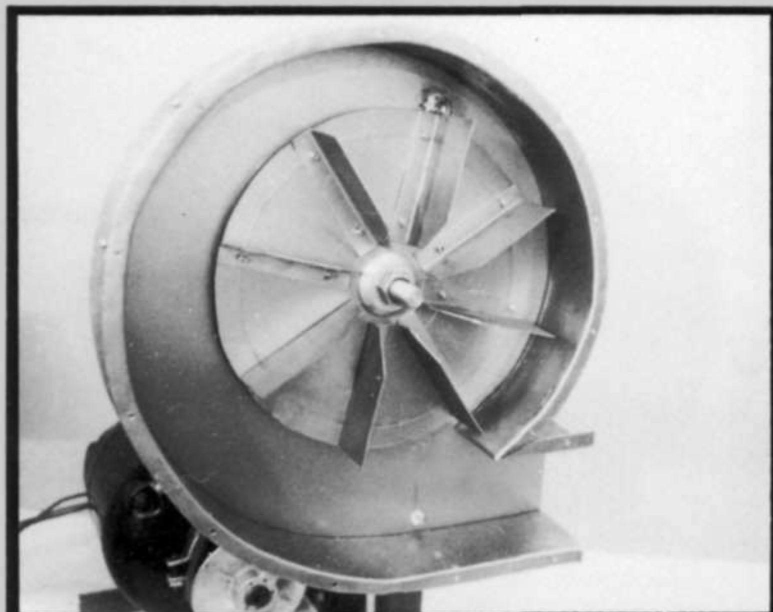
how to design & build
CENTRIFUGAL FANS
for the home shop



David J. Gingery

Lindsay Publications Inc.

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How to Design and Build

Centrifugal Fans for the Home Shop

by David J. Gingery

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WARNING

Remember that the materials and methods described here are from another era. Workers were less safety conscious then, and some methods may be downright dangerous. Be careful! Use good solid judgement in your work, and think ahead. Lindsay Publications Inc. has not tested these methods and materials and does not endorse them. Our job is merely to pass along to you information from another era. Safety is your responsibility.

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Centrifugal Fans Can be Dangerous!

Designers, builders, and users of centrifugal fans expose themselves to special dangers. The rapidly spinning fan disc with its metal blades is as dangerous as the blades of a kitchen blender. Fingers can mutilated or amputated as a result of accident or absent-mindedness. Provide guards to protect you and your children and pets from injury.

Centrifugal fans with blades which are inadequately fastened may fly apart as a result of centrifugal force. The result is a violent explosion that can cause property damage, injury, and possibly even death. Provide a sturdy housing around the fan disc to contain such an explosion. Proper construction and adequate testing is important.

These are two obvious dangers. There are probably others. They have been pointed out wherever known. Properly built and used, homebuilt fans can be every bit as safe as the blower in your home furnace, or the blower in your vacuum cleaner. Be very careful.

INTRODUCTION

Nearly everyone is familiar with "squirrel cage" fans found in home heating, air conditioning and ventilating systems. The squirrel cage is just one of many forms of centrifugal fans and blowers designed in a wide range of sizes for many different applications.

Smaller fans appear on the market from time to time as surplus or salvage, some of which may find practical use in home shop applications. It certainly makes little sense to construct a fan for a special purpose if one is readily available at low cost. But some applications require air at high pressure and volume, and the fans and blowers usually available as surplus will probably not meet the requirements. At this point it becomes practical to design and build a fan for the job at hand.

It is not difficult to design a fan for a specific purpose when the importance of the effect of various factors is understood. Although state-of-the-art fan design has become highly technical, mainly due to advances in jet propulsion and turbine technology, you can easily design a fan to suit your specific needs.

The basic principles that are discussed in this manual have remained unchanged for more than 150 years. Surprisingly, common materials and ordinary tools are adequate for construction of fans capable of delivering air at high volume and moderately high pressure. Because there are many design parameters that can be manipulated, fans can be designed to meet nearly any requirement of volume, velocity and pressure.

The most common application for a high pressure and/or high volume fan is in forcing air into a furnace or in exhausting dust and fumes. A blacksmith's forge or a melting furnace are examples of forced draft.

The exhaustion of welding fumes or the collection of dust from woodworking machines or grinders can be easily accomplished in the small shop at low cost. These are areas of safety, protection and convenience that are usually neglected because a new fan capable of doing this work is too expensive. Because such a fan is seldom found used or surplus, it is worthwhile to consider building one.

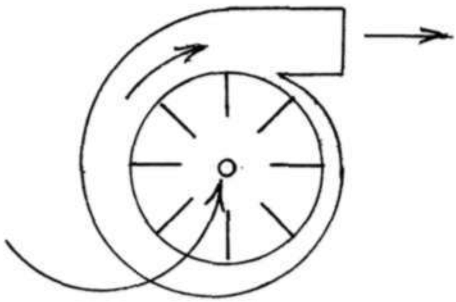
This manual will show you how to determine the diameter and width of a fan wheel and at what speed to run it to achieve the desired pressure and volume. Several methods for building the wheel are discussed. The layout procedure for the scroll housing has been simplified.

From the formula given for calculating the amount of power required to run a fan you will learn (perhaps to your amazement) how much work can be done by a common fractional horsepower motor. You will be shown how to build simple balancing equipment and how to use it. You will also learn how to test the performance of experimental fans with an easily built manometer and pitot tube. Only a few easily understood and applied formulae are needed to guide you. You will enjoy building several projects that will upgrade your shop and make your work environment safer and more comfortable.

As in any shop activity there are dangers that may not be readily apparent. You should be aware that centrifugal force and generated air pressure puts great strain upon the fan wheel and other parts of the blower. Foreign material in the air stream or loose pieces breaking off the wheel can reach velocities of up to several thousand feet per minute. Very serious injury or possibly even death can result if persons are struck. Take all practical steps to protect yourself and others. Electrical wiring must be properly installed to avoid shock and fire hazards. Always be very safety conscious in the shop, and provide protection promptly whenever danger appears.

CHAPTER I

Fan Fundamentals



CENTRIFUGAL FAN PRINCIPLES

Although there are many different designs, all centrifugal fans work in the same way. Air trapped between the vanes of the wheel or rotor is thrown outward by centrifugal force, and replacement air is drawn in at the center. As the speed of the rotor increases, so does the velocity, pressure and volume of the air delivered at the fan outlet. The housing collects or gathers the air as it is expelled from the rotor and directs it out in a single stream.

The ability to develop pressure makes the centrifugal fan the logical choice in a ducted air system. Although the disc or propeller fan can move very large amounts of air, it would have to be complex and sophisticated to do the same work done by a simple centrifugal fan when resistance to flow is a significant factor. Some commercial fans are built to deliver air at pressures that are so high that they should really be called compressors. Small shop applications will not need more than four to six inches of water column pressure or less than 3-1/2 ounces per square inch gauge.

The term "fan" or "blower" will be used interchangeably in this discussion. There are devices and mechanisms called blowers that are not fans, but any fan can be called a blower. Although actual applications may primarily use the inlet air stream rather than the discharge stream, the working principles remain the same.

PREDICTABLE BEHAVIOR OF CENTRIFUGAL FANS

Since air has mass and substance, a measurable amount of work is done by a fan. Luckily, the physical laws that govern its operation are not complicated or difficult to master.

Volume, pressure and velocity of air movement are the prime considerations in fan design. If a few physical and mechanical principles are understood, these measurable variables and the amount of power required to produce them can be predicted with reasonable accuracy before construction by calculations with simple formulas.

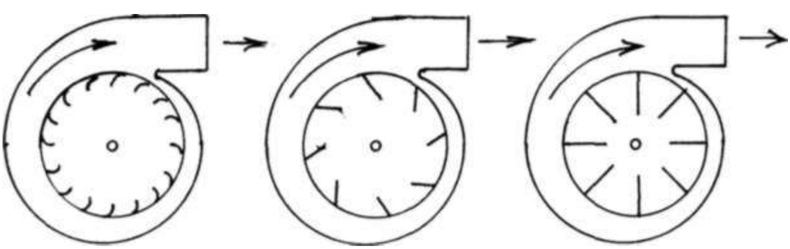
Although state-of-the-art fan engineering is very sophisticated, it is still not possible to design a perfect fan by mere mathematical calculations alone. After the calculations have determined the approximate size and proportions, a model must be built and tested. It is quite reasonable to expect the first model to meet the requirements if ample allowances are made in design. Changing the wheel speed or adjusting the inlet or outlet will bring performance to the desired level.

Although very complex formulas have been developed to take into account almost every conceivable factor, we will discuss only the most basic formulas in this manual. We will use simple math to explore the relationships between obvious, easily measured factors to determine unknown or less obvious design factors. The ignored parameters will certainly affect performance. But rather than apply a more complex formula, we will adjust an over-designed fan to compensate. In fact in many instances, this might even be the more practical way to design since the nature and composition of air can be quite different from day to day. Some fans will need to be adjusted only the first time used while others may require adjustment at each use. If your particular work requires greater precision than what you find here, you will find additional data and formulas in any of several modern engineering manuals found in most public libraries.

BASIC FAN TYPES AND APPLICATIONS

The diameter, width and speed of the wheel, or rotor, are certainly the most important elements of design. A discussion of the various fan types will help us understand the effects of these factors and how the formulas are applied.

There are three common types of fan rotors in use: forward curved vane, backward inclined vane and radial vane. Each type is shown in diagrammatic form below.



FORWARD CURVE BACKWARD INCLINED RADIAL VANE

The forward curved vane type is the familiar quiet "squirrel cage fan" found in low pressure applications. It is made up of many vanes of small area that are curved in the direction of rotation. The wheel will often be about as wide as its diameter, and the inlet opening will often be nearly as large as the wheel diameter. Although it can generate pressures up to 1-1/2" water column, it is most often operated below 1" water column.

Compared to squirrel cage fans, backward inclined vane fans have fewer vanes of greater area that are inclined away from the direction of rotation. This produces a non-overloading characteristic which makes it ideal for commercial heating, air conditioning and ventilating applications where load conditions in complex duct systems vary greatly with time. These fans must run at higher speeds to do the same work as the forward curved types resulting in noisier operation. Although capable of higher pressures, backward inclined vane fans are not usually operated above 3" water column of pressure.

The radial vane type is characterized by straight vanes of greater depth with the wheel usually being narrow in proportion to its diameter. It is capable of pressures well above 12" water column and very high velocity. Radial vane fans are usually of much heavier and stronger construction than ordinary ventilating fans, and are much noisier. This type is most often needed in shop applications. This is the basic design type discussed in this manual since it is ideal for exhausting, for conveying dust and dirt, and for forced draft applications.

The number of blades, or vanes, in a wheel has an effect on performance and efficiency. Low pressure, high volume fans will have many blades while a high pressure radial vane fan will have fewer. In general, increasing the number of blades will reduce slippage and thus increase efficiency until a point is reached where losses due to friction and reduced inlet area will offset gains. A commercially built radial vane fan might have from 5 to 12 blades while a backward inclined wheel might have from 10 to 20 blades. A curved vane squirrel cage fan might have from 30 to 60 or more vanes. Since multiples of four are easy to layout with available home shop methods, I have found that 8 blades works well for most high pressure fans, while 16 blades do nicely for higher volume needs when pressure is not the main requirement.

A number of housing types are in common use, but the familar scroll shaped housing is likely to be the most practical for home construction.

CHAPTER II

DESIGN CONSIDERATIONS

THE NATURE OF AIR

Although invisible, air has mass and viscosity, and like most substances its composition varies. It is composed of approximately 21% oxygen and 79% other gases, mostly nitrogen, and is more or less polluted wherever found. At 68 degrees Fahrenheit, 50% relative humidity and with a barometer reading of 29.92" of mercury, a cubic foot of air weighs very nearly .075 pounds. This is regarded as "standard air" in fan engineering. The air in your shop might be standard at this moment, but it will rapidly change to something other than standard. Most of the air you work with will be other than "standard," but the difference will not be very important in most instances.

Air is more dense (heavier per cubic foot) when it is cold. Warm air rises while cold air falls. It is upon this principle that a chimney works and that violent wind storms develop in the atmosphere. A cubic foot of cold air will contain more oxygen than the same volume of warm air, and it will require more power to move the colder air because it is more dense.

There are applications where the air is delivered for the sake of using its content, usually the oxygen, making the air's weight, temperature, humidity, and other attributes important factors in delivery. In some modern melting furnaces the volume of air in the blast is adjusted according to its density by specially designed controls to ensure that the required amount of oxygen is delivered. In such large volume applications even a small percentage of error is significant. We are not usually burdened with such concerns in the small shop.

When air becomes severely polluted, as in a shop where welding is done, it becomes desirable to exhaust the polluted air and replace it with clean air. Since exhausters must only move air in sufficient quantity to replace the bad air, the composition of the air is not a significant design factor. Rather we need only be concerned with providing sufficient volume of air and enough pressure to overcome resistance within the duct system. In cold weather replacement air must be heated or the work area may soon be too cold for comfort. The source of the replacement air is a factor too, and you will soon hear from your neighbors if you impose your smoke- and fume-laden air upon them.

When dust or other materials are to be conveyed in an air stream, it is the velocity or speed of the air that does the work. Obviously, a specific volume of air is required to maintain velocity within a pipe or duct of a given size. Dust collectors and conveyors require stronger fans and much greater power to achieve high velocity, volume and pressure.

With these fundamentals in mind we can consider the effects of various dimensions and proportions in the design of centrifugal fans. After doing so, some of the formulas used in their design will be easily understood and applied.

THE EFFECT OF WHEEL DIAMETER AND SPEED

Because velocity and pressure are the most easily determined factors in fan behavior, we rely heavily upon them in designing and testing. The diameter of the wheel, or rotor, and its speed, affects both factors.

Since the air is expelled at the periphery or outer edge of the wheel, it will have attained the velocity, or speed, of that surface as it is thrown off by centrifugal force. To determine air velocity it is only necessary to know the speed of the wheel's outer edge. The velocity of air increases directly with an increase of wheel diameter. Doubling the wheel diameter will double the velocity of air delivered if the wheel speed remains the same.

The circumference, perimeter or periphery of the wheel is found by multiplying its diameter by 3.1416. For example, a 10" diameter wheel will have a circumference of 31.416" (10" x 3.1416 = 31.416") A point on its perimeter will travel 2.618 feet (31.416" divided by 12" per foot = 2.618 feet) in each revolution of the wheel.

If the wheel is driven directly by a motor that runs at 1725 RPM (revolutions per minute) then its peripheral velocity will be 4516.05 FPM (feet per minute) because 2.618 feet per revolution x 1725 revolutions per minute equals 4516.05 feet per minute. Thus, the theoretical velocity of the air thrown off will be 4516.05 feet per minute. In practice air velocity will diminish as it travels away from the wheel.

Total output is affected by not only the distance traveled but by friction, slippage and other less obvious factors. These can all be taken into account with a "co-efficient of discharge factor," which is usually .8 in ordinary work. In other words, you'll get only about 80% of the output predicted by the formulas.

THE EFFECT OF AIR VELOCITY

When air is moved by force, it will displace, or push other air out of its way. Moving air exerts pressure in the direction of its travel called dynamic pressure.

If resistance to air flow is encountered, pressure builds up until the maximum capacity of the fan is reached. Such pressure, called static pressure, is exerted equally in all directions. If the entire output of the fan is confined to a tank, chamber, or other pressure vessel, the total pressure measured will be static pressure. Since the air is not moving, there is no dynamic pressure.

If a part of the air is allowed to escape from the chamber, the static pressure will drop. Since air is now moving again, the system now exhibits dynamic pressure. You might think that the dynamic pressure will increase by exactly the same amount that the static pressure decreased. Not so. In practice the moving air must fight its way through the static pressure before it can escape into the rest of the system. The dynamic pressure of the moving air will be reduced by static pressure that resists its flow. The remaining dynamic pressure is called the velocity pressure. This is the practical, working pressure that delivers the stream of moving air. The diagrams below illustrate the effects of dynamic, static and velocity pressure in a fan system.

The relationship between the velocity of the air flow and its velocity pressure is accurately predicted by the formula:

$$V = 66\sqrt{h}$$

where: V = velocity of flow in feet per second
h = velocity pressure in inches W.C. (water column)
air is assumed to be dry & 60° F.

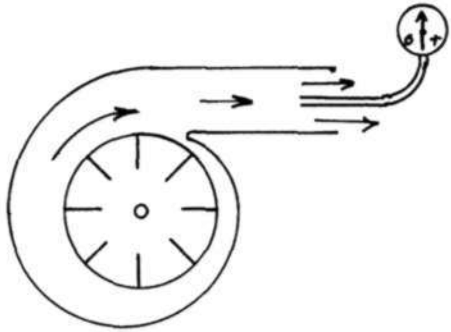
The formula may be transposed to find any factor when the other two are known:

$$V = 66\sqrt{h} \qquad \frac{V}{\sqrt{h}} = 66 \qquad \frac{V}{66} = \sqrt{h}$$

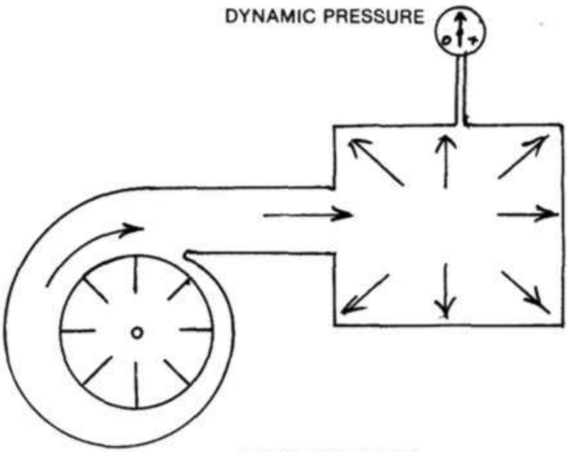
The formula in the last instance states that the velocity in feet per second divided by 66 is equal to the square root of the velocity pressure in inches of water column. We can apply this formula to our 10" wheel at 1725 rpm to determine what pressure it will produce.

$$h = \frac{V}{66} \text{ squared}$$

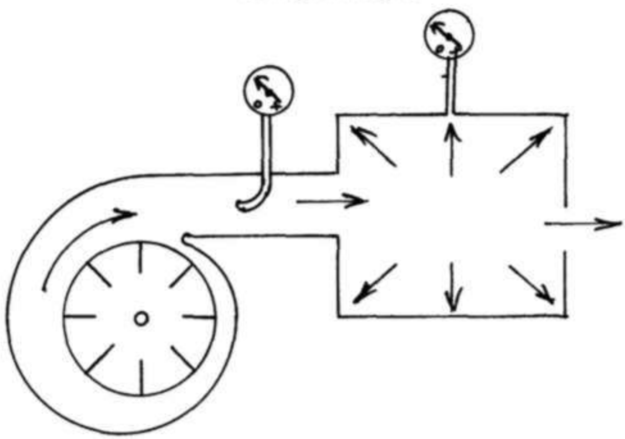
$$h = \frac{V}{66} \times \frac{V}{66}$$



DYNAMIC PRESSURE



STATIC PRESSURE



VELOCITY PRESSURE

We have already calculated that the theoretical velocity of the 10" wheel at 1725 rpm is 4516.05 feet per minute. The decimal fraction is insignificant. The value can be rounded to 4516 feet per minute.

Since the formula calls for velocity in feet per second, 4516 feet per min divided by 60 sec per min equals 75.27 feet per second. Dividing this by 66 gives 1.14. If we square this, we get

$$\text{velocity (ft/min)} = \frac{\text{velocity (ft/sec)}}{\text{60 seconds/minute}}$$

$$= \frac{\text{4516 ft/sec}}{\text{60 sec/min}}$$

$$= \text{75.27 ft/min}$$

$$h = \frac{\text{velocity}}{66} \times \frac{\text{velocity}}{66}$$

$$= \frac{75.27}{66} \times \frac{75.27}{66}$$

$$= 1.14 \times 1.14$$

$$= \text{1.3 inches water column pressure}$$

This can be verified by using the other two variations of the formula to see if the numbers calculate out correctly:

$$\frac{75.27}{1.14} = 66 \qquad \text{and} \qquad 66 \times 1.14 = 75.24$$

In another instance a fan capable of delivering air at a velocity pressure of 4" W.C. might be required. The formula is applied to calculate the velocity of flow from the pressure.

$$V = \sqrt{h}$$

$$V = 66\sqrt{4}$$

$$\text{since } \sqrt{4} = 2 \quad V = 66 \times 2 = 132 \text{ FPS.}$$

Since motor speeds are stated in revolutions per minute (RPM) and velocity must be in feet per minute, we must multiply by 60 seconds per minute: 132 FPS x 60 sec per min = 7920 FPM. If a 1725 RPM motor were to drive a fan directly, a peripheral velocity of 4.59 feet per revolution of the wheel would be required to deliver at 4" W.C. pressure:

$$\frac{7920 \text{ feet per minute}}{1725 \text{ rev per minute}} = 4.59 \text{ feet per revolution}$$

Obviously, the 4.59 feet is the distance around the wheel, or the circumference.

$$\frac{4.59 \text{ feet of circumference}}{3.1416 \text{ (pi)}} = 1.46 \text{ feet diameter}$$

$$1.46 \text{ feet} \times 12 \text{ inches per foot} = 17.5\text{'' diameter}$$

From this we know that a 17-1/2" wheel running at 1725 rpm should deliver air at 4" W.C. pressure. Of course, a smaller wheel running at a higher speed gives the same result and so does a larger wheel running at a lower speed. These combinations can be investigated by running numbers through the formula.

THE EFFECT OF WHEEL WIDTH

Fans are rated according to their ability to move a quantity of air against resistance. Ratings are usually given in cubic feet per minute against a value of static pressure in inches of water column. The theoretical capacity of a fan is the product of the velocity of flow in feet per minute multiplied by the area of the discharge times the coefficient of discharge, which has been given as .8.

The theoretical discharge opening is called the "blast area," and is calculated by multiplying the diameter of the wheel by its width and dividing the product by a factor that varies from 2.5 to 3 in practice. Since the larger value allows the greatest margin for error and inefficiency it is the safest to use in casual design. The formula to find the blast area is

$$A = (D \times W)/3$$

where: A = blast area in square inches
D = wheel diameter in inches
W = wheel width in inches

For example, the blast area of a 10" diameter wheel that is 3" wide would be 10 square inches.

$$\begin{aligned} A &= (10" \times 3")/3 \\ &= 30/3 \\ &= 10 \text{ square inches} \end{aligned}$$

Since the capacity is stated in cubic feet per minute,

$$\frac{10 \text{ square inches}}{144 \text{ sq in per sq ft}} = .069 \text{ square feet}$$

A 10" wheel at 1725 rpm delivers at a velocity of 4516 feet per minute. Its theoretic capacity is

$$\begin{aligned} &4516 \text{ ft per min} \times 0.069 \text{ sq ft} \times 0.8 \\ &= 249.28 \text{ cu ft per min} \end{aligned}$$

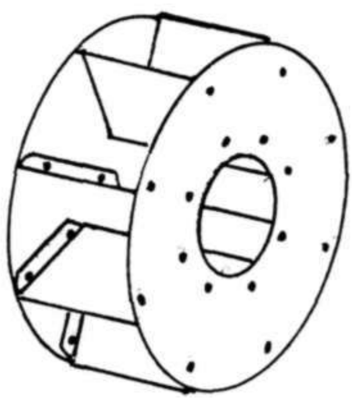
That's almost 250 cu ft per minute against a static pressure of zero. Fan output will decrease as the static pressure increases. Larger, faster and more powerful fans are needed in complex duct and pipe systems where static pressure is high.

THE EFFECT OF WHEEL-WIDTH-TO-DIAMETER RATIO

When high volume of air is the main consideration, the wheel should be made as wide as possible. There is an optimum inlet size for each design type that is found by experimentation. One limitation on wheel width is the amount of air that can freely enter the inlet. In most high volume fans there will be a double inlet allowing the wheel to be very wide in proportion to its diameter. Such fans deliver high volume at low pressure and are not able to overcome the resistance of long ducts and pipes.

When high velocity and pressure are needed, vanes will be deeper, and the wheel will be much narrower in proportion to the diameter. Inlet and outlet openings will be proportionately smaller as compared to low pressure fans.

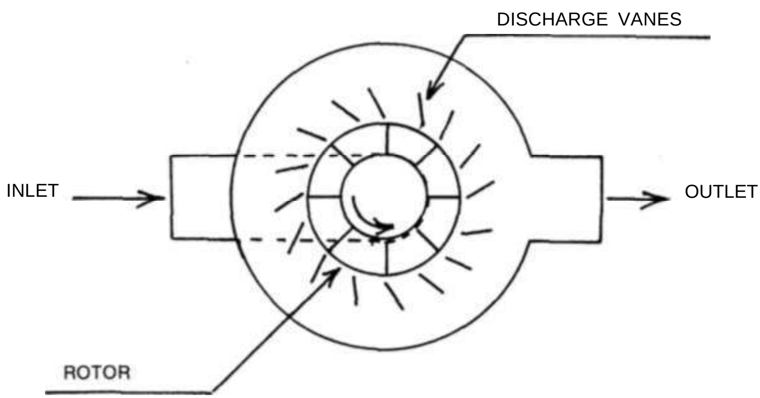
It is not only the width-to-diameter ratio that affects pressure but also the depth of the vanes. It is apparent that a wheel with very deep vanes will trap air more effectively and force it to the perimeter of the wheel more efficiently at higher pressure. It is upon this principle that the "steel pressure fan" design is based. The vanes are not only quite deep but a plate also covers both sides so that a closed channel exists between each pair of vanes as shown in the sketch.



Such rotors are very efficient. The outer plate also serves to make the wheel stronger when relatively light sheet metal is used to construct the rotor. The width of some high pressure wheels might be as little as 5% of the wheel diameter.

SCROLL HOUSING DEVELOPMENT

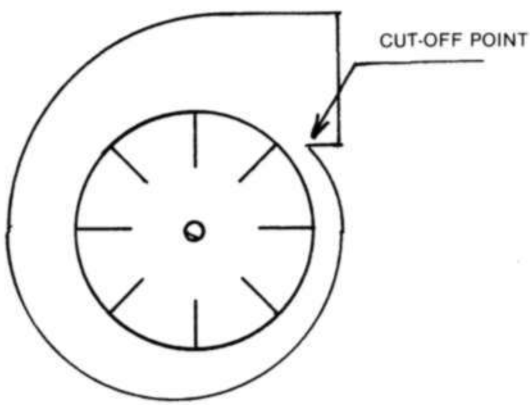
For most applications a scroll housing is the simplest and most effective housing for most fans. Some fans use a series of discharge vanes located around the rotor which direct air through a plenum chamber into one or more outlets in the housing.



Such a housing is more complex but such a design is efficient design and its compactness suits many applications.

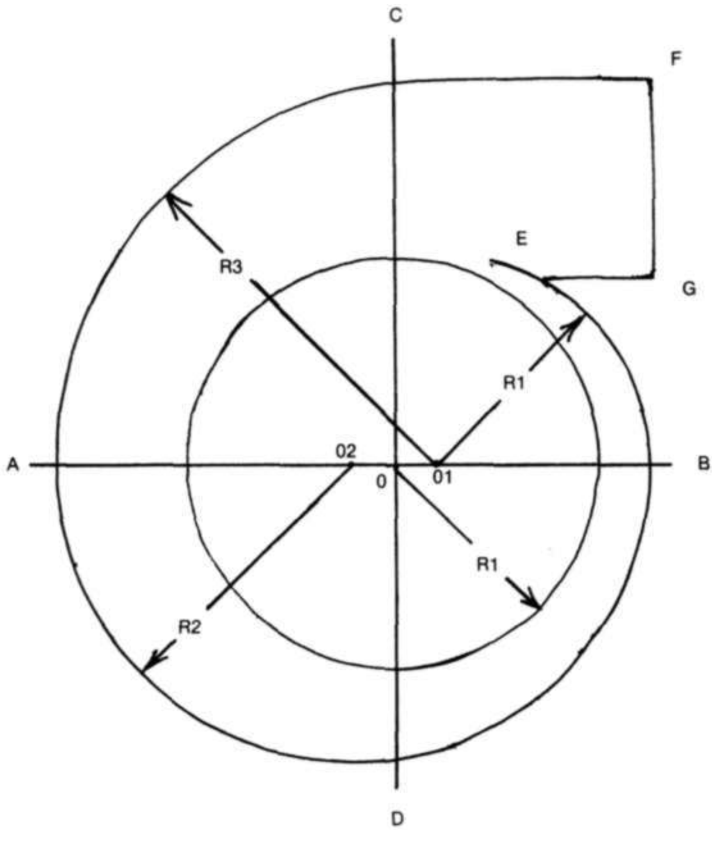
In most instances the simple scroll housing is both efficient and very easy to lay out and to build. It would seem upon first consideration that very close clearances would be necessary in fan construction. Not so. The fact is that very casually built fans with loose clearances work with amazing efficiency. Of course, very high pressure work would demand very close clearance between the wheel inlet and the housing and also at the cut-off point, but in pressures below 12" water column, these dimensions are not at all critical.

The two determining factors in scroll design are the cut-off point clearance and the blast area. Given the diameter and width of the wheel, the blast area can easily be calculated. Since the discharge opening is usually from 1-1/2 times to 2-1/2 times the blast area as a rule-of-thumb, we have the first dimensional clue from the calculation. The cut-off point is that part of the scroll that comes closest to the wheel. The clearance at this point is usually from 5% to 10% of the diameter of the rotor.



As seen in the illustration the scroll is an expanding spiral, or volute, that directs all of the air thrown off the wheel into the discharge opening. Although it would certainly require much study and experimentation to develop a nearly perfect scroll, it is not at all difficult to layout a very good approximation that will serve well in ordinary work. As an illustration and exercise let's lay out a scroll housing for a 10" wheel that is 3" wide.

Since the blast area is calculated at 10 square inches, the discharge opening can be from 15 to 25 square inches. With a clearance of 3/8" on each side of the wheel which is not excessive, the housing can be 3-3/4" wide. Thus, the discharge opening height can be from 4" to 6-1/2" high. A cutoff point clearance of 1/2" will work very well. It can be set even closer but the fan will be noisier. The entire work of layout will be done upon three centers and three radii.



Layout begins with a horizontal line, AB, intersected by a vertical line, CD, which establishes the wheel center at O. Additional centers 01 and 02 are located 1" away on both sides of center O. Additional centers Ol and 02 are located on either side of center O, the separation distance being to 10% of the wheel's diameter.

The next step is to draw the circle in the middle that represents the wheel. Rl represents the radius of the wheel of itself, or 5" for our 10" wheel. With the radius Rl and center O, the circle representing the wheel is drawn. Shifting the compass or dividers to center Ol and using radius Rl again an arc is drawn from point B past point E.

Then, shifting to center 02, radius R2 is used to draw an arc from B through D to A. The length of radius R2 is the length from 02 to B.

Next, extend the arc from A to C using Ol as center. A horizontal line is drawn from C approximately 5", or one radius of the wheel, to point F. A vertical line is drawn from point F to represent the height of the discharge opening. In this instance we choose 4" for the height to give an opening size of 3-3/4" x 4" or 15 square inches.

Finally, a line is drawn from point G to point E to complete the layout.

Notice that R2 is equal to Rl plus the distance from Ol to 02. Likewise R3 is equal to R2 plus the distance from Ol to 02. If the distance from O to Ol and 02 is increased, the distance from point C to the wheel will increase, and the practical cut-off point clearance will be greater. Such increased clearances should be used on low-pressure, high-volume fans.

GENERAL PROPORTIONS FOR FANS AND BLOWERS

The true delivery of a fan being designed is defined as the product of the area of the design's discharge area and its design velocity. This "ideal world" delivery differs from the "real world" value that we actually get in the prototype. Factors in design formulae "adjust" the calculation results so that we can put "ideal" information into these formulae and get out "real" information. All this may sound like a "guess and by-golly" design method, but surprisingly, the calculation results are near enough to measurable real-world values to be very useful.

One possible problem area involves ratios. Fan width-to-diameter and inlet-to-diameter ratios greatly affect fan performance, yet the formulae do not take them into account. Understanding the effect of some of these ratios will help us decide how much confidence we can place in our calculation results.

In a single inlet fan, the wheel is usually no wider than one-half the diameter. As the width of the wheel increases, the volume of air moved increases. If the wheel is too wide, it must pull a large volume of air quickly through a relatively small opening. The velocity of incoming air will be too great, and the wheel will starve for air.

For high-volume/low pressure fans this problem is solved by using an inlet at both sides of the wheel. The inlet must be as large as possible to allow the large volume of air in. To accomplish this, the vanes will be very shallow and numerous. The inlet diameter will be from 65% to 75% of the wheel's diameter, and often larger. The outlet area will be in proportion to the inlet area. In making a fan free breathing, you must sacrifice something. Such a fan just can't deliver volume at high pressure.

High pressure fans will be narrow in proportion to the wheel diameter, and both the inlet and discharge openings will be smaller than in a low-pressure fan. The inlet diameter can be from 35% to 65% of the wheel diameter. The width of the wheel can be from 10% to 25% of the diameter. Again, outlet size in proportion to the inlet.

It should be apparent from a comparison of these two design extremes that deeper, longer vanes will build up more pressure. Deeper vanes mean that the inlet must be smaller. (Long vanes are of no use if much of their length is exposed by a large diameter inlet.) A small inlet means the fan cannot handle high volume which in turn means that the wheel can't be as wide as other designs.

Remember, a wide fan wants to deliver higher volume than a narrower fan. To get volume, the narrow, deep vaned fan must run at higher speed resulting in a high velocity, high pressure delivery which generates more noise and greater losses from friction. You can see each type has its advantages.

A common application for a high pressure fan is a vacuum cleaner. The usual proportions are a width of 12 to 18% of the wheel diameter with an inlet from 40 to 50% of the wheel diameter. The vanes will often be a compound curved variation of the backward inclined form. Not many modern vacuum cleaners use a scroll housing. The air will be discharged through curved vanes. Some units discharge the air from one wheel into the inlet of another to form a very efficient and powerful multi-stage blower.

THEORETICAL CAPACITY OF FANS

With an understanding of the effect of various factors and components it is possible to calculate the theoretical capacity of any fan given the size of the wheel and its speed. Since no fan is 100% efficient there must be an allowance for slippage and for internal losses due to friction. To do this, we multiply theoretical results by a factor of 80%, which is called the coefficient of discharge.

A practical formula for calculating fan capacity is the product of the blast area multiplied by the discharge velocity giving a theoretical capacity, which is then multiplied by the coefficient of discharge.

$$C = B \times V \times D$$

where

C = capacity in cubic feet per minute
B = blast area in square feet $(B = DW/3)$
V = velocity in feet per minute
D = coefficient of discharge (.8)

Given a 10" diameter wheel 3" wide at 1725 RPM, what will be its theoretical capacity?

The blast area is

$$\frac{10'' \times 3''}{3} = 10 \text{ sq inches}$$

$$\frac{10 \text{ square inches}}{144 \text{ sq in per sq ft}} = .0694 \text{ sq ft}$$

For these calculations it is reasonable to approximate the blast area as .07 square feet.

The peripheral velocity of the wheel is the product of the circumference of the wheel multiplied by its speed.

$$10'' \times 3.14.6 \times 1725 \text{ rpm} = 54{,}192.6 \text{ in per min}$$

$$\frac{54{,}192.6 \text{ in per min}}{12 \text{ in per foot}} = 4516.05 \text{ ft per min}$$

A value of 4516 is close enough for these calculations.

4516 ft/min x 0.07 sq ft x 0.8 = 252.896 cu ft/min

Having already found the velocity pressure of a 10" wheel at 1725 RPM to be 1.3" W.C. we can expect our blower to deliver approximately 250 cubic feet per minute at a velocity pressure of 1.3" W.C. against zero static pressure. Output will be reduced as static pressure increases until delivery stops at a static pressure of 1.3" W.C. Resistance to flow can be at either inlet or outlet.

POWER REQUIRED

Since the nature and composition of air can differ so greatly from time to time and from place to place, some very complex formulas have been devised to calculate fan power requirements. All of them are based upon the standard horsepower of 33,000 foot-pounds per minute.

Modifying factors are introduced to take into account temperature, humidity, barometer reading, etc. for most work discussed here the basic expression is,

where

HP = horsepower required
C = volume of air in cubic feet per minute
h = pressure in inches water column
E = efficiency which varies from .40 to .60

If the value of C is stated in cubic feet per second the formula is then,

$$HP = \frac{C \times h \times 5.2}{550 \times E}$$

This is because

$$\frac{33{,}000 \text{ ft-lb per min}}{60 \text{ sec per min}} = 550 \text{ ft-lb per sec}$$

The formula can be further simplified if the efficiency factor is assumed to be .50.

$$550 \times 0.50 = 275$$

We can plug this new value into our formula:

$$HP = \frac{C \times h \times 5.2}{275}$$

Applying the formula to our 10" x 3" wheel at 1725 RPM, which delivers approximately 250 CFM at 1.3" W.C. pressure we have,

$$\frac{250 \text{ cu ft per min}}{60 \text{ sec per min}} = 4.17 \text{ cu ft per sec}$$

$$\frac{4.17 \text{ cu ft/sec} \times 1.3\text{'' WC} \times 5.2}{275} = \frac{28.19}{275}$$

$$= .103 \text{ HP.}$$

Since, .125 equals 1/8, we will need slightly less than 1/8 hp to power this fan.

Now we have a reasonable idea of what to expect from a fan wheel of given size, and how much power it will require to run it at the speed to give desired delivery. We also know how to modify its behavior. It will be a relatively simple matter to build a fan with a variable inlet and outlet size in order to find optimum delivery. A variety of wheels can be used and the wheel center can be moved within the housing to vary the cut-off point clearance.

After discussing some construction methods and materials, we can look at some simple equipment for testing and evaluation that will be a help in achieving an ideal design for specific jobs at hand.

CHAPTER III

CONSTRUCTION METHODS AND MATERIALS

While the various parts of a fan can be made of many commercial materials, only a few can be considered for use in small home shops. Wood, including plywood, composition boards, plastics and sheet metal can all be formed with simple tools and limited skills. Fiberglass with polyester resin, such as used for boat building and auto body repairs might offer some interesting possibilities if cost is not a consideration. If the fan must move only air at normal temperature, plywood, composition board or light sheet metal will be adequate. These materials are very easy to work.

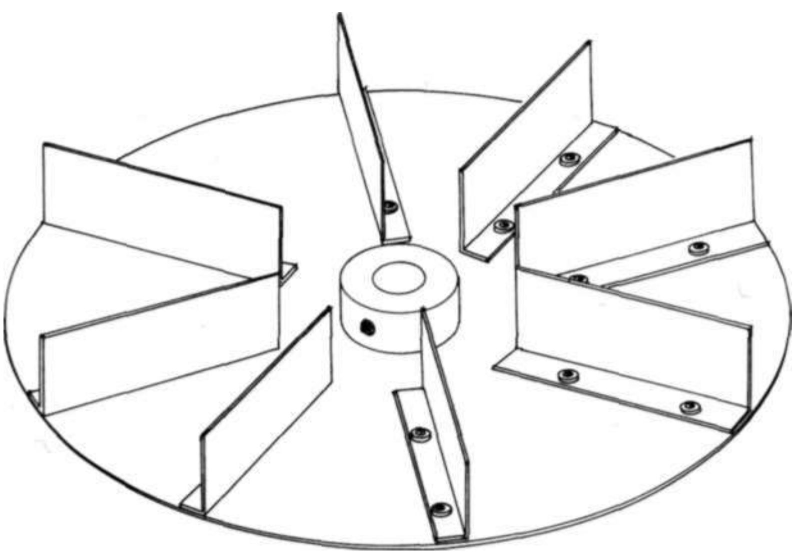
If abrasive materials, corrosive fumes or high temperatures are involved, the fan may have to be built entirely of metal, and some parts may have to have a protective coating of some type. If high velocity and pressure are required, a heavier metal must be used, and construction might entail welding or brazing.

WHEEL CONSTRUCTION

The wheel, or rotor, is the first consideration in any fan. In a commercially built fan it might be a casting in metal, a plastic molding, a die stamping or possibly a welded assembly. It might also be a combination of stamped, cast or molded parts assembled with screws, bolts, rivets or other means. In the small limited shop it is probably most practical to build the wheel of sheet metal and to join the members with bolts, rivets or by welding or brazing.

The simplest wheel to build is the straight radial vane type. All others can be considered a variation of it. Since the dual inlet squirrel cage fan is so easily found in surplus or salvage, it makes little sense to construct such a wheel. In this manual we'll discuss only the single inlet exhauster or pressure fan.

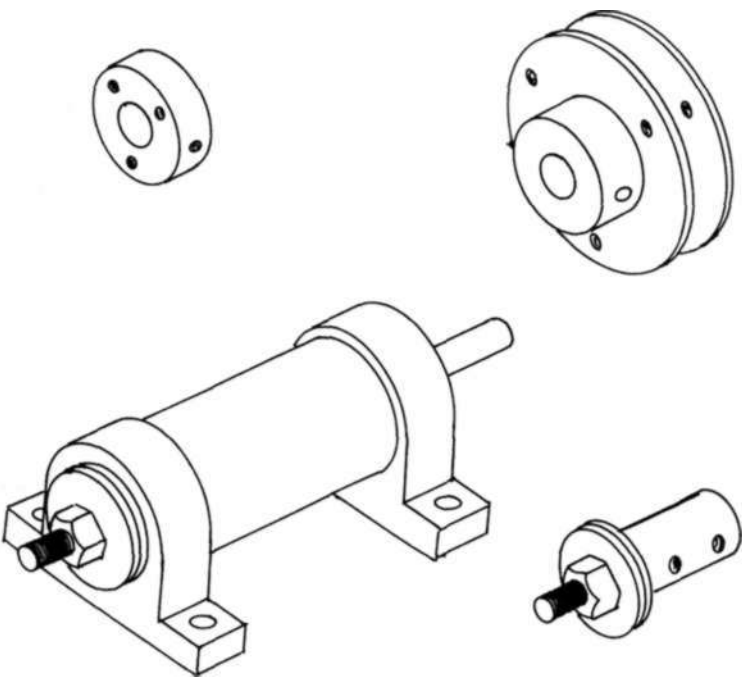
The components of a simple open-sided wheel are a disc of metal for the back plate, a hub and some vanes with a flange for fastening to the back plate with bolts, screws or rivets. My favorite fasteners for light duty wheels are "pop rivets" because they are easily installed from one side with the tool and they will not work loose as bolts or screws may. I use steel rivets in all but the lightest duty wheels since they are much stronger than aluminum. Of course heavy duty wheels should be assembled with solid hammer set rivets or by welding.



AN OPEN SIDED WHEEL ASSEMBLED WITH HOLLOW RIVETS

The hub can be a standard set-screw collar, a small pulley or the hub from a discarded fan wheel found in salvage. If the fan is to be belt-driven rather than mounted directly on the motor shaft a threaded arbor such as those used for saws and grinders can be used for mounting. There are also threaded motor shaft adaptors available in either right or left handed threads.

Universal fan hubs can be had from some professional heating and air-conditioning suppliers. These hubs are fitted with set screws and are bored to fit standard shaft sizes. Some are made with milled keyway. They are drilled and tapped for three or more screws that are used to mount the back plate on the hub. A set screw collar or a discarded pulley can be modified by drilling and tapping the screw holes.



AN 8" OPEN SIDED WHEEL

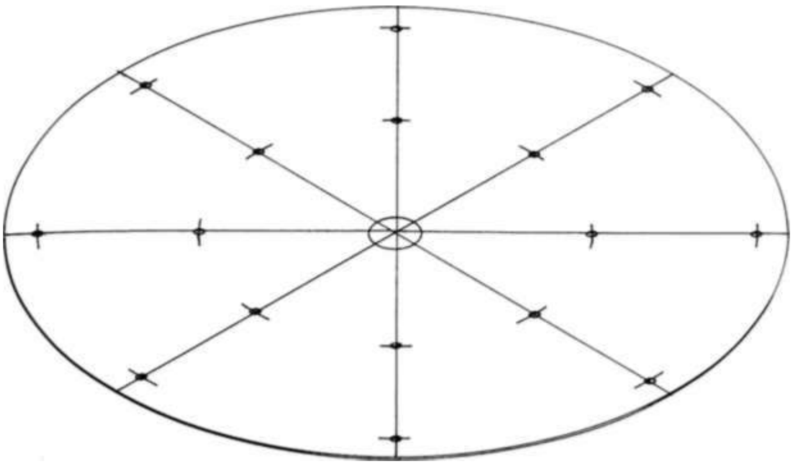
All layout work should be done carefully with a sharp scriber. Attempt to make everything as precise and symmetrical as possible to avoid later balancing problems. If a brake is not available, flanges on the vanes can be bent in a vise or by clamping between two heavy bars of metal. All holes should be carefully center punched before drilling.

To make the wheel backplate draw an 8" circle on sheet metal with a compass or dividers. A horizontal line is then drawn exactly through the center. Next, a vertical line is drawn through the center of the circle at exact right angles to the horizontal line. At this point, the circle is divided into four equal segments.

By drawing two additional diagonal lines through the center we will have divided the circle into eight equal parts. Next, locate the mounting holes by using a compass or dividers to mark each of the eight radial lines 2" from the center and 3-1/2" from the center. Center punch carefully, and drill sixteen 1/8" holes.

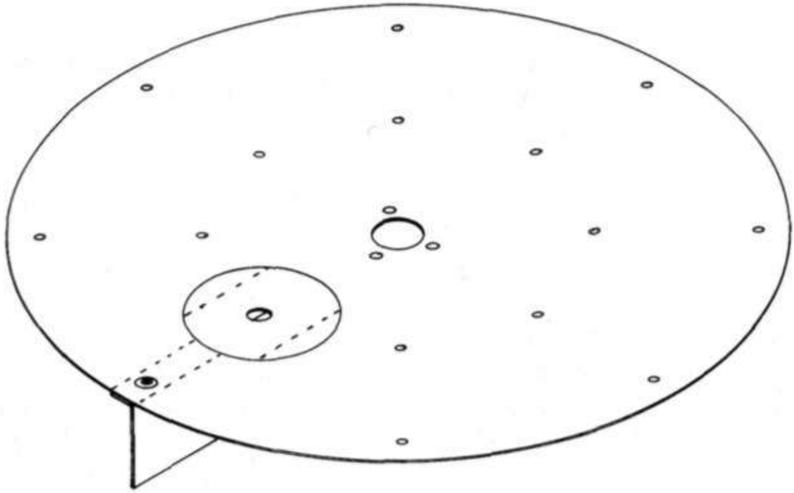
Center punch the center hole carefully and drill to 1/2" or to the size of your arbor. If the center hole is drilled off center, the fan will probably be useless. At the very least, the fan will be difficult to balance. I recommend starting with a small hole and enlarging it progressively with larger bits rather than to attempt to drill it to full size in a single step. A tapered repairman's reamer can be used if you don't have larger bits, or you can carefully scribe the center bore with the dividers and file to the line with a round file.

Now, cut out the backplate. Sheet metal of moderate weight, up to 24 gauge, can be cut with snips. Compound lever snips, or aviation snips, can cut up to 22 gauge iron with considerable effort and fatique. A variable speed electric jig saw with a metal-cutting blade works quite well on heavy sheet metal up to 16 gauge. Heavier plate might best be cut with a torch or a metal-cutting band saw. A stroll through the local scrap metal yard might turn up a disc of nicely stamped heavy sheet metal that will save a lot of labor.

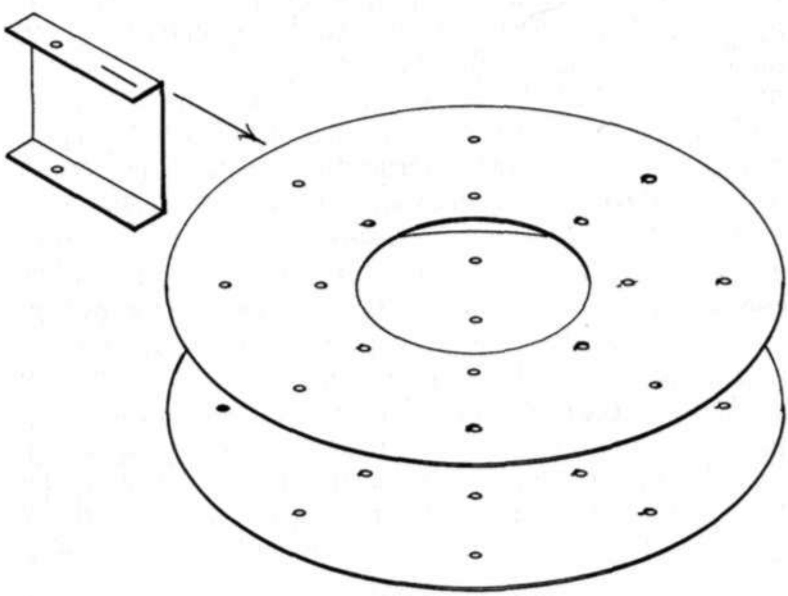


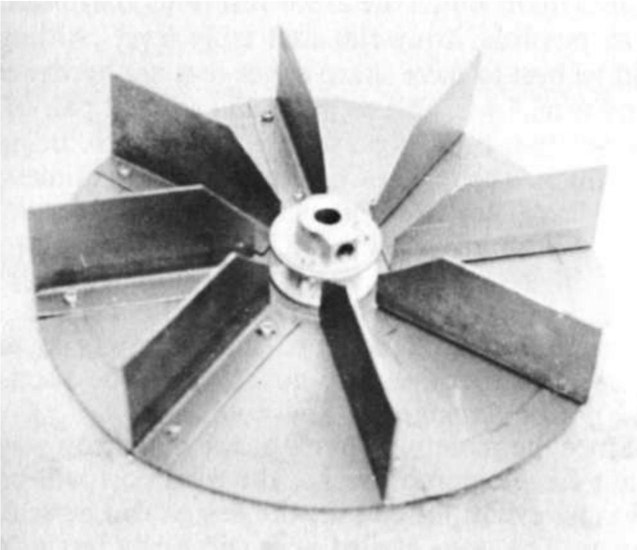
If the backplate is to be mounted on a hub, the hub itself can be used as a drilling template while centered upon the plate with a shaft through both the hub and the plate. You can use drills which can pass through the tapped holes in the hub but do not damage the threads. Use the holes in the hub as a template to drill the holes in the backplate. Next, enlarge these holes so that the screws can pass through the backplate and into the tapped hub holes. In this way, you are assured that mounting holes will line up with the purchased hub. A reinforcing washer can be used at each hole to stiffen the plate if necessary.

Drill only one hole in each vane mounting flange, and mark the center of the second hole with a sharply scribed line on the flange. Fasten the vane to the backplate with one rivet. Next, locate the center line of the second hole through the hole in the plate. Clamp the flange to the plate, and drill the second hole. It is unlikely that both pairs of holes could be precisely drilled in any other way with the ordinary hand tools in a small shop.



If you are building a double-plate, steel pressure wheel, both plates should be drilled together to assure uniformity, symmetry, and ease of assembly.





BALANCING THE WHEEL

Careful layout, cutting, drilling and assembly will result in a wheel with near-perfect balance, satisfactory for operation at moderate speeds. Larger and faster wheels will need to be balanced to reduce vibration at speed.

Equipment for dynamic balancing such as used to spin-balance auto tires is beyond consideration in small shops. Fortunately, it is easy to build a set of "balancing ways" with which to test the static balance of wheels and other simple forms. The object is to create a pair of perfectly smooth and level ways upon which to rest the arbor, or axle, of the wheel. If the wheel is out of balance, it will roll as gravity pulls the heavy area to the lowest point.

To balance, simply add weight to the point opposite the heavy area; that is, to the highest area. Or you can drill holes on the heavy side to remove the undesired weight. When the wheel is in balance it will rest in any position upon the ways without turning.

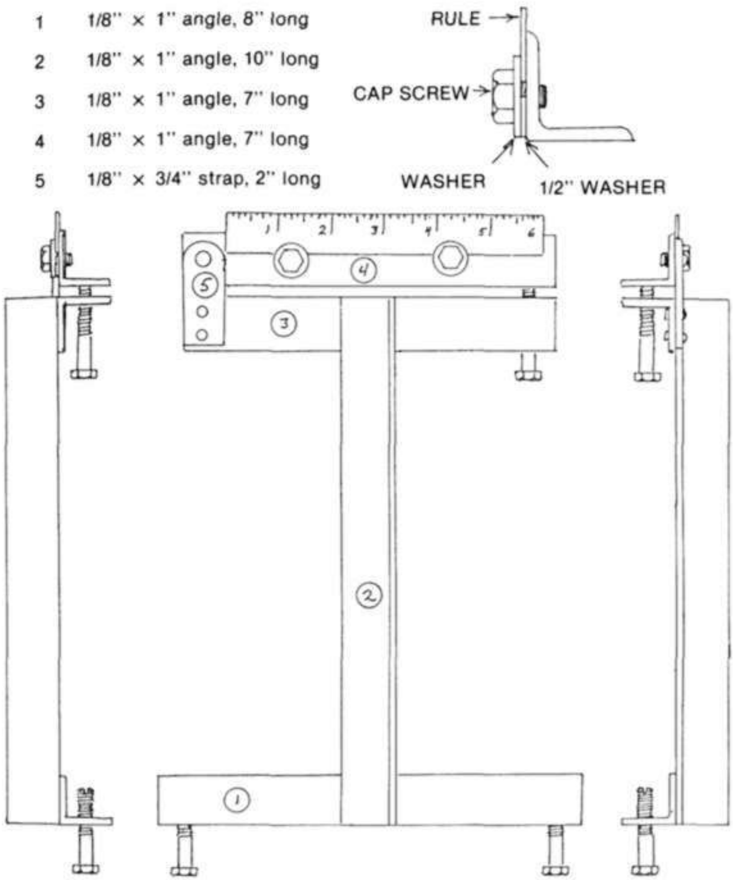
The most important parts of the balancing ways are the edges upon which the arbor will rest. They must be as thin as possible, truly flat and truly level. Although it would be best to have sharp edges that are hardened and ground true, I was able to build an excellent pair of ways by using a pair of 6" rigid stainless steel rules. Being only .046" thick, their edges are nearly as frictionless as a sharp edge. They are quite hard and durable, and the edges are ground very smooth. A few passes with a fine stone will remove any burrs or minute roughness. Although I used very cheap rules rejected because of errors in numbering, I designed the mounting system so that should expensive, top-quality rules be used, they would not be damaged in any way.

Since the remaining parts of the balancing ways can be built from common scrap, the total cost will be very low in any event. Of course, the design can be scaled up or down. The one detailed here will easily test a 24" x 5-1/2" wheel that is mounted on a 3/4" arbor.

Welding is the preferred method of joining the frame parts, but rivets or bolts can be used also. The assembly must be rigid so that it will remain solid and level once adjusted.

A look at the sketch will reveal most of the details. An opposing pair of side frames are made of 1/8" x 1" angle iron. The pivots for the ways supports can be riveted or welded to the frames. The joint should be snugly riveted but free enough so that the weight of the support will operate it without binding as the leveling screws are adjusted. The 6" steel rules are fastened to the ways supports with short 1/4" cap screws and washers with half-washers used to hold the surface of the washers snug against the rule.

After both frames are mounted on an 8" square of 3/4" plywood with 1/4" x 1" flat-head machine screws, the holes for the base leveling bolts can be drilled. Note that a slot is cut in the end of each of the leveling bolts with a hacksaw so that they can be adjusted from the top with a screwdriver.



An accurate level is needed to set up the balancing ways. I use the one built into my machinist's square. Although it is not a precision level, it works very well when used with care.

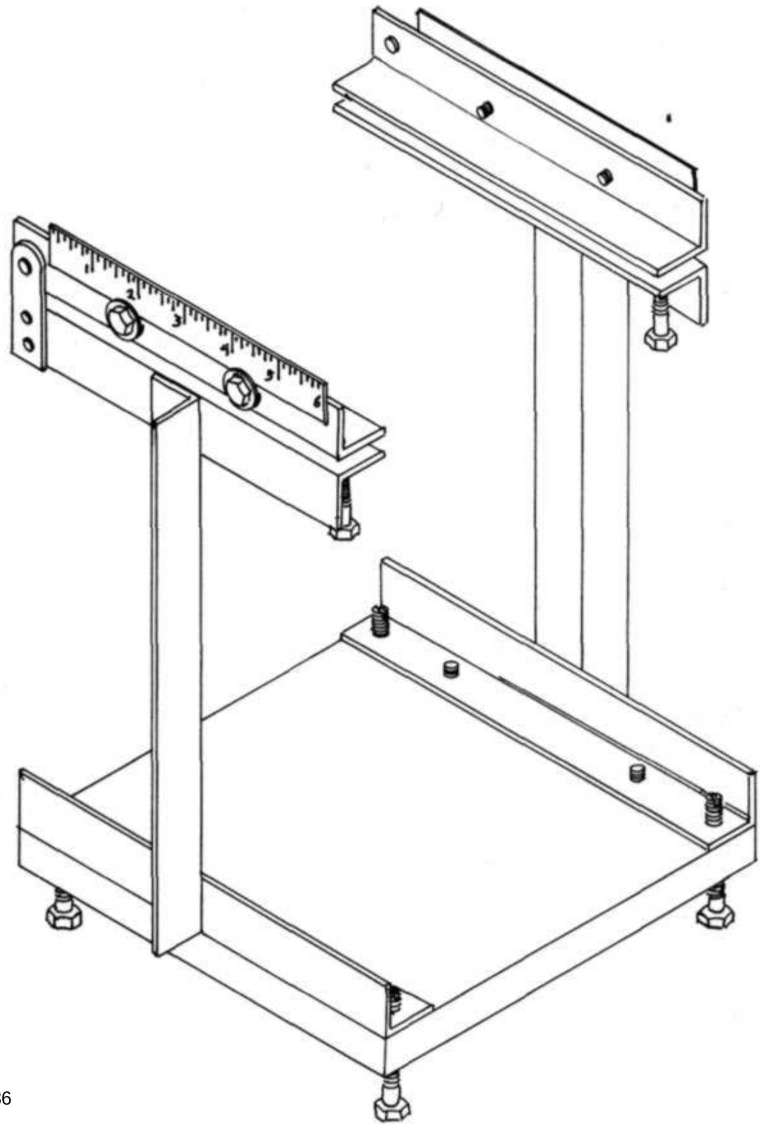
There are significant differences in the quality and accuracy of levels, differences not always perceptible to the naked eye. A magnifying glass is usually helpful in reading a poorer quality level.

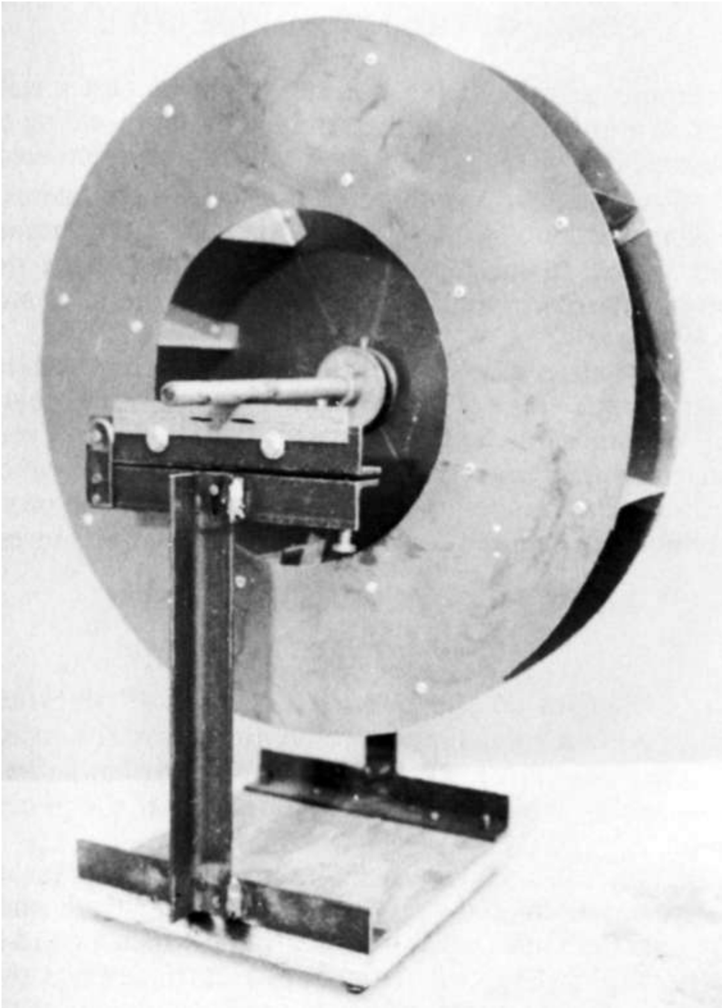
A level can be tested by reversing it upon a level surface. If it reads the same in both directions, it is true.

First, the base of the balancing ways is adjusted so that the unit rests solidly on the bench. If the bench is not rigid, work on the floor. Simply adjust the base legs until the whole assembly appears to rest reasonably level in both directions. There can be absolutely no wobble.

Next, by turning the adjusting screws to raise or lower each support and by testing with your bubble level, adjust each way separately until both are truly level.

Finally, rest a truly straight arbor across the ways, and test it with the bubble level. A final adjustment is made in the base legs to bring the arbor to level.





The best arbor for testing is a length of ground and polished shaft, but clean, smooth cold-rolled steel round will work. Hot-rolled bar stock or cheap shafting that has been plated will not be reliable. Any flat spot or roughness on the shaft or ways will destroy accuracy.

The ultimate accuracy of the balancing ways depends upon their being truly level in all directions, although surprisingly good results can be had even if the ways are not quite true.

FORMING THE HOUSING SCROLL

A commercially built fan housing might be cast metal, molded plastic, sheet metal assembled by spot welding or other means, or sheet metal die-stamped and assembled.

Occasionally, you may find a discarded commercial housing and modify it to accommodate a custom, home-built wheel. In most such cases, you would make the housing narrower and reduce the size of the inlet and outlet.

Most often it will be best to build the entire housing from scratch. Plywood used with a composition board like Masonite, in combination with sheet metal is a very convenient material with which to form a housing when moving room-temperature air. These materials will probably not be adequate for other gases and temperatures.

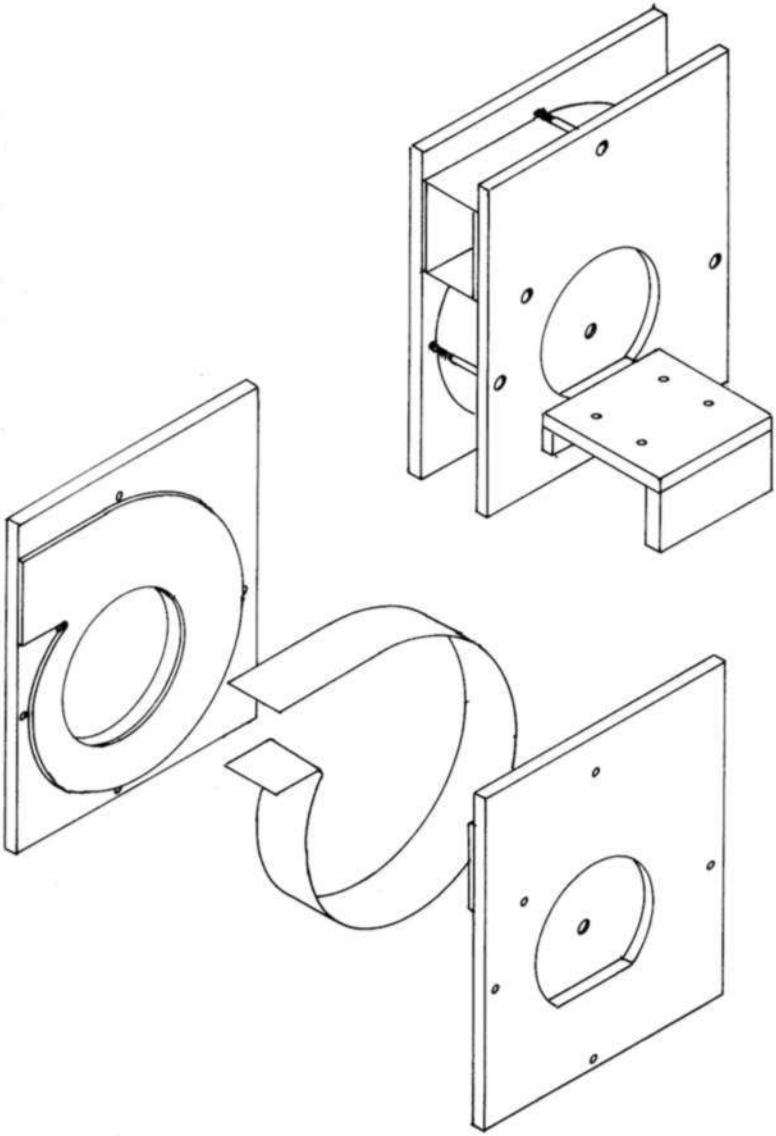
BUILDING A PLYWOOD HOUSING FOR AN EXHAUST FAN

Two rectangles of plywood of 1/2" to 3/4" thickness will provide a solid support for both the scroll housing and the motor drive, whether it be a belt drive or a direct drive with the wheel mounted directly on the motor shaft.

The scroll sides can easily be cut from 1/8" Masonite and mounted on the plywood side supports with small screws. These sides act as a form around which a band of sheet metal is wrapped. Four or five carriage bolts pull the two plywood side supports together clamping the sheet metal scroll between them.

Since most standard motor shafts will probably be too short to reach through the full thickness of the support and the scroll for direct drive of the wheel, one side of the plywood side supports will need a large circle cut into it to allow part of the motor housing to extend somewhat into the fan housing.

Notice in the sketches that the center cut-out for motor clearance is only through the back support. The



scroll form of 1/8" Masonite is drilled in the center only for the motor shaft.

To permit installation of the wheel on the shaft, another large circle will have to be cut on the inlet side through which the wheel can be slipped onto the shaft. An adapter is installed over the inlet side hole to reduce the inlet opening to the correct size.

A motor or arbor support can be built from plywood and dimension lumber. The dimensions of the motor support can be determined by examining the motor to be used, or from the arbor if a belt drive is to be used.

The discharge opening can be converted from the rectangular shape as constructed to the more standard round pipe shape with a light sheet metal fitting.

A handle of pipe or dowel can be added to make the unit easier to move.

A switch mounted in a standard electrical box should be installed for safety and convenience. Be sure the wiring is of adequate size for the amount of current drawn. Use a three-wire grounded plug on portable fans. Be very careful to protect yourself and others from electrical shock. Many people are accidentally electrocuted every year. Ask for professional help if you are not fully confident of your own ability and knowledge.

The photo is of a 9" x 3" fan driven by a 1/4 hp motor that does very nicely as an exhaust fan for welding. The inlet is 5" diameter and the discharge is 3".

MAKING AN ALL METAL HOUSING

We may need to move gases other than air that are corrosive, gases that carry abrasive materials suspended in them, gases of high temperature and/or humidity. In such cases, the fan housing should be metal.

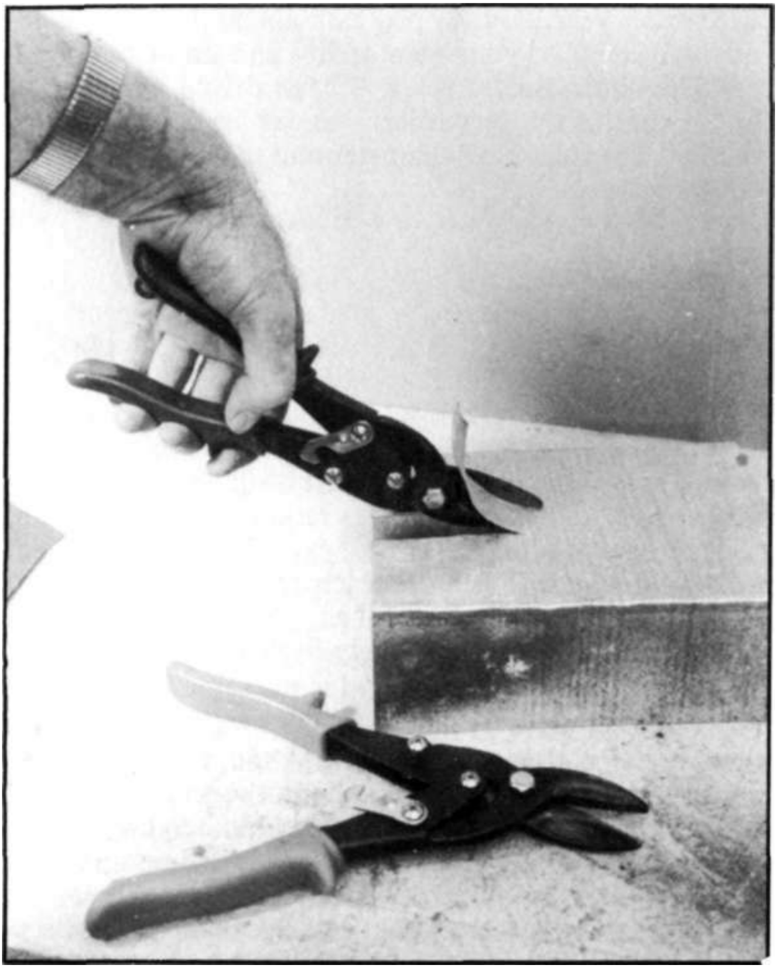
It is not as difficult to form a sheet metal housing with ordinary hand tools as might first be imagined.

In most cases light sheet metal will be used, but if a very strong welded assembly is required, consider using 16 gauge or heavier metal. A heavy metal housing may need welding which is no problem if you have the equipment and the skill. Even metal as light as 22 gauge can be arc welded, although it is likely to warp and buckle.

The general layout method is the same for all scrolls but a margin must be added to the profile when lighter sheet metal is used so that a seam can be made. If the housing must be absolutely air-tight, the seam can be soldered or brazed. It may also be acceptable to use a caulking material so long as it will withstand temperature and other operating conditions.

Steel sheet metal as heavy as 22 gauge can reasonably be cut with aviation pattern snips which are made in both right and left hand cut as well as straight cut. You need at least the right and left hand snips if you hope to produce a smooth cut. The left hand snips do approximately the same thing as the straight cut snips if you must economize. It is the compound lever action of these snips that makes it possible to cut heavier metal with them. By using the right and left hand snips alternately the waste can be curled away so that the cut may advance.

Pliers may have to be used to pry the waste away when heavy metal is being cut. The cut will be much



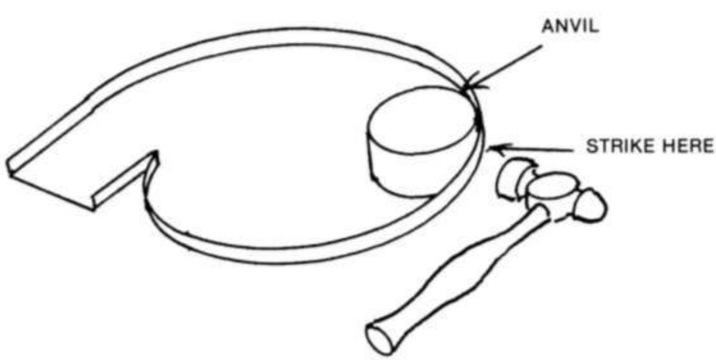
smoother if you do not permit the blades to close at the tip. Rather, re-open the snips, and take a fresh bite just before they close entirely so that the metal won't be deformed leaving a jagged point. With patience and practice you'll be able to produce smooth cuts.

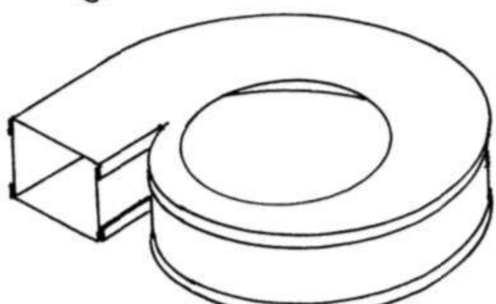
If a smooth exterior surface is important for any reason, then the addendum for the seam should be added to the profile. The addendum can be added to the shell, but the final appearance will not be as pleasing.

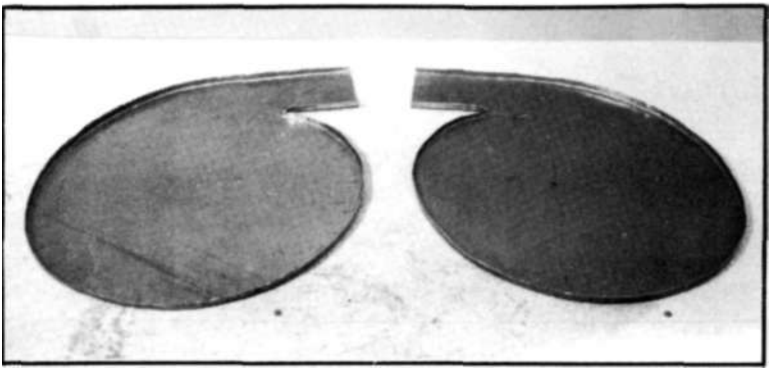
It is a simple matter to turn a flange on the profile if a turning or edging machine is available, but such equipment is usually not available in the home shop. Although a bit tedious, a simple flange can be formed with pliers, hammer and anvil. The secret of success lies in raising the flange gradually.

First, bend the flange up about 20 degrees with the pliers all the way around. Make a second pass with the pliers to raise the flange a bit beyond 45 degrees.

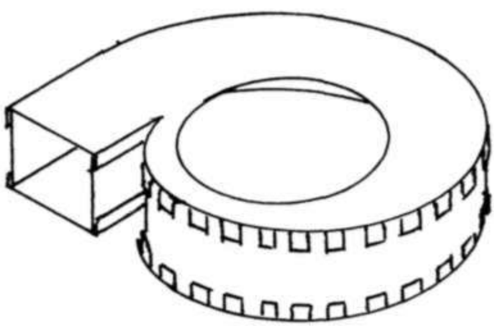
A third pass can be made with the pliers, or you can then begin to finish the job with hammer and anvil. The anvil can be any piece of metal that has more mass and weight than the hammer and that will fit inside the curve of the flange. A casual stroll through the scrap metal yard will turn up something appropriate: a small fly-wheel, cast iron pulley, a short piece cut from a bar of 2" or 3" round. An old flat iron such as grandma used to iron grandpa's Sunday shirt makes an excellent anvil for such work. Although the flange will be rough and wavey at first, you will soon learn how to strike it just right to cold forge it smoothly against the anvil.





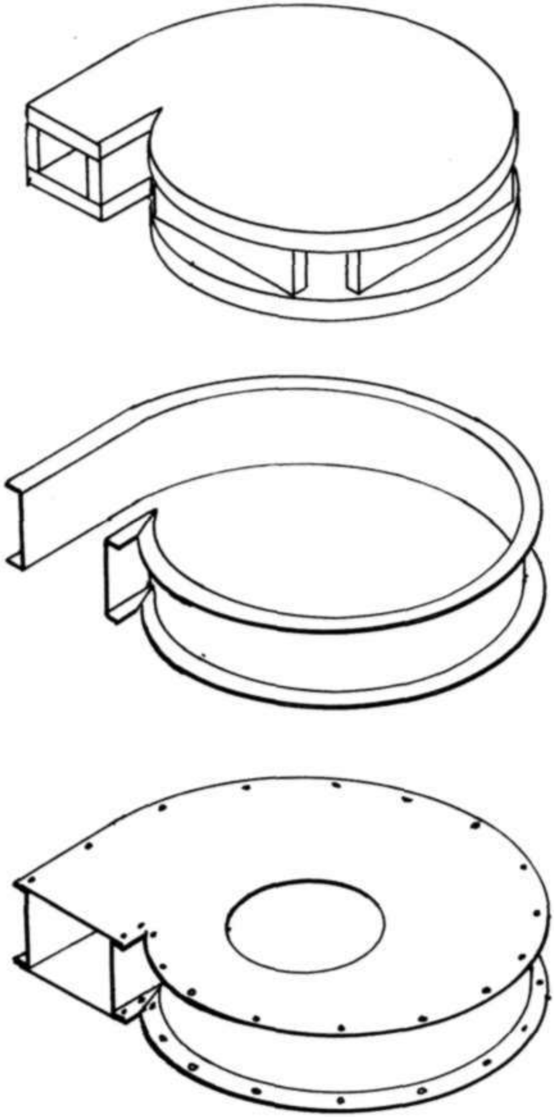


An alternate method of forging the flange is to cut notches in the addendum so that tabs from 3/8" to 1/2" wide are formed. Then each alternate tab can be bent at right angles to form a support for the shell and then the remaining tabs are bent over the shell to complete the seam. When carefully done the appearance can be quite good and such a seam can be made tight by soldering or brazing.

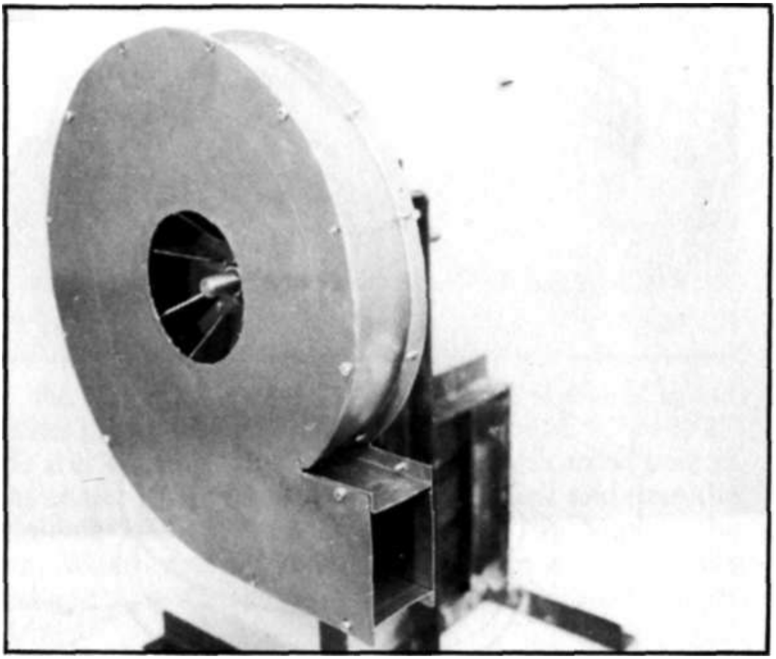


While either the flange or tab seam might be the most attractive, both are tedious to form by hand. Since appearance is usually not important in a fan housing, we can use the easier method of putting the flange on the shell instead of the scroll profile. In doing so, most of the time-consuming labor is eliminated.

By forming up the shell so that the flange faces outwards, a convenient surface for assembly is formed. In



addition there will be no screw ends inside the housing. If light sheet metal is used, it will be easy to use machine screws and nuts for assembly instead of self-tapping sheet metal screws. When turned to the outside the flange must stretch to conform to the curve of the scroll instead of compressing and puckering as happens when turned to



the inside. This method produces a very neat and practical fan housing for most purposes.

A 3/8" wide seam is a convenient width with which to work. Twice this width, or 3/4", must be added to overall width of the housing to provide two flanges. The scroll profiles are layed out with a 3/8" addendum all around so that they will fit over the flanges on the shell. The shell is rolled around a plywood form made to the size of the scroll.

One side of the scroll can be permanently sealed by riveting, soldering, brazing, or caulking. The other side should be attached with self-tapping screws or machine screws and nuts to allow removal in the event repairs are needed. A gasket must be installed on this removable side if an air-tight seal is necessary.

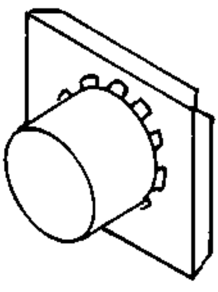
DISCHARGE FITTINGS

Since the discharge opening of the fan will be rectangular and since many applications require round pipe fittings, it will probably be necessary to attach a transition fitting to the discharge opening.

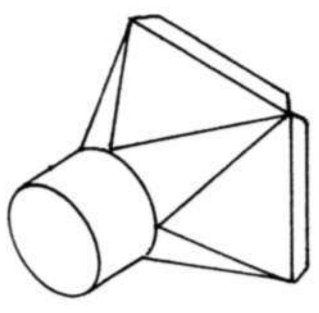
It may be acceptable in some cases to simply fit a plate with a short round collar over the discharge opening. It is a simple matter to cut a round hole in the plate and to install a collar by notching the edge and folding tabs.

A formal transition piece is easy to make and will offer less friction to the air. Not only that, it will look better. There are some applications where a formal transition fitting must be used. It is well worth the effort to learn the simple yet fascinating layout procedure called "triangulation." Once you have mastered the process you can produce neat sheet metal transition fittings for any purpose.

The only problem with triangulation is that the procedure looks complicated at first glance. But it isn't. Since the average pattern layout time is less than 15 minutes per fitting, you know the procedure can't be too awfully difficult.



TAB COLLAR



TRANSITION

TRIANGULATION

As the name of the process implies, triangulation is a process of solving a series of triangles in order to find the true length of the lines from their apparent lengths as seen in a plan view of the transition piece.

Notice that in the isometric view of a transition on page 49 that a number of triangles are seen contained within the shape of the transition body. The lines that extend from the lettered corners of the rectangular base to the numbered points on the circular part are actually the bending lines.

If you already had on hand a transition of the correct size, it would be a simple matter to measure the lines and progressively lay out a pattern for a duplicate on a piece of sheet metal. Such is not usually the case, though. Believe it or not, it is actually easier to take the dimensions from a full size plan view than it would be to take them from an existing transition.

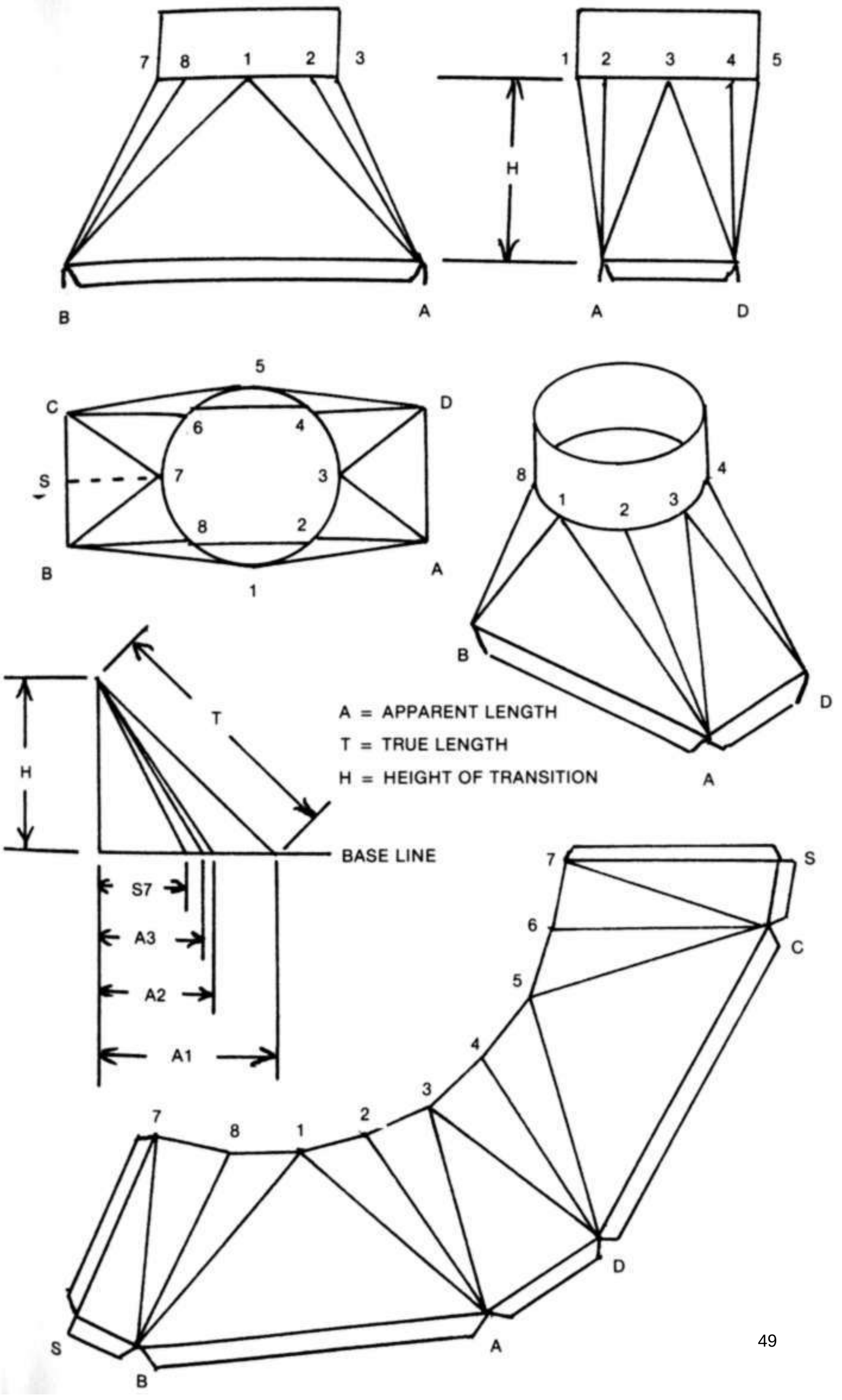
Because lines are not usually drawn on a transition piece, you must learn to visualize them in your mind. In your first attempts actually drawing the lines will probably help you develop the pattern, but as you master this simple art, you will find the lines are not necessary.

On page 49 is an isometric view of a transition piece we'll attempt to layout as an example. Below it is the pattern needed to form up that shape.

The seam, two sides of the base, and a portion of the circular collar are not visible in the isometric view. Fortunately, invisible parts are identical to those we can see. The isometric view has no value in the layout process since none of its dimensions are of true length. It only helps us visualize the shape of the figure we will lay out. Notice that the same triangles contained in the isometric shape are seen also in the full pattern.

The top plan view shows the true circular size as well as the true size of the base rectangle. The circular part is divided into eight equal parts which are numbered for reference. The corners of the rectangular base are lettered, and the location of the seam is indicated by the dotted line from *"S"* to point 7 on the circular collar.

If the transition were cut from point "S" to point 7 and flattened out, you would have the full pattern stretched out in front of you. We'll do this cutting and flattening in our imagination as we transfer dimensions


7 8 1 2 3
B A
1 2 3 4 5
H
A D
5
C D
6 4
S 7 3
8 2
B A
1
8 4
1 2 3
B D
A
T
H
A = APPARENT LENGTH
T = TRUE LENGTH
H = HEIGHT OF TRANSITION
BASE LINE
S7
A3
A2
A1
7 S
6 C
5
4
3
7 8 1 2
D
S A
B

one at a time from the top plan view to the full pattern. The result is called "the stretchout."

Above the top plan view is a side elevation view drawn full size. To its right is an end elevation view. Because neither of these views are used in the layout procedure, it is not necessary to draw them each time you lay out a pattern. They are provided to help you see the figure from all useful angles.

Notice that only a very few dimensions in the side and end elevations are true in length. In pattern layout practice only the height is taken from the side or end elevation, and all other dimensions are taken from the top plan view. It is not necessary to draw the views to know the height, but the top plan view must be drawn, and its circular part divided.

Every dimension seen in the top plan view is true, but some of them actually represent the base of a triangle. Because neither the vertical leg of the triangle or its hypotenuse can be seen in the top plan view, a "solution triangle" is needed to transfer a few of the dimensions to the full pattern.

This sounds confusing, but it is not at all difficult in practice. Absolutely no mathematics is involved, and you do not even have to understand Pythagorian theorem to solve the triangles. We simply draw a duplicate triangle and take the dimension of its hypotenuse from the duplicate with the dividers or compass.

All triangles are made up of three sides and three angles. A right triangle always contains a 90-degree angle between two sides. If you draw a duplicate of an existing triangle such that base and vertical dimensions of both are the same, then the hypotenuses will also be identical.

Each of the lines in the isometric view and the full pattern extending from the corners of the rectangular base to the circular part, such as *A-1*, *B-1*, and *D-4*, is actually the hypotenuse of a triangle. The same lines seen in the top plan view are much foreshortened because they are being viewed from an angle. If we could remove these foreshortened lines, we would see the base of the triangle. The base is seen in true length since we are viewing it "straight on" and not at an angle.

The height of all of the triangles in this instance are the same. Since the vertical line that represents the height of the triangle is merely a dot in the top plan view, we must take the true height from the end or side elevation.

It is now a simple matter to draw a duplicate triangle on another piece of paper with the base length taken from the top plan view, and the height taken from the side or end elevation.

For example, the distance from *A* to 1 in the top plan view represents the base of a right triangle the height of which is *H* in the end elevation. The hypotenuse of this triangle is the true length we use to lay out the pattern. Each of the true dimensions we will need is found in the same way.

You don't need to draw a new triangle for every triangle that must be solved. You can draw a straight horizontal line that is longer than any of the dimensions you'll be working with in the top plan view, and let it serve as the base line. Next, draw a vertical line at exactly right angles to it with exactly the same height as the transition piece. This single "L" shape drawing can be used for all the solution triangles since all of them are right triangles with the same heights but with different base lengths. For the sake of clarity, let's call this "L" our solution jig.

Notice that the height does not include an allowance for a joint at the base or at the collar end of the transition.

When the apparent length of any of the dimensions in question is applied to the base its true length can be easily measured as seen in the sample on page 49. For instance, if the dimension *A* 1 is taken from the top plan view and transferred to the base of "L" shape solution jig, then the true length of *A*1 is hypotenuse. We can lay out dimension *A2* taken from the top plan view of the solution jig, and find its true length by measuring the hypotenuse. We use the solution jig over and over. There is no need to construct a new one each time.

In practice, once you've put the dimension on the base of the solution jig, you don't need to actually measure the hypotenuse. Use dividers or a compass to move dimensions from the top plan view to the solution jig base line. The length of the hypotenuse which represents the true length can be transferred to the growing full pattern with the same dividers or compass. It takes much longer to describe the process than it does to perform it.

Now compare the lengths of the line drawn from the corner marked *A* to the point on the circle marked 3 as it appears in the various views on page 49. Notice that its length is different in each of the views. That is because line *A3* is not shown in its true length in any of these views. The full pattern length of *A3* must be the true length.

The length from *A* to 3 on the top plan view is transferred to the base line of our solution jig. We draw in the hypotenuse to form a solution triangle. The length of the hypotenuse is the true length of line A3. It is this length that is transferred with dividers to the full pattern.

The process is actually that simple.

Each of the bending lines seen on the isometric view from the lettered corners to the numbered points on the circular collar is actually the hypotenuse of a triangle, the base length of which is clearly seen in true dimension in the top plan view. As you look straight down at the triangles from above you can see only the length of the base. It is a simple matter to transfer that apparent dimension to the solution triangle to find the true dimension in its hypotenuse.

A practice exercise will help to make it all very clear. You need a clean piece of paper, compass or divider, a ruler and a pencil. A draftsman's triangle is useful to draw the corners and the solution triangle at true right angles. But for heaven's sake, don't panic. The work is not exact or demanding of any artistic talent.

First, we need a top plan view. Draw a rectangle to represent the base of the transition piece as seen from above. The illustration on page 49 happens to be 2" long and 3/4" wide, but you can use any dimension that is convenient.

Use the compass or dividers to draw the circle exactly in the center of the rectangle. The circle in the illustration is 1" diameter. You can locate the circular part off center in actual applications, but in this exercise doing so just makes it a little more work to develop the pattern because each dimension is unique. Divide the circle through its center both vertically and horizontally and then divide each of its quarters exactly in two to make up 8 equal divisions.

Letter the corners, and number the divisions. Draw lines from each of the corners to the nearest point on the circular collar to indicate the bases of the triangles from which the true lengths of the bending lines will be found.

Draw a horizontal line that is longer than any of the base lines. Next, draw a vertical line at a right angle to it. The height of this vertical line must be the same as the true height of the transition piece. The height in the illustration is 1".

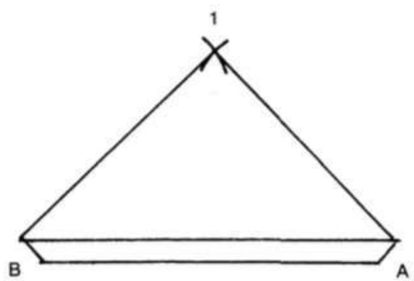
Now you have everything you need to lay out the full pattern. We'll do it one step at a time, and it will not take very long. You're risking nothing but a bit of paper and a few minutes of time.

In practice, the distance between the points of division on the circle are not precisely true but are usually close enough for ordinary work. There is also a very slight error in the height of the triangle used to establish points 1 and 5 on the full pattern, but the 1/32" error is not large enough to justify drawing a second solution jig, or solution triangle. For the purpose of this exercise you can ignore such tiny errors.

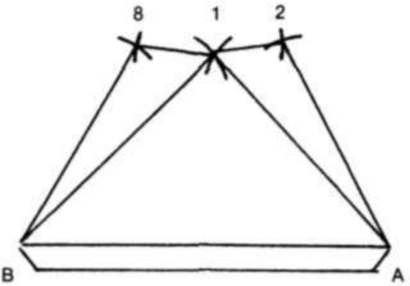
The distances between the circular divisions and the rectangular base dimensions can be transferred directly from the top plan view to the full pattern layout.

The layout may be started with any length from the base of the transition piece, although it is customary to begin with the widest dimension whenever possible. To start, simply draw a line as long as the length of line *B* to *A (BA).* Make a tiny sharp punch mark at each end of the line from which the compass or dividers can pivot. The narrow flange that will be used to fasten the transition to the fan outlet can be added now if you like. Points *A* and *B* are the first reference and pivot points in the layout procedure.

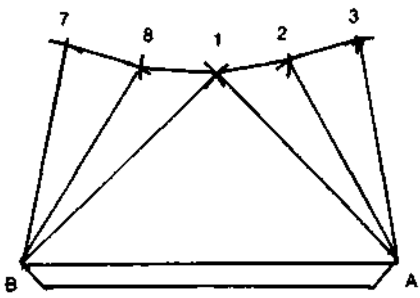
Find the length of *A*1 on the top plan view, and apply it to the base line of the solution jig. Draw in a hypotenuse to form the solution triangle. The length of the hypotenuse is the true length of *A*1. Spread the dividers, putting each point on one end of the hypotenuse. To transfer this true length to the full pattern, use the punch mark at *A* as the pivot point and scribe a short arc with the free divider point. Since *B*\ is the same length, use the same divider separation and point *B* as the pivot to scribe a second arc. Make this second arc intersect the first. The point of intersection of these arcs establishes point 1 of the circular part of the pattern.



Set the dividers on the distance between 1 and 2 on the top plan view, and scribe short arcs to represent points 2 and 8 on the pattern layout. Draw a solution triangle to find the true length of line *A2* in the top plan view. Using both *A* and *B* as pivots, scribe arcs with the true radius *A2* intersect with arcs 2 and 8 on the pattern layout.



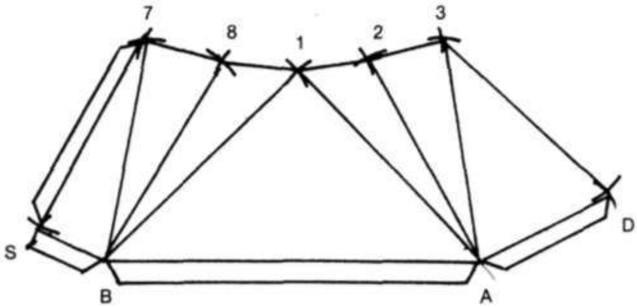
It should now be obvious that pattern development is not much more difficult than a child's "connect-the-dot" puzzle. As you might expect, the next step is to scribe short arcs to represent points 3 and 7. Then draw a solution triangle using the apparent length of *A3* to get the true length of *A*3. Note that *A3* and *B7* are the same length in this example. Points 7 and 3 on the full pattern are formed at the intersection of the arcs. At this point one half of the circular collar portion of the pattern is complete.



Now it is necessary to transfer the end dimensions of the rectangular transition piece base so that the circular part can be advanced. The dimension *SB* is taken directly from the top plan view and is transferred by scribing a short arc to the left of pivot point *B* on the pattern layout. Draw a solution triangle to find the true length of *S7* using the apparent length taken from the top plan view. Set the dividers to this true length. Using point 7 on the pattern layout as a pivot, scribe a short arc intersecting the arc drawn from point *B*.

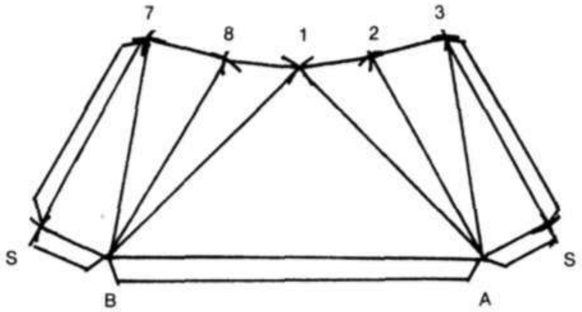
Sooner or later, you will need to add an addendum at the base for the joint and at the left hand edge for the seam. You may wish to make that addition now.

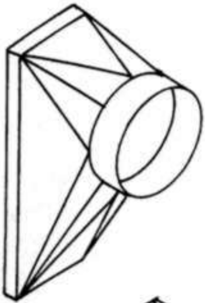
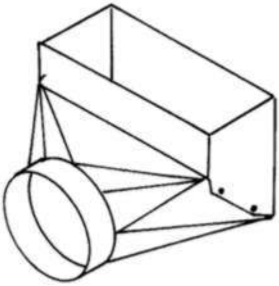
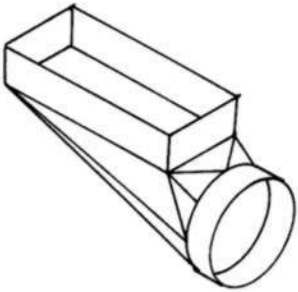
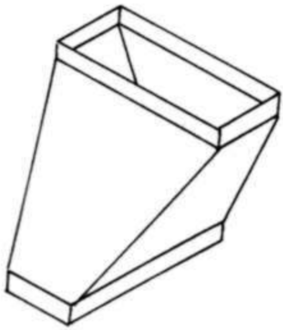
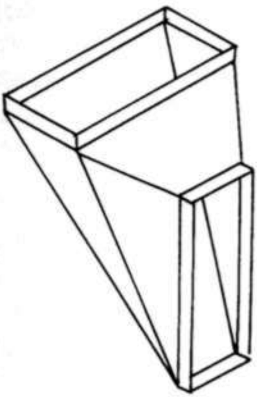
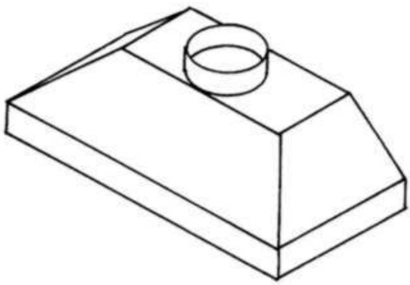
In the same way that *SB* was developed, the dimension of *AD* is transferred directly from the top plan view to the pattern. The true length of *D3* is found and is used to scribe an arc intersecting with the *AD* arc to establish point *D* on the pattern layout.



The full pattern is now completed by simply following these procedures to lay out the remaining triangles seen on the top plan view. After you have worked your way around the top plan view, you will have a full pattern in which the full circle and all four sides of the rectangular base have been established.

With the collar centered in the top plan view, the transition piece is symmetrical. You could save time by laying out half a pattern, cutting two, and assembling the halves. You save layout time, but increase your assembly time. This option exists only for symmetrical transition pieces.



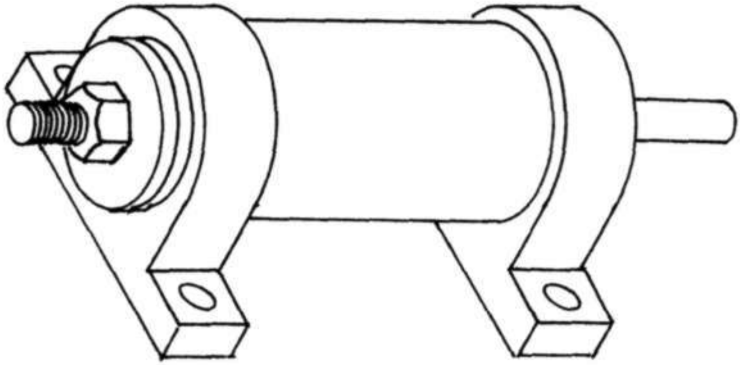
The triangulation method works well with many shapes and configurations that may prove very useful in forming hoods, dust collectors, and other fittings in your shop. A few are illustrated on page 57.

MOTOR AND DRIVE APPLICATIONS

The simplest and most economical way to drive a fan is to mount the wheel directly on the motor shaft, eliminating the need for bearings, additional shafts, pulleys, belts, etc. If you are building a fan for a specific job, and the air requirement is constant, a direct drive fan is usually the best choice.

In some applications it may be necessary to change the fan speed from time to time, or the motor on hand may not be of the correct speed, or its shaft or bearings may not be substantial enough to support the wheel directly. In such cases a belt drive is always best.

Pillow block bearings are available in many sizes and grades. Ball bearings are generally considered superior to sleeve bearings but they are also more costly. When properly installed and lubricated, sleeve bearings will last many years in intermittant service as in the home shop. There are some very convenient and durable "saw-arbors" available that offer a pair of bearings, pulley, and a shaft with a threaded end, heavy washers, and nut. Such an arbor makes mounting a fan wheel quite easy.



A simple framework of wood or metal is all that is needed to mount the motor or the motor and belt-drive to the fan housing. Some motors have studs at the drive end that provide a convenient way to mount to the fan housing. Brackets can be fitted to the motor, too. Combinations of these two approaches may have to be used to mount a peculiarly designed motor.

Pulley sizes for a belt drive are easily worked out by simple calculations. The product of the driving pulley diameter multiplied by its speed in RPM will always be equal to the product of the driven pulley diameter multiplied by its speed in RPM.

For example, a 3" motor pulley at 1725 RPM will drive a 1-1/2" fan pulley at 3450 RPM.

$$3 \times 1725 = 5175$$

Since 1-1/2 x RPM must also equal 5175,

$$\text{RPM} = \frac{\text{product}}{\text{pulley diameter}}$$

$$\text{RPM} = \frac{5175}{1\text{-}1/2} = 3450$$

If you have a fan that must run at 3450 RPM and a motor that runs at 1725 RPM, you need to know what size pulleys to apply. Since the fan must run faster than the motor, the fan pulley must be smaller than the motor pulley. Select a pulley for the fan from those you have on hand, suppose a 1-1/2" diameter. Multiply its diameter by the desired speed.

$$1\text{-}1/2 \times 3450 = 5175$$

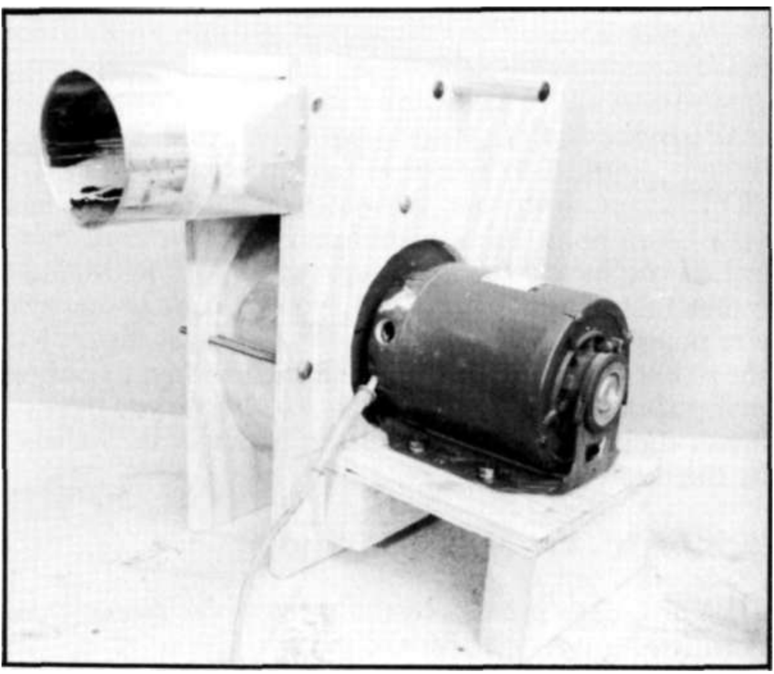
Then divide the product by the speed of the motor to get the correct pulley size for the motor.

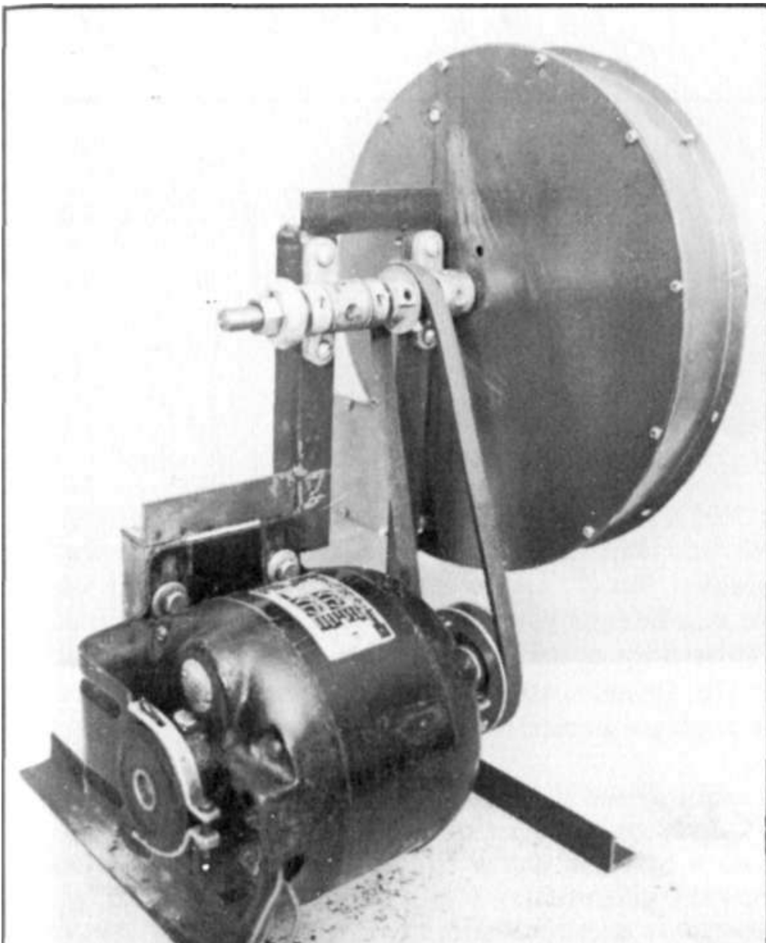
$$\text{diameter} = \frac{\text{product}}{\text{rpm}}$$

$$\text{diameter} = \frac{5175}{1725} = 3$$

Divide the greater speed by the lesser speed to get the speed ratio. The ratio of pulley diameters must be the same. In the example above the ratio was two to one. The fan speed is twice the motor speed, and the motor pulley is twice the diameter of the fan pulley. You could combine a 4" driving pulley with a 2" driven pulley for the same effect, or any combination of pulleys so long as their diameters are of the correct ratio.

Obviously, the calculations work exactly the same when a speed reduction is required. A 1-1/2" motor pulley at 3450 RPM will drive a 3" fan pulley at 1725 RPM.





CHAPTER IV

Fan Testing and Air Measurement

MEASURING AIR

So far, we have designed a centrifugal fan with a 10" x 3" wheel to run at 1725 RPM and deliver 250 cubic feet of air per minute at a pressure of 1.3" water column. Such a fan can be built and successfully applied to a job without adjustment or modification. Some applications, however, will require more accurate control of air volume and pressure. Some means of measuring these attributes will be needed.

Although air may seem intangible, it has weight and mass. Air is usually described in terms of its pressure, velocity and volume of flow, of which pressure is most easily measured. Since there is a relationship between pressure, velocity, and volume, the unknown values can be calculated once the pressure is known.

Very sensitive, accurate instruments have been developed for laboratory use, but these are not usually found in the small shop because of their high price. Fortunately, it is easy to build simple measuring equipment of sufficient accuracy for our needs. We can make all of the measurements we need with a simple manometer, pitot tube and orifice. We need no other equipment.

Sometimes the exact composition of air must be considered during fan design. Much of what has been learned in fan design can be found in engineering formulas and tables. For most of our needs we can use "standard air," which is what you and I breathe at a temperature of 68 degrees Fahrenheit, a relative humidity of 50%, and a barometer reading of 29.92" of mercury. Such standard air weighs approximately .075 pounds per cubic foot, and it is composed of approximately 20% oxygen and 80% nitrogen by volume, with very small amounts of other gases and pollutants.

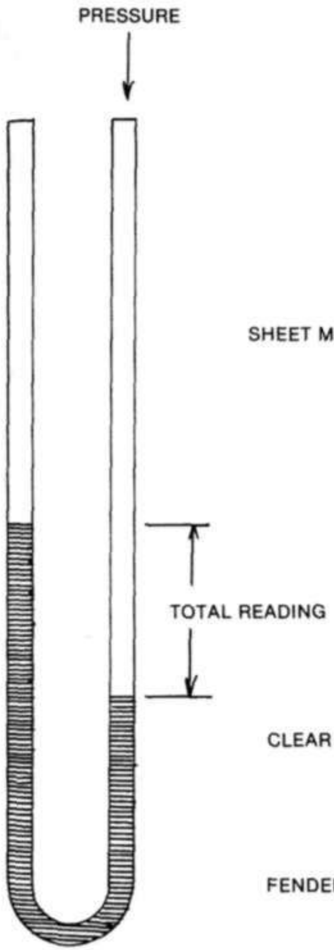
As air's density and weight varies with its temperature and humidity, its behavior in a fan system will vary. Cold air is more dense, and thus heavier, so a fan will deliver a greater weight or quantity of air when it is cold than when it is warm. Racing auto engines and airplane engines develop more power in cold air because there is more oxygen in a given volume of cold air than in hot air. It will require more power to deliver cold air, and this can be a significant factor.

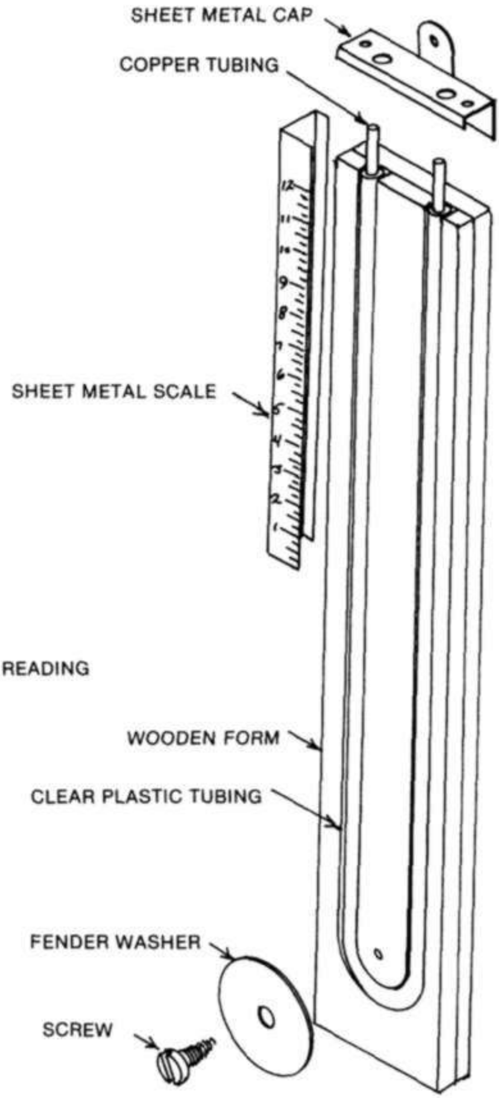
In an application where a fan is to provide air to a furnace, we don't have exotic equipment to weigh the air. The best we can do is provide a fan with excess capacity, adjusting its delivery with a simple gate or shutter at the intake when observing the nature of the flame. In this way we dispense with the need for expensive, complex equipment and the technical formulas required to interpret and apply the readings.

THE "U" TUBE MANOMETER

Although fans can be built to deliver very high pressures sometimes needed by industry, most fans develop low pressures, usually below one pound per square inch. Mechanical gauges are available to indicate these low pressures but they are very costly. Most of us could not justify even a moderately priced gauge for the home shop. Fortunately, an expensive gauge is unnecessary. A simple manometer will do.

A "U" tube manometer displays the effect of pressure on a column of liquid within a transparent tube. When the tube is filled with water, it is called a "water gauge." One ounce of pressure will raise the water column 1.73 inches. If the tube were filled with mercury





which is much more dense than water, the same one ounce of pressure would raise the mercury column just .127 inches. Water is obviously the correct gauge liquid for our low pressure needs.

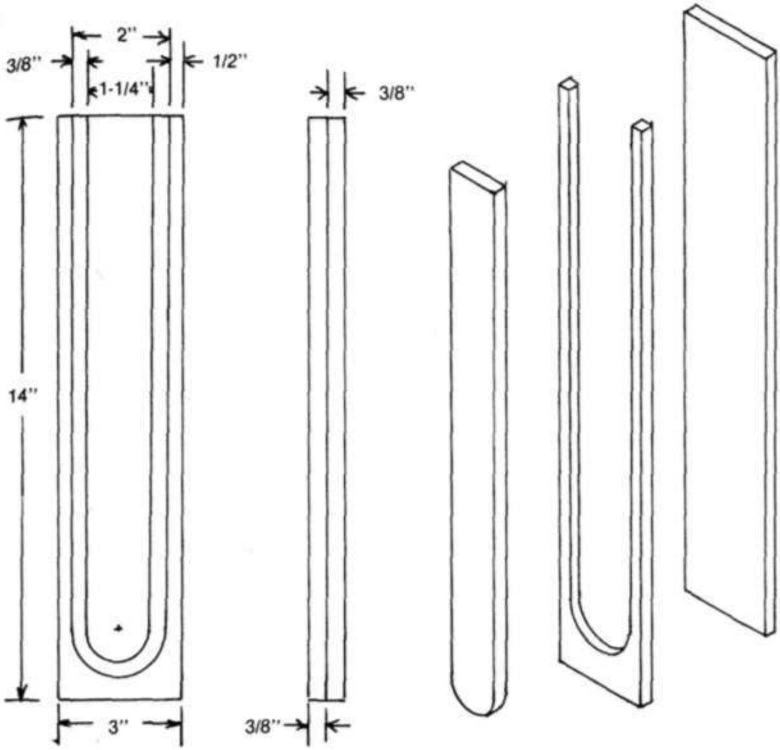
A manometer is simply a transparent tube that is formed in the shape of a "U" and that is partially filled with water. When pressure is applied to either leg of the gauge, the water is forced down in one leg and up an equal amount in the other. The difference between the level of the water in both legs provides a reading of the pressure applied.

A simple direct reading scale can be prepared by mounting a scale of sequentially numbered marks at 1/2" intervals next to one leg. A half-inch rise in one leg would be a 1" difference since the other leg will move a 1/2" in the other direction.

A very durable and accurate water gauge can be made with clear plastic tubing and other common materials. Most commercial manometers use a tube with an internal diameter of 3/16" to 1/4" although other sizes are sometimes used. Inexpensive clear plastic tubing with a bore of 1/4" and an outside diameter of 3/8" is readily available in most hardware stores. Not only does it make an excellent gauge, but it also serves as a convenient connecting hose to the pitot tube or the gauge port in the fan system.

It is important to minimize distortion in the "U" bend of the manometer. Since plastic tubing is soft and flexible, it requires an accurately-made mounting to mold the bend and to hold the legs erect. With 3/8" thick wood and simple hardware, a mount can be built that provides a uniformly shaped channel of 3/8" width into which the tubing can be pressed.

All of the elements of a water gauge are shown in the drawing on page 65. The wooden form provides a snug channel into which the tubing is pressed. A fender washer and wood screw hold the tubing in place at the bottom end while a simple sheet metal cap with holes for the 1/4" copper tubing holds the manometer tube in the mount. In addition to anchoring the manometer tube in the mount, the copper tubing also provides a convenient connection for the hose to the pitot tube or test port. A simple scale marked out on sheet aluminum that is bent around the body of the mount to a snug sliding fit enables you to "zero" the gauge before use. If each numbered increment is 1/2" then the readings indicated are 1" of water column.



Dimensions for a wooden form are suggested on page 67. Although a 3/8" channel could be routed out, it is probably easier to make the form in 3 pieces and join them with brads and glue. A coat of clear varnish just before the tube is finally pressed in will make the whole unit more durable and attractive. A small amount of food coloring in the water makes the column more visible. Although laboratory practice sometimes calls for distilled water to be used, our colored tap water will not affect the accuracy of our casual measurements.

The readings of the manometer in inches of water column are converted to ounces per square inch by dividing by 1.73. A scale can be prepared to read directly in ounces per square inch with major divisions at 55/64" intervals since 1.73" of water column is equal to 1 ounce per square inch. The readings in ounces per square inch are converted to inches of water by multiplying by .58.

$$\frac{1.73''}{2} = .865'' = \frac{55}{64} \text{ inches (approx)}$$

$$= \text{a 64th short of } 7/8''$$

(You must divide 1.73 by 2 because one side of the manometer rises while the other falls. The difference between .865 and 55/64 is about .006"—insignificant in our work.)

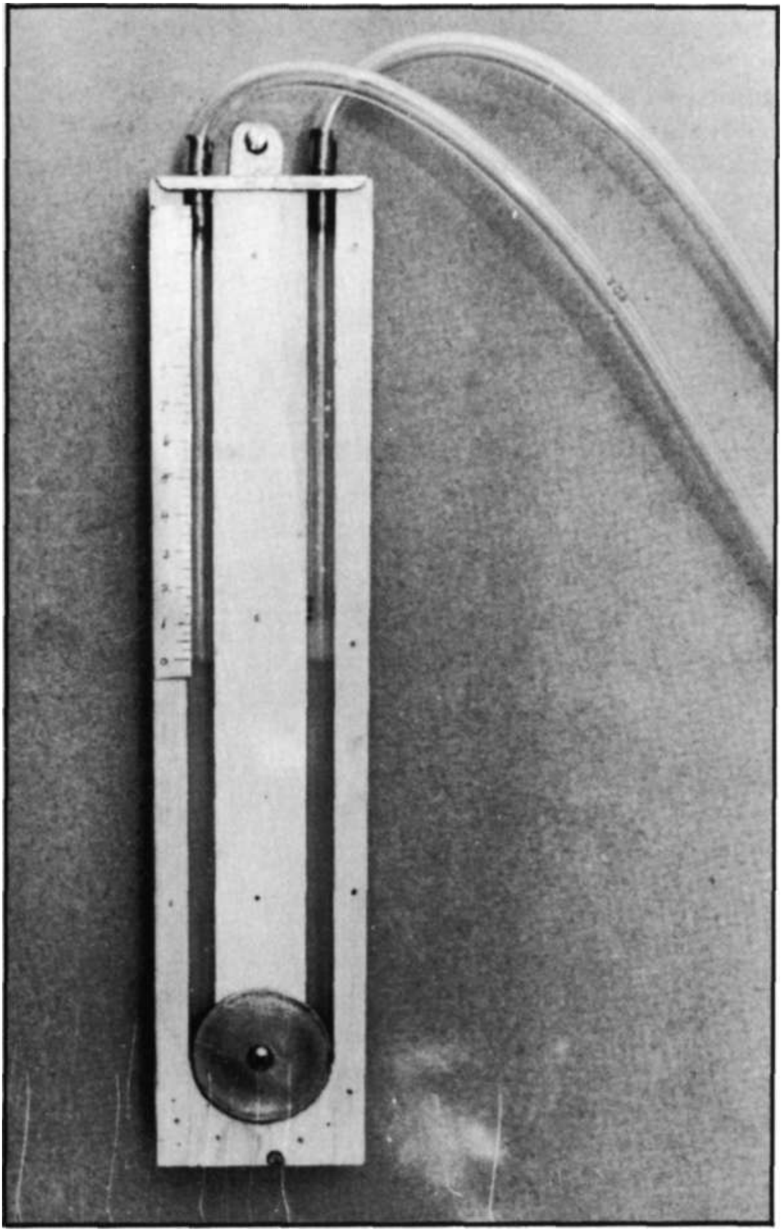
MEASURING PRESSURE

Soon after you build the manometer, you will discover how sensitive it is. If you connect a hose to either leg and blow gently at the end of the hose, the opposite column will rise. Although some commercial manometers are graduated finely enough to read even very tiny pressures, we need not be concerned with anything finer than 1/8" water column.

We have already discussed the three types of pressure found in a fan system: velocity pressure, static pressure, and dynamic pressure. Of these, velocity pressure is the most useful for design work.

While the dynamic pressure is the source of both static and velocity pressure, its measurement is of very little use to us in testing. Since dynamic pressure acts only in the direction of flow, you can read it by merely connecting a hose to the manometer and placing the opposite end squarely into the air stream. This will be the highest reading in the system since it indicates the total pressure.

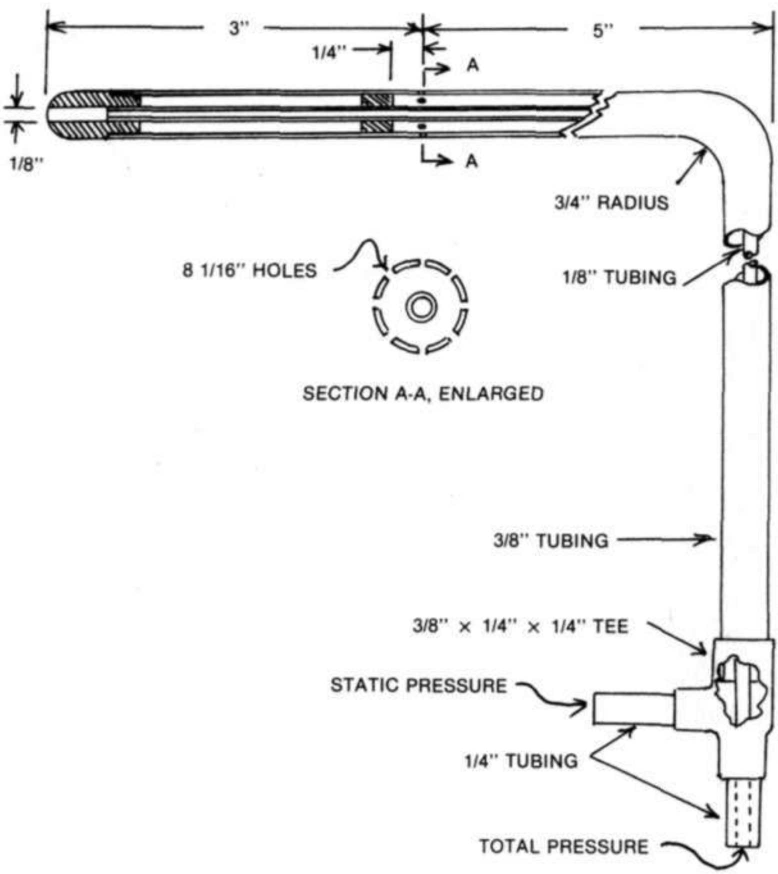
Static pressure is the result of resistance to the dynamic pressure, and it acts in all directions. It can be read directly by placing the test hose anywhere in the system where it will not receive the forward pressure of the stream of air. The reading is normally taken from a tap in the duct that is at right angles to the stream and where no obstruction might deflect the stream into the tap.



Dynamic pressure minus static pressure is velocity pressure. When one leg of the water gauge is connected to a port to read dynamic pressure and the other leg is connected to a port to read the static pressure, the velocity pressure can be read directly from the manometer. The static pressure and dynamic pressure oppose each other. The difference, velocity pressure, is, in a sense, calculated within the gauge.

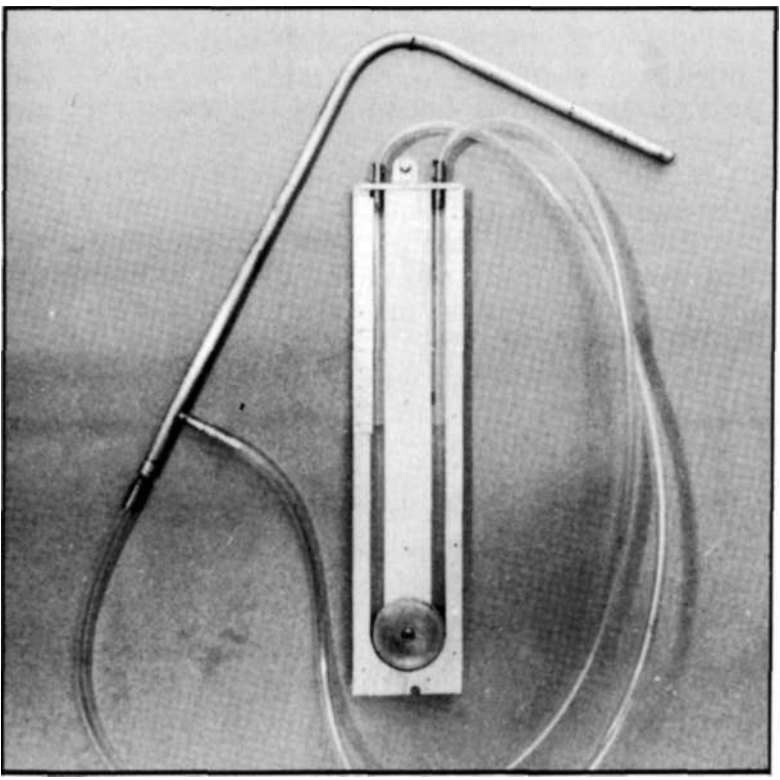
THE PITOT TUBE

While two gauge ports might be the best idea when a gauge is to be installed permanently in a system, a "pitot tube" ("pee-toe tube") is the correct tool to use when a number of fans or systems are to be evaluated. It is a tube within a tube that applies both dynamic and static pressure simultaneously to the water gauge.



Pitot tube construction and principle of operation is simple. The 1/8" copper inner tube extends all the way through the outer tube. When aimed into the air stream, total dynamic pressure is delivered to one side of the gauge by a hose connected to the lower end shown in the construction view.

Notice that there are 8 small holes in the outer tube 3" from the rounded end at section *A—A.* Static pressure enters the large tube here and is applied to the other side of the gauge through a hose connected to the tee. The gauge will indicate the difference between dynamic pressure and the static pressure which is the velocity pressure.



Although the specifications and dimensions given in the drawing on page 70 differ slightly from official specifications for a laboratory grade pitot tube, the difference in accuracy is negligible for our needs. If you need a laboratory grade instrument you can find exact specifications in engineering manuals.

For the outer tube 3/8" copper tubing is chosen because the officially specified 5/16" tubing can be difficult to find. The 1/8" inner tubing may be hard to find as well. I salvaged mine from a discarded thermocouple taken from a gas furnace safety valve. Sometimes 1/8" copper tubing can be found in refrigeration units or in automotive oil pressure gauge connections. The 3/8" x 1/4" x 1/4" copper sweat tee is probably most easily purchased from a commercial refrigeration supplier.

You'll need a nosepiece and a center support near the ring of holes. These parts can be fabricated from 3/8" diameter brass or steel rod, plastic, or even wood turned to shape on a small lathe or in a drill chuck. Metal parts can be soldered, but you need an appropriate glue for other materials.

It was easy to turn a brass nosepiece on my small lathe and solder it into place.

Use new copper tubing if at all possible. Copper tubing gets very stiff with age and bending. You might have a problem with the short radius bend if you use old tubing.

A propane torch provides ample heat for soldering the joints. Clean the joints brightly, and use rosin core solder or solid solder with rosin paste flux.

The eight 1/16" holes in the outer tube should be spaced as equally as possible, and free of burrs both inside and out. Official specifications are for holes of not less than .040" for most classes of work, but there is no need to be that precise here.

Assembly is easy if you follow the correct sequence.

First, solder the 1/8" tubing about 1/4" into the nosepiece. Solder the support piece in its proper location on the 1/8" tubing. After drilling the 1/16" radial holes in the 3/8" outer tubing, slip the inner tube assembly into the outer tube, and solder the nosepiece to the outer tube.

There is no need to solder the support piece to the outer tube, and to attempt to do so would probably make a mess of the radial holes.

Next, solder the 1/4" tubes into the tee, and solder the tee to the free end of the 3/8" outer tube. Now bend the tubes around a 3/4" radius.

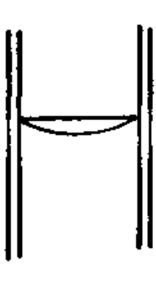
The 1/4" static pressure tube soldered to the tee should face in the same direction as the nosepiece and should be exactly parallel to it. You will not be able to see the nose of the pitot tube when it is inserted into an air stream. You'll be able to aim the nose by observing the 1/4" static pressure tube which will remain outside the air flow.

The final step is to solder the 1/8" tube into the 1/4" tube so that there can be no leakage between the outside of the surface of the 1/8" tube and the inside surface of the 1/4" tube. Polish the instrument brightly, and give it a coat of clear lacquer.

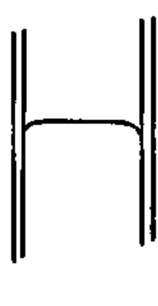
TAKING PRESSURE READINGS

To read velocity pressure directly in any fan system, connect hoses from each of the 1/4" tubes to each leg of the water gauge.

Adjust the manometer so that both legs are vertical. Next adjust the scale so that zero is adjacent to the surface of the water in one leg. The meniscus, or curvature of the surface of the water, must be taken into account. Although the meniscus is slight in small tubes, it can be a significant factor in larger diameter tubes. A water meniscus is concave, but a mercury meniscus is convex.



WATER



MERCURY

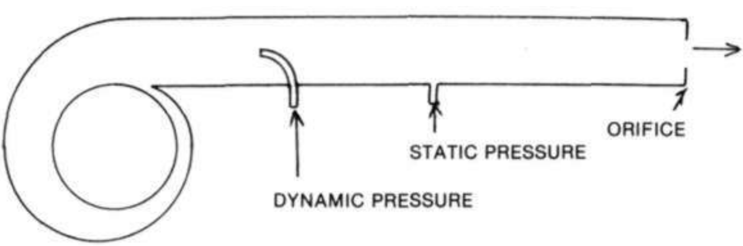
When very precise readings are needed, in hundredths of an inch for instance, the meniscus can introduce substantial error. For our purposes it will be adequate to merely insure that you read consistently either at the edge of the meniscus or at the base. Reading at the base of the meniscus will usually give a more accurate reading.

INTERPRETING AND APPLYING THE READINGS

When a fan system is equipped with gauge ports, each leg of the manometer is connected to a port. The difference between the static pressure in one leg and the dynamic pressure in the other gives a reading of the velocity pressure. These ports are built into the system and cannot be moved.

A pitot tube is mobile and can be moved about to indicate the pressure in various locations throughout the system. Obviously, the permanent ports will indicate pressure only at one point in the system, but the pressure varies to some extent in every system.

Pressure variation is usually greatest in large cross-sectional area duct systems. Usually, many readings are taken as the pitot tube is moved to points on an imaginary grid on the duct cross section. The readings are added together, and the total is divided by the number of readings to give the average reading over the total area of duct. In very small duct systems a single reading in one location will be adequate.



If only a single pressure can be measured, static pressure is desired over dynamic pressure.

Notice that the system shown on page 74 has an orifice that resists air flow creating static pressure within the duct. If a fan has sufficient capacity to deliver enough air to fill the system and to produce a given static pressure as air escapes through the orifice, the static pressure created is an indication of the amount of work being done.

This reading can provide a way of comparing fans. A more powerful fan will produce a greater static pressure reading, while a weaker fan produces a lower reading.

Once a system is working well, the static pressure reading can be used as a standard, or benchmark, for future operation. Should the system performance decline or should the system be modified, this benchmark can help in getting the system up and running again.

Except when used to measure pressure drop in air traveling through an orifice or other metering device, the static pressure reading is not useful in volume flow or velocity calculations. Formulae for such calculations are quite complex, and readings must be very accurate. The equipment discussed here is not sensitive enough to be useful for these calculations.

Sensitive pressure-actuated switches can be installed in a system to light an indicator, sound a warning, shut down the system or even make adjustments that will raise or lower pressure. Static pressure, the force that operates these controls, can indicate whether or not a fan is operating correctly.

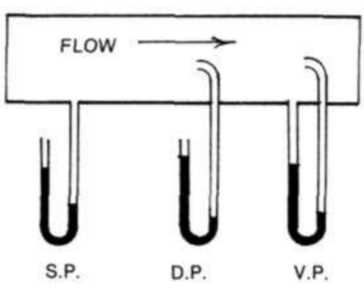
A simple device called a "sail switch" can be installed in the air stream to do the work of the pressure switch. It is simply a broad surface, usually thin plastic stretched over a wire frame, that catches the moving air much like the sail on a sailboat to actuate a mechanical switch. Here, the dynamic pressure is the force that operates the sail switch.

In some instances the dynamic pressure reading might represent the actual velocity pressure. Most often static pressure is subtracted from dynamic pressure to give velocity pressure, and this, in turn, is used for volume flow calculations. When there is no significant resistance to air flow (no static pressure) in a system, a simple disc or propeller fan is usually the best choice.

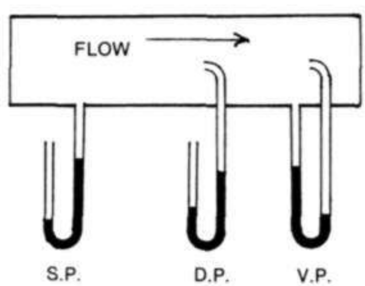
Gauge ports can be installed in either the suction or pressure half of a system. Since pressure is above atmospheric pressure on the pressure half of the system, readings will be positive when one leg of the gauge is left open to the atmosphere.

Readings in the suction half of the system will be negative when one leg of the gauge is left open to the atmosphere since the pressure is below atmospheric pressure.

Readings taken with the pitot tube will be the same whether in the suction or pressure duct when both legs of the gauge are connected to give velocity pressure readings. The velocity pressure reading will always be positive when the inlet tube is pointed into the direction of air flow.



PRESSURE



SUCTION

The most important use of the velocity pressure reading is to determine the velocity of flow, and thus, the volume of air being delivered. A pressure-to-volume conversion chart is shown below.

THEORETICAL VELOCITY OF AIR AT 62 DEGREES FAHRENHEIT DISCHARGED INTO ATMOSPHERE

PRESSURE OZ. P.S.I.	PRESSURE INCHES W.C.	VELOCITY, FT. PER SEC.
.006	.01	6.61
.012	.02	9.35
.023	.04	13.20
.040	.07	17.40
.058	.1	20.90
.115	.2	29.5
.173	.3	36.2
.231	.4	41.8
.260	.45	44.3
.289	.5	46.7
.346	.6	51.2
.404	.7	55.3
.461	.8	59.1
.519	.9	62.7
.577	1	66.1
.866	1.5	80.9
1.153	2	93.5
1.442	2.5	104.0

PRESSURE OZ. P.S.I.	PRESSURE INCHES W.C.	VELOCITY
1.731	3	114
2.020	3.5	124
2.308	4	132
2.579	4.5	140
2.885	5	148
3.462	6	162

The values in this chart are calculated from the basic formula:

$$V = 66\sqrt{h}$$

Figures above the line in the pressure column indicate ounces per square inch and those below the line indicate inches of water column.

The velocity indicated is in feet per second. Multiply feet per second by 60 to calculate feet per minute for volume flow calculations.

The values in this chart are calculated from the formula: $V - 66/h$. Figures above the line in the pressure column indicate ounces per square inch, and those below the line indicate inches of water column. The velocity is in feet per second. Multiplying feet per second by 60 gives flow in feet per minute.

VOLUME FLOW CALCULATIONS

Very accurate instruments are available that can measure volume of flow directly, but they are generally too expensive for the home machinist. Fortunately, it is simple enough to calculate the volume of flow from pressure readings. Since velocity is easily calculated from the measured pressure, we go one step further to calculate volume of flow by multiplying the area through which the air flows by its velocity.

It is very important to realize that the apparent area through which the air flows may not be the actual area upon which to base the calculations. Friction causes turbulence within a duct that tends to restrict flow near the duct surface, and causes the velocity near the center to increase.

A high reading taken near the center of a duct and applied to the total cross-sectional area as measured would certainly render a false volume calculation. This high central reading is just not representative of air flow in every point in the duct.

The same principle applies to lower readings taken near the duct surface. Here again, the low reading is not representative of air flow in every point in the duct.

Even an average of readings in a section of duct might be misleading unless all modifying factors are taken into consideration.

Because losses due to friction and turbulence can be significant in even short lengths of duct, a simpler, more direct means of measurement and calculation is desireable.

The simplest way to determine volume of flow is to measure velocity pressure as air flows through an orifice.

Find the corresponding velocity in the chart, or calculate it from the formula given. Multiply the volume in feet per second by 60 to find feet per minute. Finally, multiply the cross-sectional area of the orifice in square feet by the velocity in feet per minute to get the APPARENT volume of flow in cubic feet per minute.

A modifying factor, the coefficient of discharge, of from .60 to .93 determined by the type of orifice used must be introduced into the calculations. A sharp-edged orifice in a thin plate is the easiest to make and use. Its factor of .60 is the most reliable of all. It is necessary only to cut a very accurate hole in a piece of sheet metal or durable material with very sharp clean edges.

A 4" round orifice has a total area of 12.5664 square inches which is .087 square feet.

$$\text{area (sq in)} = \text{diameter} \times \text{diameter} \times .7854$$

$$= 4 \times 4 \times .7854$$

$$= 12.5664 \text{ square inches}$$

$$\text{area (sq ft.)} = \frac{\text{area (sq in)}}{\text{144 sq in per sq ft}}$$

$$= \frac{12.5664}{144}$$

$$= .0872527 \text{ -or- } .087 \text{ approx}$$

$$\text{actual area} = \text{measured area} \times \text{factor}$$

$$= .087 \times 60 \text{ (thin plate orifice)}$$

$$= .0422 \text{ sq ft}$$

If the velocity pressure at the orifice is 3" W.C. then the velocity is 114 feet per second or 6480 feet per minute.

$$
\begin{aligned}
\text{velocity} &= 66 \times \sqrt{\text{pressure.}} \\
&= 66 \times \sqrt{3} \\
&= 66 \times 1.732 \\
&= 114 \text{ feet per second} \\
&= 114 \times 60 \\
&= 6840 \text{ feet per minute}
\end{aligned}
$$

The volume of flow is then 355.68 cubic feet per minute.

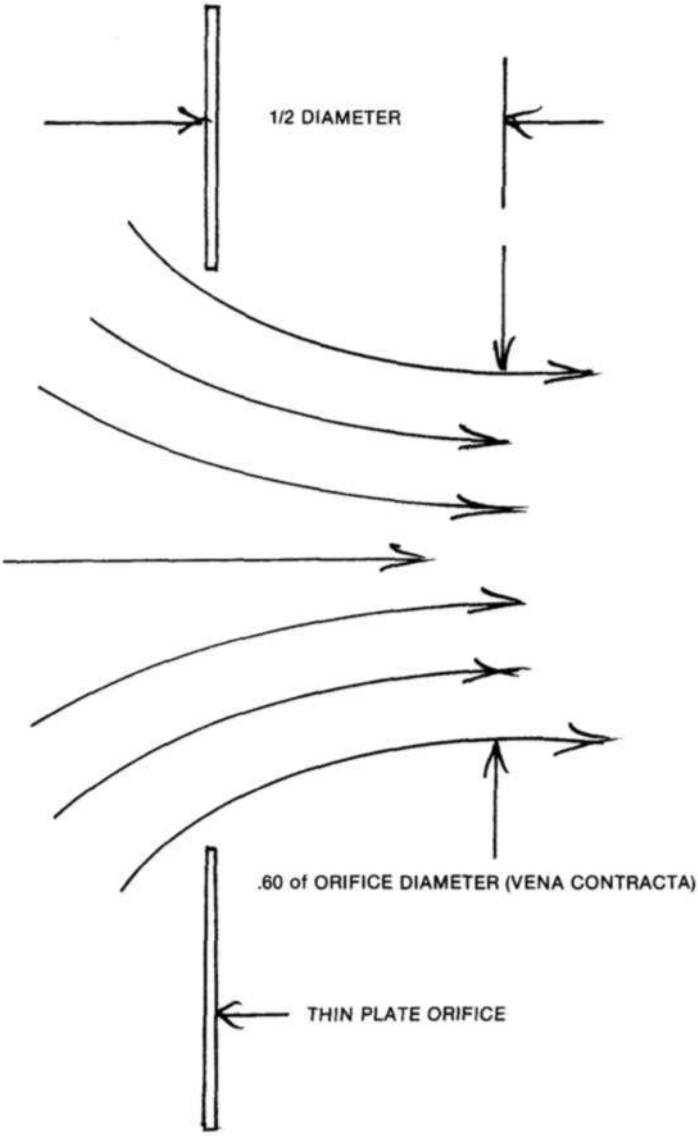
$$
\begin{aligned}
\text{volume} &= \text{area} \times \text{velocity} \\
&= .052 \times 6840 \\
&= 355.68 \text{ cubic feet per minute}
\end{aligned}
$$

AIR FLOW IN ORIFICES

Air flow through an orifice results in the "vena contracta" effect. Fluid flowing through an orifice of some cross-sectional area will contract into a stream just beyond the orifice having a cross-sectional area less than that of the orifice. This is the result of moving air hitting still air and creating turbulence.

Although the effect is not visible in clear air, it can be observed by introducing smoke into the air stream, thereby making air flow visible. The phenomenon is sometimes clearly visible in a stream as water flows between two rocks.

It has been established that the location of the smallest cross-sectional area of the contracted air stream is located approximately 1/2 orifice diameter down stream from the orifice, and that it is .60 of the diameter of the orifice. Remember, these numbers apply to thin plate orifices only. The vena contracta of a 3" orifice, for instance, will be 1-1/2" away from the orifice, and will be 1.8" in diameter.



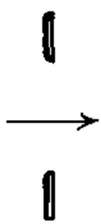
A distinct advantage of the thin plate sharp edge orifice is that it introduces practically no friction. The only modifying factors needed when using it are those resulting from the vena contracta effect.

There are several other useful orifices. The most interesting is the long radius or "well formed" orifice which is designed to conform to the natural shape of the vena contracta. Its coefficient of discharge is as high as .93, meaning there is less loss than the thin plate variety. It has much value in some laboratory work.

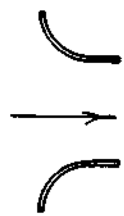
When the leading edge of the thin plate orifice is rounded, the coefficient of discharge increases to as high as .65 to .75.

Other types and forms vary in the coefficient of discharge and each has some feature that makes it appropriate to some particular use. A few are shown below.

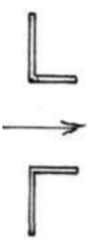
SHARP EDGE ORIFICE



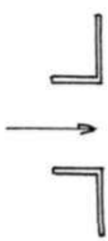
ROUND EDGE ORIFICE



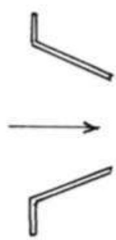
LONG RADIUS WELL FORMED ORIFICE



SHORT PIPE NOZZLE



REENTRANT NOZZLE

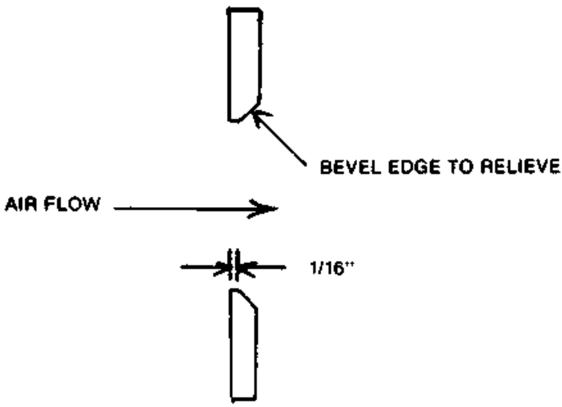


CONVERGING NOZZLE

The most important considerations in fabricating an orifice are accuracy in dimension and shape, and very smooth surfaces.

The sharp edge orifice must have very sharp, clean edges. Ideally the work would be done in a lathe, drill press or milling machine so that the orifice will be truly round. In many shops it will be necessary to use an electric hand drill with a hole saw, a hand brace with a fly cutter, or possibly sheet metal snips.

Other materials besides sheet metal can be used. The work can be done with a router or other wood working equipment. Plastic or compostion board can be similarly worked. If the material is thicker than 1/16", the downstream side should be relieved so that the air stream passes through a thickness of 1/16" or less.



CALCULATING ORIFICE AREA

The purpose of the orifice is to cause the total output of the fan to flow uniformly through a specific area so that its volume of flow can be determined. Given the air speed, we can easily calculate the volume of flow if the area is known.

In very large systems several orifices may be used and their totals combined. The idea is to reduce the discharge area which increases velocity and insures uniform flow. In a large duct or pipe, the air will flow faster near the center while moving more slowly near the walls of the duct due to friction and turbulence. There are also areas near elbows, tees and other obstacles where turbulence or friction may affect flow.

The air flow can be very fast in one portion of a duct but be very nearly still in another. Readings in one of these unusual areas would be useless since they do not indicate what is happening in the system overall. It is easy to check the entire area of the orifice for uniformity of flow. When uniformity of flow is assured, any reading will be typical of the entire area.

A sample problem will illustrate the use of an orifice. To find the actual apparent area of a circle, the square of the diameter is multiplied by .7854. This area is then multiplied by the orifice coefficient of discharge. For example a 4" orifice has an area of 12.5664 square inches.

area = 4 x 4 x .7854

= 12.5664

vena contracta area = area x coefficient of discharge

= 12.5664 x .60

= 7.53984 or approx 7.54

$$\text{area (sq ft)} = \frac{\text{area (sq in)}}{144}$$

$$= \frac{7.54}{144}$$

$$= .052 \text{ sq feet}$$

If the velocity is 8000 feet per minute then the volume of flow is 416 cubic feet per minute.

volume = area x velocity

= .052 sq ft x 8000 ft per min

= 416 ft per min

CHAPTER V

Shop Fan Applications

COMMON APPLICATIONS

With the understanding of basic principles and with the ability to test and evaluate a fan, it is possible to design and build a fan for a specific purpose and to determine whether it is performing as required.

It has already been pointed out that the common squirrel cage fan is of little use in the shop other than for heating/cooling or light-duty ventilating or exhausting. These are low pressure jobs that do not require special fan systems.

For applications such as fume exhausting, dust collection and forced draft for combustion fans must be carefully designed and built. Fortunately, it is not necessary to have a degree in engineering to build a fan system. Application of the rudimentary principles we have explored will generally produce a system adequate for the small shop.

Often a single fan is installed in some permanent location in the shop. Attached to the suction side of the fan is a trunk line with many smaller lines branching off to various locations to pick up dust and fumes. Valves, such as blast gates, can be used to open and close branch lines as they are needed.

A system is "balanced" by adjusting flow in various sections to correct or improve performance. Baffles, gates, or orifices may be needed in branches or main ducts to increase velocity in one area or reduce it in another. The object is to use the total volume of air efficiently. It may be necessary to increase the speed of the fan to raise total system velocity, or to reduce the fan speed to relieve motor overload. Observation and common sense are often the only skills needed to balance a system.

Many dust collector systems are overloaded with branch lines. Velocity in the branches is too low. It is usually possible to bring the system to good operating balance by merely installing stops in each branch so that unused ones can be shut off. A stop, or blast gate, is a simple rectangle of sheet metal slid into a pocket installed in the branch duct to block the blast of air.

If too many gates are shut off, the main duct air velocity will drop just as if too many gates are open. Material will precipitate out of the air stream and pile up in the duct.

A properly balanced system will have sufficient velocity in all areas to carry any material picked up all the way through the system, using a minimum of power and without overloading the motor.

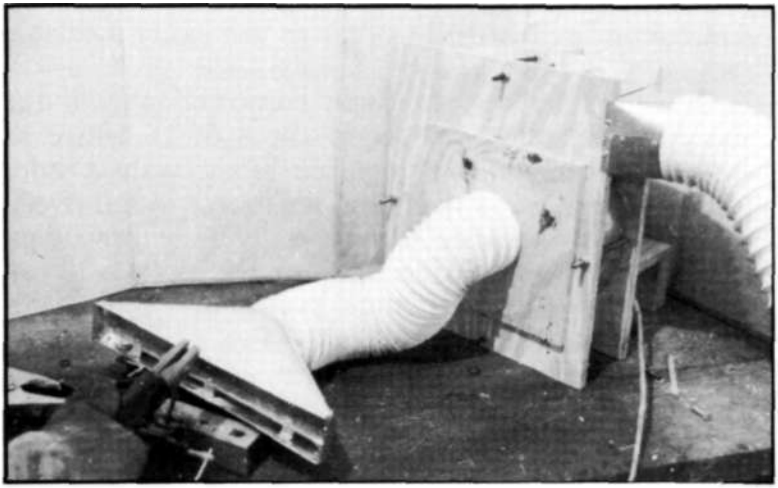
FUME EXHAUST FANS

The removal of obnoxious and often dangerous fumes and vapors is frequently neglected in small shops—especially in private shops where no regulations can be enforced to require protection of workers. Fumes and vapors from welding equipment, paint, chemicals and other processes can be injurious to the health even in small doses. Unfortunately, the ill effects are often not realized until after long exposure when it is usually not possible to restore the damaged respiratory or circulatory systems.

You should carefully consider installing an exhaust system in your shop. Built around a fractional horsepower motor you may already have on hand, an exhaust system is as easy to build as it is wise to install.

The right place to start is by identifying dangerous conditions in your shop. Most often, an easily-moved portable unit works best in a small general shop. It is easier and less costly to whisk away fumes at their source than to replace all of the air in the shop area.

A 9" x 3" portable fan is shown exhausting the fumes of a welding job through flexible hoses. Built mostly of plywood, light gauge sheet metal and standard hardware, it cost little to build.



The fan rotor is driven directly by the 1/4 hp motor shaft at 1725 rpm. Capable of moving 200 cubic feet of air per minute at 1" W.C. pressure, it efficiently carries away the welding fumes before they fill the shop and become a hazard.

Although domestic clothes dryer hose is not as durable as the hose that should be used for this type of work, it is inexpensive and available almost anywhere. Dryer hose can be used if a metal fume collector and a short length of metal pipe are used to protect the hose from heat and sparks.

The dimensions given can be modified to some extent to accommodate the equipment you have on hand. Although 3/8" thick plywood might be used for the housing, heavier 1/2" is much stronger, while 5/8" or 3/4" is preferred. For the scroll forms 1/8" masonite or other composition board is adequate. The wheel should be made of 26 gauge or heavier sheet metal, but the shell can be as light as 30 gauge flashing material. A 2" die-cast pulley was used for the wheel hub. Steel blind rivets were used to assemble the wheel.

The double plate design is used to provide reinforcement and extra strength to those wheel components fabricated from relatively light material. Although the inlet and outlet can be increased, if desired, to 6" diameter or more for greater volume at lower pressure, I used a 4" diameter on my system in order to use easily available dryer hose.

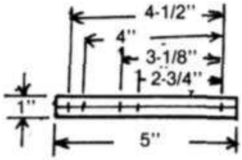
It would be best to begin construction with the rotor. Two 9" discs and eight vanes of 26 gauge or heavier sheet metal are required along with a hub, machine screws, blind-rivets and the tool to install rivets.

Layout of the wheel is most easily done directly on the metal with wing dividers or trammel points. If you don't have dividers or trammel points, you can improvise with a narrow strip of light gauge metal. Simply scribe a sharp line on the metal strip, and carefully measure out the radius of the circles to be marked. Punch a small hole at each end of this radius. By inserting an awl through each hole, you have, in effect, built a make-shift trammel.

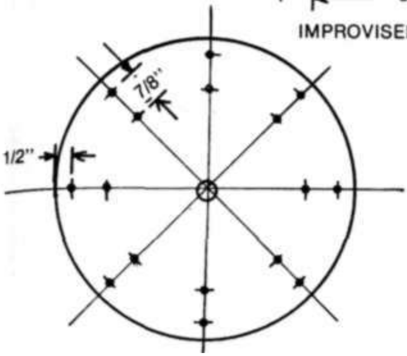
Scribe a horizontal line longer than the diameter of the circle, and intersect it with a vertical line to form a cross. The working center is at the intersection. Use the dividers or trammel to scribe the circles from the working center. Lay out the back disc and the front disc, and cut them out. Clamp them together to drill the rivet holes. Mark the discs so that they will be assembled in the same relative positions as drilled. Eighth inch steel rivets are adequate.

Although it is possible to produce a small, hand-made fan rotor that is very nearly balanced, it should be tested on the ways and precisely balanced. Even a slight variation will damage the fan over a period of time. Besides, an out-of-balance wheel can be dangerous.

Next, build the scroll. Eighth inch masonite works

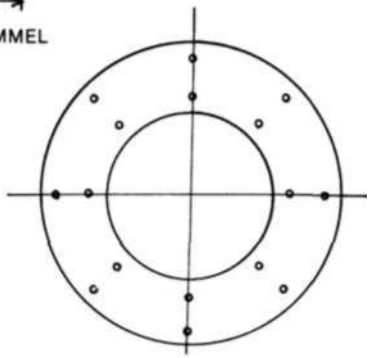


IMPROVISED TRAMMEL



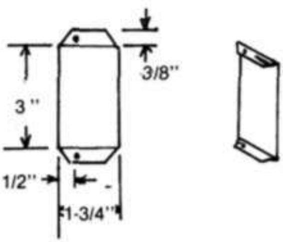
BACK DISK

9" DIAMETER WITH CENTER HOLE
TO FIT THE MOTOR SHAFT



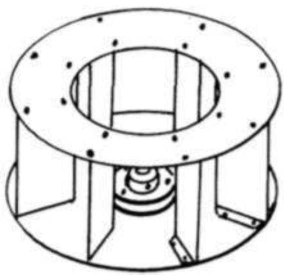
FRONT DISK

9" DIAMETER WITH 5-1/2"
CENTER HOLE



VANES

MAKE EIGHT



INSTALL HUB WITH SCREWS

ASSEMBLE ROTOR WITH RIVETS

very well. Two scrolls are needed. The radius centers are shifted 1" on each side of the wheel center, which is nearly the required 10% of the wheel diameter.

With a wheel width of 3", the blast area is 9 square inches. A housing width of 3-3/4" provides 3/8" clearance on each side of the wheel. Since the outlet area should equal 1-1/2 times the blast area, we calculate a blast area of 13-1/2 square inches.

outlet area = 1.5 x blast area

= 13.5 square inches

If the discharge opening is 3-3/4" wide, the height is calculated to be about 3-1/2".

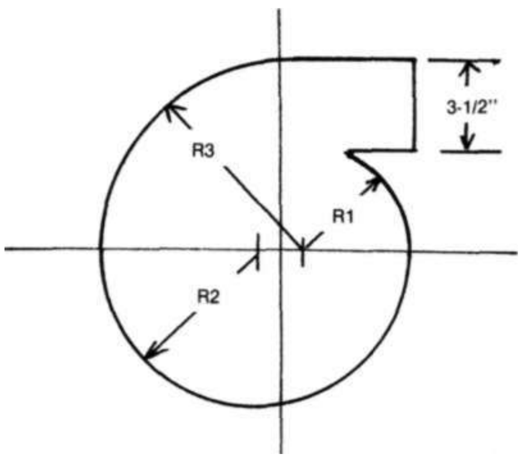
area = height × width

-OR-

$$\text{height} = \frac{\text{area}}{\text{width}}$$

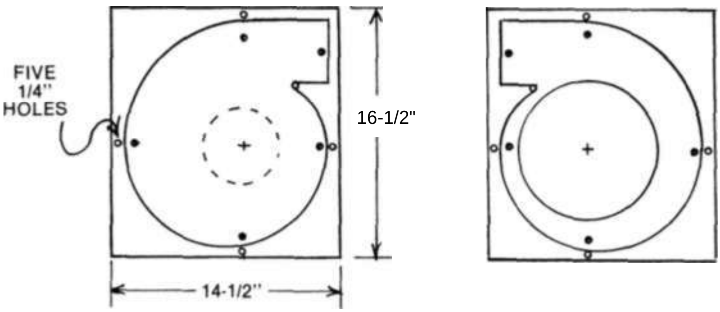
$$= \frac{13.5}{3.75}$$

= **3.6 inches (about 3-1/2")**

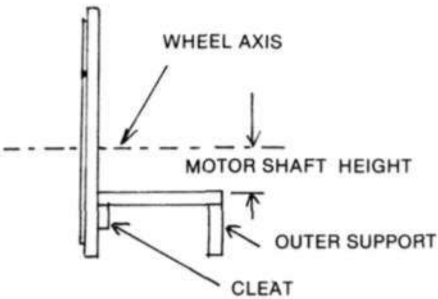


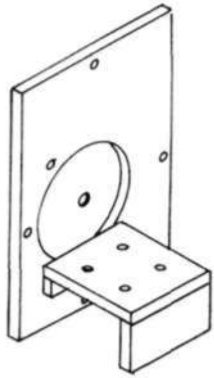
R-1 on the layout is the same radius as the wheel, or 4-1/2". *R-2* is 2" greater than *R-1*, or 6-1/2". *R-3* is 2" greater than *R-2*, or 8-1/2". A piece of material 13" wide and 15" tall will be required for each scroll. Only the outline is cut at this time. The fan wheel center holes will be in the same location but of different diameters in each scroll.

The support panels should be built of at least 1/2" thick plywood. They are 1-1/2" greater in height and width than the scrolls. Two 14-1/2" x 16-1/2" panels are required. Carefully center the scrolls on the support panels both vertically and horizontally so that the assembly will rest squarely on a flat surface when complete. Fasten the scrolls to the support panels with five small screws near the outer edge. Drill five 1/4" holes in each panel as shown for the carriage bolts that will join the two halves together. In this way two opposing scrolls are made.



One panel will be the rear panel, upon which the motor support is fastened. The other panel will be the front panel, through which the wheel is installed. The choice is determined by the direction of motor rotation so as to give the correct direction of discharge. A circle slightly larger than the wheel diameter is cut through the front panel using the wheel axis as center. Cut through both the panel and the scroll. A hole large enough for the motor shaft is drilled through the rear panel. If the motor shaft is not long enough to reach through both panels and the wheel hub, a clearance hole large enough for the motor housing is cut in the rear support panel only, leaving the scroll intact except for the shaft hole. A 7" diameter clearance hole is large enough for most fractional horsepower motors.



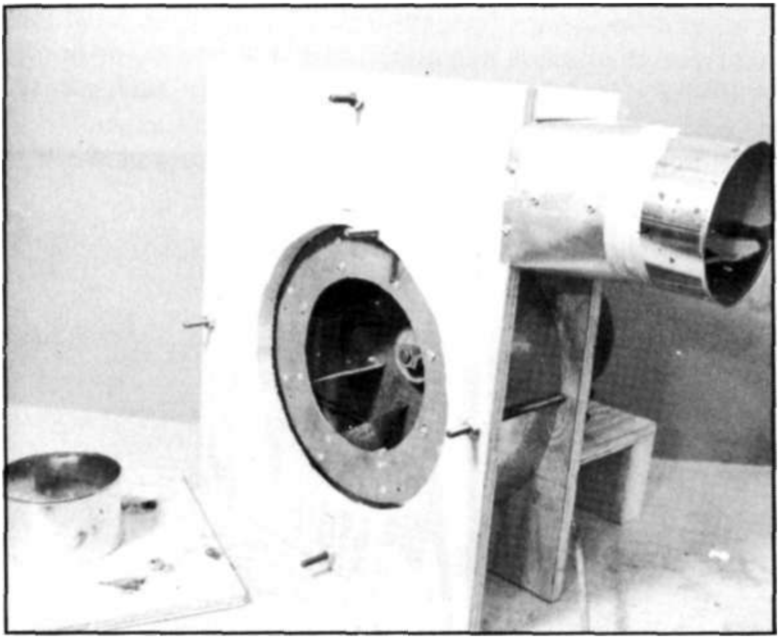


The size and location of the motor support shelf must be designed specifically for the motor to be used. The shaft height is measured from the center of the shaft to the base and is transferred to the rear support panel to locate the motor bracket. A rectangle of plywood large enough for the motor base can be fastened to a solid wood cleat on the housing. An outer support should be added as shown to support the motor weight.

The scroll shell can be light galvanized sheet metal such as is used for roof flashing. It is available in coils of various lengths and widths from hardware and building supply stores. A 4" width can be used for this unit without trimming. The two scrolls take up 1/4" of the space between the supports leaving 3-3/4". A four foot length of flashing will be more than enough.

Five 1/4" x 6" carriage bolts join the two support panels. Assemble them loosely so that the shell can easily be slipped in between the panels. Begin by bending one end of the shell neatly over a 1/4" bolt to form the cut-off end of the shell. Hook it over the carriage bolt that passes through the lower part of the discharge. Study the photographs below.

Snug up the nut on the bolt enough to hold the shell in position, and thread the remainder of the shell between the rest of the bolts and the scroll form. Pull the shell up tight before you tighten the nuts. Trim any excess metal from ends of the shell. The adapter or transition piece is fastened to the discharge opening with rivets or screws.



The inlet fitting can be a sheet metal sleeve fastened to a plywood cover about 10" square. I made both the inlet and outlet on my fan 4" in diameter merely for convenience. The inlet could be as large as 6" in diameter for greater volume of air at lower velocity and pressure.

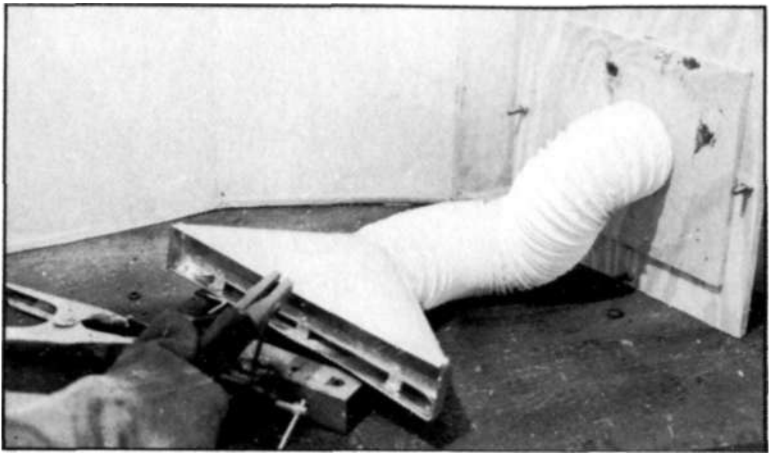
When fully loaded, this fan will deliver about 200 cubic feet of air per minute at a pressure of slightly more than 1" W.C. Since a 4" diameter hose has an area of .087 square feet, the calculated air velocity is about 2300 feet per minute in the hose.

$$\text{velocity} = \frac{\text{volume per min}}{\text{duct area}}$$

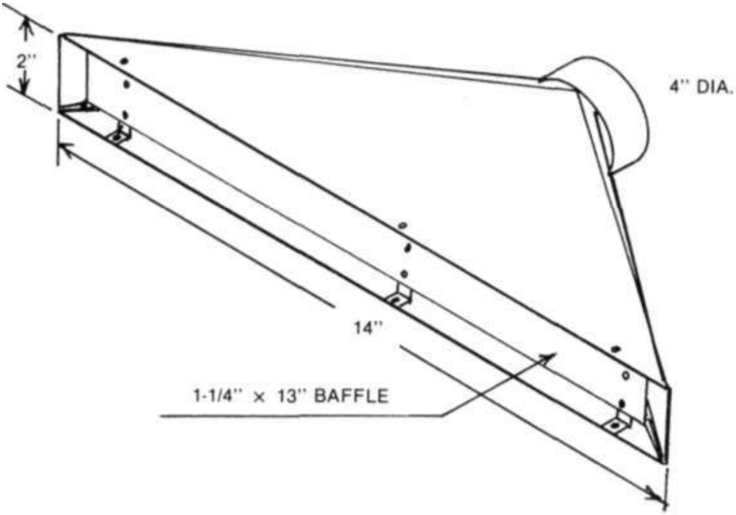
$$= \frac{\text{200 cu ft per min}}{\text{.087 sq ft}}$$

$$= \text{2298.8 feet per min}$$

If you use a 6" diameter pipe which has a cross-sectional area of .196 square feet, the velocity would be 1020 feet per minute which is adequate for a fume collector hood. A 6" inlet could also feed two 4" hoses. In such a case, of course, the discharge should be enlarged as well.



The collector hood can be made broad and narrow to establish a good draft over a wide area. Keep in mind that the air velocity drops off very rapidly as the distance from the hood increases. From 3" to 4" is about the maximum distance allowable. When the opening of the hood is greater in area than the pipe that supplies it a baffle is added so that the air velocity is maintained uniformly over the entire area.



The collector can be of any practical shape to suit the job at hand. The one illustrated works very well for all of my welding needs when placed within 3" to 4" of the work.

DUST COLLECTORS

When materials such as sawdust or grinding and polishing dust are to be carried away, the fan must be sturdier and more powerful than when only air with fumes or vapor is moved. A fan constructed very much like the one just described but of heavier material could certainly be used, but welded 16 gauge steel or heavier is a far better method. Of course, the shaft and bearings must be heavier as well.

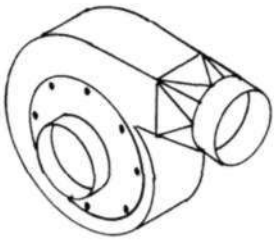
A substantial amount of air can be moved with a 1/4 hp or 1/3 hp motor, but most dust collection applications will require higher air velocities. A motor from 1/2 hp to 1 hp will be required. It is interesting to note that motor horsepower increases as the cube of the increase in fan rotational speed. In other words, if you double the speed of any centrifugal fan, you will double its volume and pressure, but the power requirement will be increased by a factor of eight.

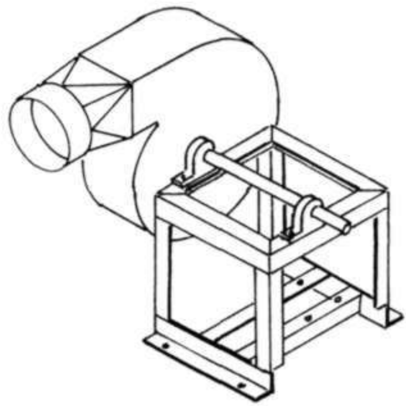
$$2 \times 2 \times 2 = 8$$

Tripling shaft rotation requires an increase of power equal to 3 to the third power or 27.

$$3 \times 3 \times 3 = 27$$

The sketch below shows a welded unit mounted upon an angle iron frame so that it can be belt-driven. This is a desireable design for a dust collector since the fan speed will almost surely have to be adjusted in order to balance the system initially and for possible later addition of branch lines.





Note that the scroll housing is made with an opening on the inlet side large enough to admit the wheel. The inlet adapter also serves as the cover. A gasket can be used to make the cover airtight. The discharge transition can be of lighter gauge metal than the housing. The angle iron frame can be of any convenient dimensions so as to accommodate both the fan and the motor. The upper rectangular frame supports two self-aligning pillow blocks, preferably ball bearings. The cross rails in the lower frame are located so that the motor pulley will fall in line with the fan pulley. The four upright spacers of angle iron are long enough to provide clearance for the motor.

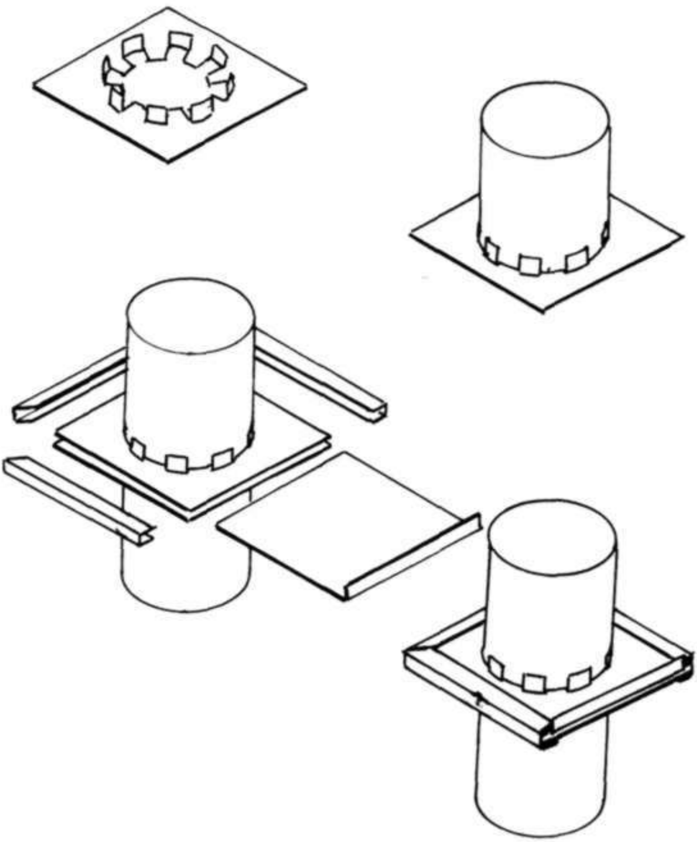
Air velocity in dust collector systems is maintained at 3500 feet per minute and above. The primary goal is to keep the air moving fast enough so that the dust will not settle out and pile up in the system. It is a simple matter to determine the total volume of air required from the area of the suction main since area times velocity equals volume. The suction main is sized to equal the total area of the branches plus 20%. There are losses within any system due to friction, turbulence, and especially to the entry loss at each collector. Although there are formulas, factors, tables and charts to aid in calculating such losses, it requires condiderable study to understand them and apply them effectively. For a small and simple shop it may be simpler to provide an excess of capacity of up to 50% in a belt-driven fan and reduce or increase the fan speed to balance the system.

A one horsepower motor will run a 12" x 4" fan at up to 2600 rpm, moving more than 500 cubic feet of air per minute at above 5" W.C. This is more than adequate for most small shop systems.

Even a smaller 1/2 hp or 3/4 hp motor will do an amazing amount of work. Unless the main and branch lines are very long, it is likely that such a system would run with a static pressure loss of 3" W.C. or less. A 6" diameter suction main could feed up to three branch lines effectively.

Additional branches can be installed in the system if blast gates are provided to prevent air from entering an unused collector and from lowering the suction of the system. The blast gate stop is removed from the collector while in use and slipped into the gate of an idle collector to balance the system. Of course, to open all of the gates would reduce the velocity in some of the branches. To close all of the stops would reduce velocity in the main and dust would settle out and plug up the system. In a small home shop worked by only one or two persons a very complete system can be operated with little power. Some machines, such as a wood lathe or a jointer, may require the full capacity of the fan. You can close other branches while they are in use. An ordinary damper will not serve very well in a dust collector system because of the high velocity and pressure.

Blast gates are easy to make. Two squares of sheet metal about 1" larger than the branch pipe diameter are prepared with a hole in the center about 3/4" smaller in diameter than the pipe. The hole is notched all around to the full diameter of the pipe to form tabs. Each alternate tab is bent vertically to form a socket for the pipe. The pipe is slipped into the socket, and two or more sheet metal screws are installed to hold the assembly secure while the remaining tabs are bent inside the pipe. The joint is soldered to make it airtight. The screws are removed to leave the inside surface smooth so that lint and shavings won't accumulate to stop up the pipe. Three channels are formed to join the two halves together and they are soldered to leave a smooth sliding fit for the stop gate. The gate is merely a rectangle of sheet metal cut to fit the slot, having one end bent at right angles to form a handle.



THE CYCLONE SEPARATOR

The mysterious-looking hopper-like devices seen on factory rooftops, often towering high above industrial plants and feed mills, is called a cyclone separator—generally referred to simply as a "cyclone." These devices slow air speeds down to allow dust and dirt carried in the air stream to precipitate out.

It is not difficult to build one for your shop, and surprisingly, small ones actually work better than large ones. Although there are variations in design, the proportions shown are time-tested and work very well if the air velocity at the inlet is between 3500 and 4000 feet per minute.

A cyclone can have greater capacity than a shop vacuum cleaner. Better yet, there is no bag to resist air flow, waste power, and periodically change.

The air and dust enter at the edge of the separator causing the air to whirl like a cyclone inside the cylinder. In the center where the air whirls fastest, centrifugal force throws the dust particles outward where they lose velocity and settle out to the bottom of the hopper. The clean air escapes out the top at a greatly reduced velocity.

Although it sounds far too simple to be true, the device is very efficient. Some commercial builders have claimed 99% efficiency. When you see a cyclone with a significant amount of dust escaping at its outlet, it is an indication that the hopper is too full or that the air volume is too great. Either emptying the waste container or reducing the fan speed will solve the problem.

DEVELOPING CYCLONE PATTERNS

A brief study will show you that the cyclone separator is made up of cylinders and cones plus a short length of rectangular duct and a transition. You learned how to lay out a transition piece earlier. The remaining cyclone shapes are among the easiest sheet metal patterns to develop.

The unit shown on page 100 is a 5" cyclone. It is easy to change the dimensions proportionately. Simply divide all of the dimensions by five and multiply the result by the new inlet dimension.

Notice that the main body diameter is four times the inlet diameter, and that the top outlet is twice the inlet diameter. These are the two critical proportions.

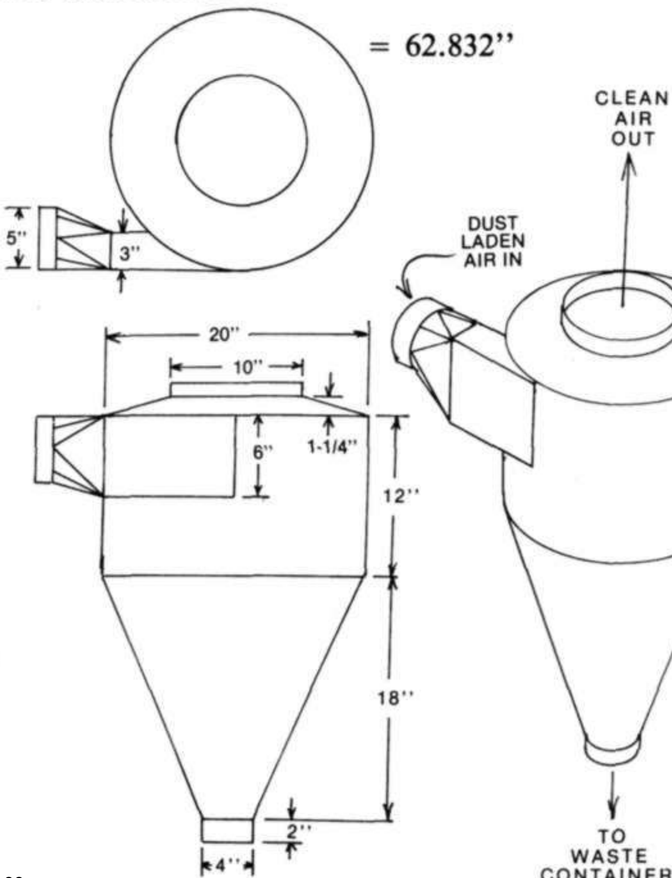
The lower cone section is as tall as its diameter. The rectangular duct section is the same cross-sectional area as the round inlet pipe.

The lower outlet size is not critical to the operation of the cyclone, but it must be large enough to permit free passage of the full capacity of the unit by gravity. No waste is stored inside the cyclone. Some sort of container must be used to accumulate the waste.

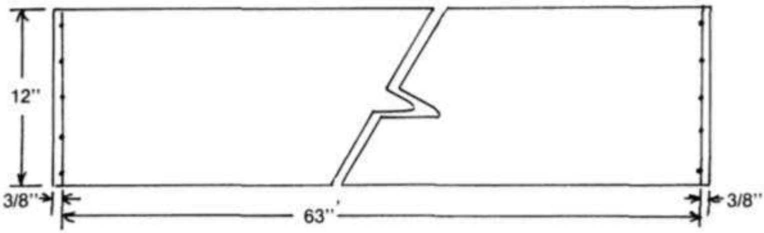
The cylindircal portion of the body is merely a rolled up sheet of metal 12" high and 63-3/4" long. The length is calculated by multiplying the diameter by pi (3.1416).

$$\begin{aligned} \text{circle circumference} &= \text{diameter} \times 3.1416 \\ &= 20'' \times 3.1416 \\ &= 62.832'' \end{aligned}$$

THE 5" CYCLONE SEPARATOR



The calculated length is rounded off to 63" for convenience and 3/8" is added to each end for a simple lap seam to be joined by rivets, sheet metal screws, or bolts and nuts. For a 5" unit, 26 gauge galvanized iron is adequate. Larger cyclones must be of heavier material.



The top and bottom sections of the cyclone are truncated cones—that is, cones with the pointed end cut off. The opening formed by this truncation is a circle. Both this circle and the circular base have their respective centers lying on the vertical line of the truncated cone.

The patterns for each cone are developed by a system of radial lines. Although the pattern development is similar to triangulation, the true lengths of the radial lines can be taken from the side elevation, or from a solution triangle that represents the side elevation.

To illustrate, we'll concern ouselves with developing a pattern for just one of the truncated cones. The remaining cones can then be developed by the same process.

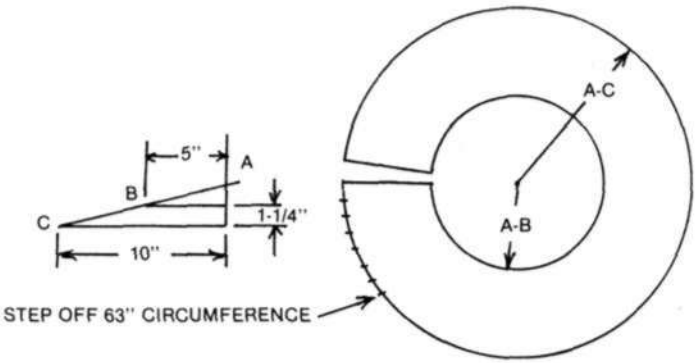
As in the transition piece pattern, a top plan view of the cone can be divided into a number of segments by radial lines. The base circle and the circle formed by truncation have common centers. They are said to be concentric, or equally centered cones. All radial lines drawn from the common center to the outer edge of the large diameter will be the same length. Once the true length of this single radial line is found, there is no further need for a top plan view. In fact, for so simple a form, it is not necessary to draw a top plan view.

A single solution triangle that represents a side of the cone as it is seen in cross-sectional view provides all dimensions. Two arcs are drawn using the common center to represent the outside edge of the base and inside edge of the top opening. The radius for these arcs is found in the solution triangle.

Only two radial lines are drawn in the actual layout as seen in the illustration. These lines represent the ends of the pattern that will be pulled together and fastened to form the truncated cone.

First, draw a full-size solution triangle as shown. The overall diameter of the cyclone is 20". Draw the horizontal base of the solution triangle 10" long, this being one half of the 20" diameter. Obviously, this 10" is the distance from the outside edge of the cyclone to the vertical center axis.

Now, draw the vertical edge of the solution triangle about 2-1/2" tall. You will note from the side view of the cyclone that the top opening of the top cone is a circle 10" in diameter. Also note that this opening is 1-1/4" above the base of the cone.

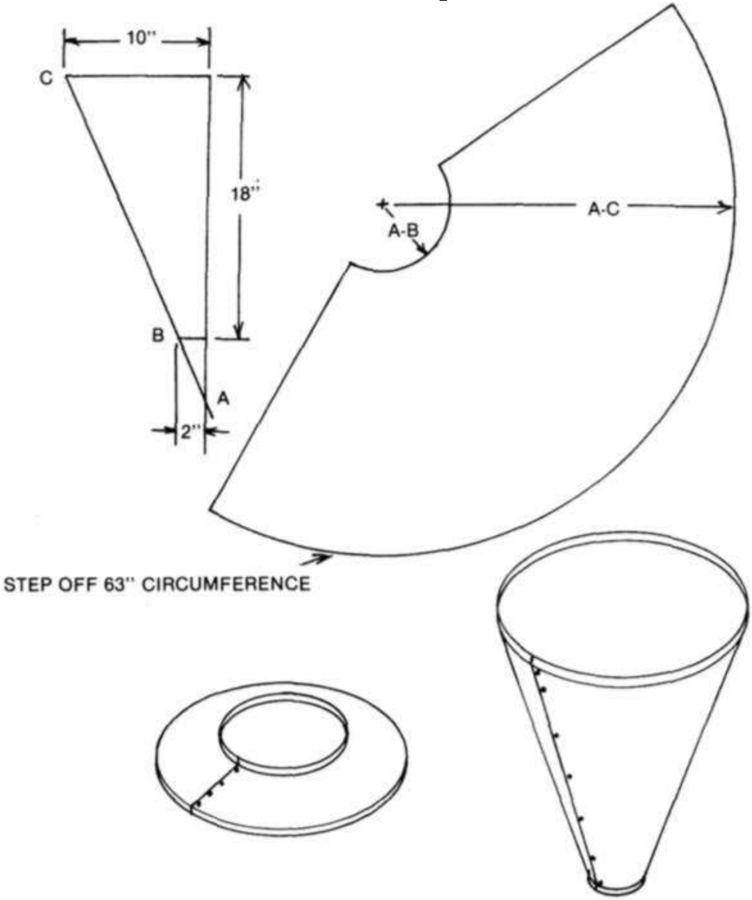


Extend a second horizontal line 5" from the vertical center axis at a point 1-1/4" above the base line to represent one half of the 10" top opening.

Draw a diagonal line upward from point C at the base, touching point *B* at the opening height until it intersects the vertical center line at point *A*. This is the solution triangle.

The distance from *A* to *B* on the solution triangle is the true radius of the arc that forms the opening at the top of the cone, or in other words, the radius of the inside circle of the concentric circles found in the pattern stretchout. The distance from *A* to C is the true radius of the arc that forms the base of the cone, or outside circle.

The calculated circumference of the second cone base is the same 63" from the cyclindrical portion. It should be obvious that if we want these two pieces to mate properly, they'll have to have the same circumference. Using the radii from the solution triangle, draw two concentric circles. Draw one radial line through both circles to create one end of the pattern.



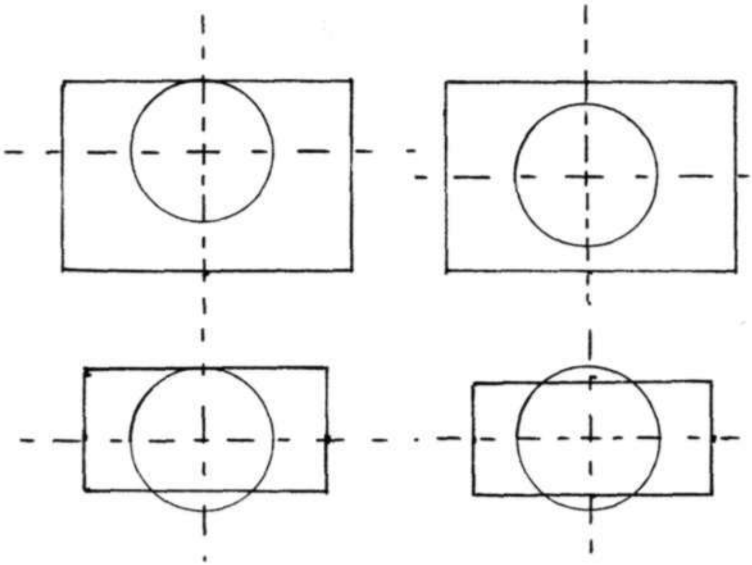
Next, step off 63" of length on the outer circle with dividers to find the other end of the pattern. Draw a second radial line through this 63" point to complete the layout. In this way we have developed a pattern for a cone having a circumference of 63" that will match up with 63" circumference of the cylindrical form developed earlier.

An extra amount of metal is added to each end to form a flap or tab for making a lap seam. For a small, light-gauge unit, 3/8" is sufficient. A similar amount is added at the inner and outer diameters for the joints.

Of course, the cones can be made in two or more pieces. Simply add an amount for the seam wherever laps are made.

The pattern for the bottom cone is developed in exactly the same manner.

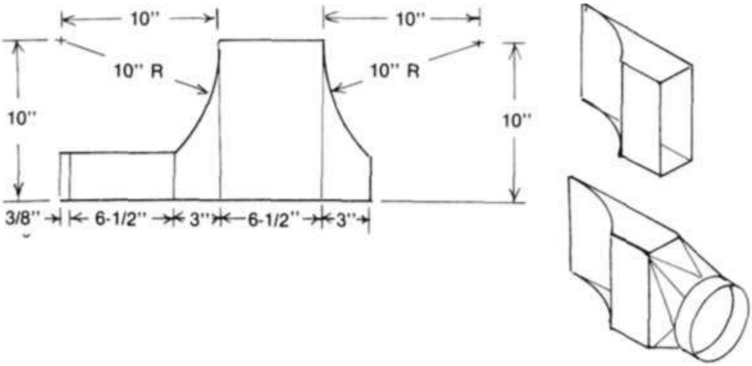
The inlet transition differs only slightly from those discussed earlier. Rather than being equally centered over the rectangular base, the circular portion has two of its sides parallel with two sides of the rectangular portion.



Not Equally Centered   Equally Centered

The layout procedure is almost the same as that for a single-plane-rectangular-to-round transition piece. In this case, however, there are no duplicate dimensions as found in a pattern for an equally centered transition piece.

The rectangular portion of the inlet duct must have the same cross-sectional area as the round portion of the inlet so that there is no change in velocity through the inlet duct. The rectangular portion of the inlet duct is a simple short rectangular duct with one end cut as shown to conform to the curvature of the cyclone body so that air enters at the edge of the cyclone.

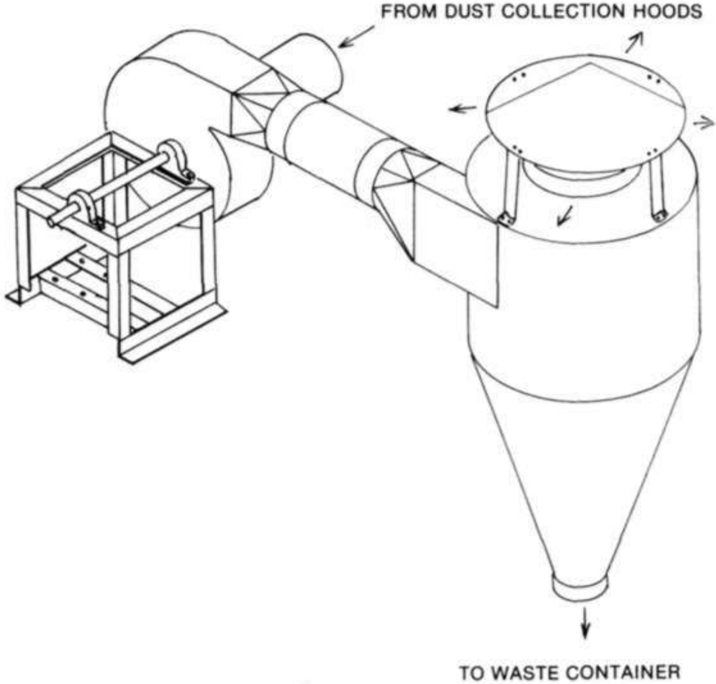


Be sure to add a flap or tab to the rectangular portion of the inlet transition duct so that it can easily be joined to the cyclone body. A similar flap should be added to the edge of the rectangular end of the transition duct so that it can be joined to the duct from fan outlet.

All of the parts of the cyclone can be assembled with sheet metal screws, bolts, or rivets. I recommend you solder all seams to make them airtight.

If the cyclone is to be exposed to the weather, a conical hood should be installed at a height over the cyclone discharge opening equal to half the diameter of the opening. This hood should have a diameter of about one third greater than that of the discharge opening to keep out the weather.

A collar can be fastened to the lower outlet so that the cyclone can be connected to a bin or other waste con-



tainer. If any part of the discharge air from the cyclone is returned to the work area, it must be filtered to remove fine particles of dust that could be hazardous to the health, particularly if the air contains silica such as from grinding and sanding machines.

BALANCING THE SYSTEM

Too much air will not only throw dust out of the cyclone discharge, but moving more air than necessary wastes power. On the other hand, with too little air, waste will clog the system.

Using the manometer, you should check the system to see that the fan is running fast enough to provide sufficient air velocity at the collectors. The pitot tube can be used in any of the inlet branches or in the main duct where the branches join up before entering the fan inlet.

Be sure that no dust is being collected by the system when you test it, or you stand the chance of plugging up the pitot.

The simplest way to take the needed measurements is to drill a 3/4" hole to admit the pitot tube in the branch pipe very near the point where it joins the collector hood.

Although it is difficult to specify an exact ratio for every inlet type of hood, it is reasonable to assume that a static pressure reading in the inlet hood should be approximately 1-1/2 times the actual velocity pressure in the hood. If, for example, a static pressure reading near the hood is 1-1/2" W.C., the actual velocity pressure should be approximately 1" W.C., which indicates a velocity of 3966 feet per minute. In this case, you could reduce the fan speed slightly if desired. To do so is not all that important since the excess velocity is only about 10% greater than the minimum required. A reading of 2" would be too great, indicating a reduction in fan speed.

Use in the shop will be the most reliable test of all. If all of the hoods are collecting dust, and if waste is not piling up in the suction main or flying out the cyclone discharge, you have a good system.

Many rules-of-thumb have been developed over the years. Here are a few that you may find useful.

The fan discharge pipe should never be smaller in cross-sectional area than the suction pipe, but never so large that velocity is reduced below system requirements.

Elbows in the system should have long turning radii, and should be as few in number as possible. A long radius elbow offers as much resistance to the flow of air as a length of straight pipe 10 diameters long. Resistance in a short radius elbow is much greater.

Branches should enter the suction main at an angle of 45 degrees. The suction main should be increased proportionately as each one or two branches are added the greatest diameter being at the fan intake. This need not be observed if only one branch is to be used at a time, with the others being shut off with blast gates. Be sure the suction main is not so large at the inlet that the air velocity is reduced.

All of the joints in the piping system should have smooth inside surfaces. Every joint should be soldered, or should be sealed with tape or caulking to eliminate leaks. System efficiency can be greatly reduced by many small leaks.

Safety first! Employ safety precautions wherever possible. Belt drives most certainly must be guarded. All electrical work must be done properly to eliminate overloads, shock hazards and fire hazards.

Within these few pages it is impossible to fully explore any single aspect of fan technology. You have learned the basics, though. You can design and build a fan for a particular application and have a clear idea of what sort of performance to expect. Commercially-built exhaust fans can cost several hundred dollars if purchased new. But with little more than basic hand tools, you can build a custom fan for a fraction of the new cost. Quite often, most of the materials needed are already on hand or easily available as surplus or salvage.

APPENDIX

Some of the most frequently used formulas and tables are repeated in this section for quick reference.

COMPOSITION OF AIR	By Weight	By Volume
Oxygen	23.1%	21%
Nitrogen	76.9%	79%
	100%	100%

STANDARD AIR

temperature: 68 degrees Fahrenheit
relative humidity: 50%
pressure: 29.92 inches of mercury

WEIGHT OF STANDARD AIR:

.075 pounds per cubic foot

VELOCITY OF FLOW AND VELOCITY PRESSURE RELATIONSHIP:

$$V = 66\sqrt{h}$$

V = velocity in feet per second
h = velocity pressure in inches W.C.

The velocity in feet per second is equal to the square root of the velocity pressure in inches W.C. multiplied by 66.

Also: $\frac{V}{\sqrt{h}} = 66$ -and- $\frac{V}{66} = \sqrt{h}$

AREA OF A CIRCLE:

Multiply the square of the diameter by .7854.

CIRCUMFERENCE OF A CIRCLE:

Multiply the diameter by 3.1416 (pi).

PERIPHERAL VELOCITY OF A FAN WHEEL:

Multiply the circumference in feet by the speed in RPM to learn the velocity in feet per minute.

ONE OUNCE OF PRESSURE PER SQUARE INCH GAUGE: is equal to 1.73" W.C.

ONE INCH WATER COLUMN: is equal to .58 ounces per square inch gauge.

BLAST AREA:

$$A = \frac{D \times W}{3}$$

where: A = Blast area in square inches
D = Wheel diameter in inches
W = Wheel width in inches

The blast area is equal to the wheel diameter in inches multiplied by the wheel width in inches and the product divided by 3.

THEORETICAL VELOCITY OF AIR AT 62 DEGREES FAHRENHEIT INTO ATMOSPHERE

PRESSURE (OZ. P.S.I. / INCHES W.C.)	VELOCITY, FT. PER SEC.	PRESSURE	VELOCITY
.006 / .01	6.61	1.731 / 3	114
.012 / .02	9.35	2.020 / 3.5	124
.023 / .04	13.20	2.308 / 4	132
.040 / .07	17.40	2.579 / 4.5	140
.058 / .1	20.90	2.885 / 5	148
.115 / .2	29.5	3.462 / 6	162
.173 / .3	36.2		
.231 / .4	41.8		
.260 / .45	44.3		
.289 / .5	46.7		
.346 / .6	51.2		
.404 / .7	55.3		
.461 / .8	59.1		
.519 / .9	62.7		
.577 / 1	66.1		
.866 / 1.5	80.9		
1.153 / 2	93.5		
1.442 / 2.5	104.0		

The values in this chart are calculated from the basic formula:

$$V = 66\sqrt{h}\ .$$

Figures above the line in the pressure column indicate ounces per square inch and those below the line indicate inches of water column.

The velocity indicated is in feet per second. Multiply feet per second by 60 to calculate feet per minute for volume flow calculations.

DISCHARGE OPENING:

From 1-1/2 to 2-1/2 times the blast area.

THEORETIC CAPACITY OF FANS:

$$C = B \times V \times D$$

where: C = Capacity in cubic ft per min (CFM)
B = Blast area in square feet
V = Velocity in feet per minute
D = Coefficient of discharge (.8)

VOLUME OF FLOW IN AN ORIFICE:

Multiply the area of the orifice in square feet by the velocity of flow in feet per minute. Multiply the resulting product by the coefficient of discharge. (The coefficient of discharge for a thin plate sharp edge orifice is .60.)

POWER REQUIRED TO RUN A FAN:
Assuming 50% efficiency

$$HP = \frac{C \times h \times 5.2}{275}$$

where: HP = Horsepower required
C = volume of flow in feet per second
H = Pressure in inches W.C.