3. (b)	Q1-11. (a)
Q1-4. (a)	Ql-12. (a)
Ql-5. (b)	Q1-13. (d)
Q1-6. (d)	Q1-14. (a)
Q1-7. (a)	Ql-15. (c)
Ql-8. (d)	Ql-16. (c)

Q2-1. Static, velocity and total pressures.

Q2-2. Average velocity = (300 * 4 + 450 * 4 + 600)/9 = 400 fpm
Flow rate = 18 * 18 * 400/144 = 900 cfm

Q2-3. (b) Area = 18 * 20/144 = 2.5 ft^2

Q2-4. (b) Flow rate = 2.5 ft^2 * 400 fpm = 1,000 cfm

Q2-5. (b) Flow rate = area * velocity = A * V
= 4 ft^2 * (4005 * $\sqrt{2-1.5}$) = 11,328

Q2-6. (d) See figure 2-19

Q2-7. (c) See figure 2-19,(1.0-0.71) * 20/2 = 2.90

Q2-8. (a) cfm = Velocity * $\pi d^2/4$ <-- (area of circle)
cfm = 600 fpm * 3.1416 * (20 * 20/144)/4 = 1,309 cfm

Q2-9. By Fan Laws

Condition	1	2
cfm	10,000	11,000
rpm	600	660
sp	1.1	1.33
hp	6.0	7.99

Q2-10. rpm = 600 rpm * 4.0/4.5 = 533 rpm (assuming same power to the shaft)

Q2-11. A = fan curve
B = system curve

Q2-12. Point C, point of operation

Q3-l. False	Q3-4. True
Q3-2. False	Q3-5. True
Q3-3. False	Q3-6. True

Q4-1. (a) control airborne concentrations, dust, and vapor to keep below explosive or flammable levels

(b) control toxic particulates, gases, and vapors to keep below exposure levels (TLVs) harmful to workers.

Q4-2. From Table 4-1, 50-100 fpm, use 100 fpm as recommended capture velocity.

Q4-3. cfm = 3 ft * 2 ft * 100 fpm = 600 cfm

Q4-4. Hood, duct, air cleaner, fan and stack.

Q4-5. (b) 0.05 grains/thousand ft^3

Q4-6. (a) Capture, Face
(b) 16, 20

Q5-1. (c) (50 ft * 100 ft * 8 ft) * 60 changes/hr/60 = 40,000 cfm

Q5-2. (a) 4 ft * 8 ft * 100 fpm = 3,200 cfm (maximum)

Q5-3. (c) Area of circle = $\pi d^2/4$ =
= (3.1416 * 3 ft * 3 ft)/4 = 7.1 ft^2

Q5-4. (a) Exhaust area in attic is not large enough
(b) Inlet opening area into building is not large

Q6-1. True
Q6-2. True
Q6-3. True

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