FANS AND BLOWERS PART 1
Adapted from material provided by Lau Industries, a division of Tomkins Industries, Inc.

INTRODUCTION

The words “fan” and “blower” are frequently used interchangeably. In HVAC/R terminology, the term fan may refer to any device used to create artificial currents of air. A blower is generally considered to be a particular type of fan—specifically, a motor-driven centrifugal fan that delivers air to the conditioned space under pressure.

There are two basic types of fans used in HVAC/R applications—axial and centrifugal. They are classified according to the direction of air flow through the impeller (the rotating blades that push the air). In an axial fan, air flow is parallel to the shaft. Figure 1 shows an exploded view of a typical axial fan. In a centrifugal fan, air flow is radial through the wheel, and therefore perpendicular to the shaft. Figure 2 shows an exploded view of the component parts of a centrifugal fan.

As stated previously, an axial fan gets its name from the fact that the direction of air flow is parallel to the impeller’s axis of rotation. Axial fans can handle large volumes of air at low static pressures. They operate at high speeds, and can be noisy. Axial fans are divided into three main groups—propeller, tubeaxial, and vaneaxial.

A propeller fan consists of two or more blades rotating within a ring or frame. Propeller fans are often used as exhaust fans and condenser fans. They normally are not attached to duct systems. A tubeaxial fan is simply a propeller fan mounted in a tube or cylinder. Tubeaxial fans are more efficient than propeller fans, but they create a spiral discharge pattern. A vaneaxial fan is housed in a cylinder like a tube-axial fan, but the vaneaxial fan includes stationary guide vanes designed to remove much of the “swirl” from the discharge air, thus improving the fan’s performance. Note that Figure 1 shows a vaneaxial fan. Vaneaxial fans are more efficient than tubeaxial fans, and can reach higher pressures.

In some axial fans, the blade pitch or angle can be adjusted to control the amount of air handled. The placement of the impeller along the shaft also can be critical to the performance of the equipment. When replacing the impeller of an axial fan, you must make sure not only that the replacement has the same number of blades and the same diameter as the original, but also that the blade pitch and the position at which the impeller is attached to the shaft are correct.
CENTRIFUGAL FANS

A centrifugal fan is similar to a centrifugal pump, except that it “pumps” air instead of water. As shown in Figure 2, the flow of air as it enters the inlet is parallel to the axis of the impeller shaft. Air is redirected within the impeller wheel, and is moved outward by the centrifugal force created by the rotating blades. The scroll combines the air that passes out between the blades into a single, large airstream.

Centrifugal fans are sometimes called “squirrel cage” fans because of the shape of the impeller wheel. They are widely used in HVAC/R applications to distribute conditioned air through ductwork. There are several types of centrifugal fans, categorized according to how the blades are fixed to the impeller. The three most common impeller designs are radial, backward-curved, and forward-curved.

As shown in Figure 3A, the blades of a radial centrifugal impeller extend straight outward from a central hub, parallel to the radii of the wheel. Centrifugal fans with radial blades provide high pressures at high speeds. They are sometimes used as cooling fans in electric motors.

The blades of a backward-curved centrifugal impeller are inclined in the direction opposite rotation. As shown in Figure 3B, the blades actually may be either straight or curved. Centrifugal fans with backward-curved blades are generally more efficient than those with forward-curved blades. Backward-curved blade fans rotate at about twice the speed of forward-curved blade fans, and are therefore noisier.

As shown in Figure 3C, the blades of a forward-curved centrifugal impeller are inclined in the direction of rotation. The forward-curved impeller usually has a larger number of smaller blades than either the backward-curved impeller or the radial impeller. The lower speeds of forward-curved blade fans minimize shaft and bearing size. As a result, centrifugal fans with forward-curved blades can be made smaller, lighter, and less costly. There is, however, a sacrifice in efficiency, which means increased power. Also, although fans with forward-curved blades have a wide operating range, a risk of overloading the motor exists if static pressure decreases. (Overloading is normally not a problem with backward-curved blade fans).

MIXED-FLOW FANS

As the name implies, a mixed-flow fan utilizes principles of both axial and centrifugal impeller designs. Visualize a propeller mounted in the inlet of a centrifugal impeller. (In practice, the design is less obvious.) Air flow in a mixed-flow fan is primarily parallel to the axis of rotation, but the shape of the blade introduces a radial flow component (generally small) at the discharge. The impeller may be mounted in either a cylindrical or volute housing. (A volute is the outer casing that contains the spinning rotor in a centrifugal device.)

FAN CLASSIFICATIONS

Most fans are rated according to industry standards developed by ASHRAE (American Society of Heating, Refrigerating, and Air Conditioning Engineers) and AMCA (Air Movement and Control Association). Fan manufacturers publish rating tables to show the speeds and static pressures that a particular fan is capable of producing. If the specifications exceed certain given conditions, the “class” of the fan changes. The class indicates the structural limitations of the wheels, bearings, and housing. Under the most recent AMCA standards, there are three classes of centrifugal fans. (There used to be four.) Figure 4 shows how fan classification standards are designated for single-width centrifugal fans with backward-curved blades. To qualify as meeting the requirements of a specific class (Class I, II, or III), a fan must be capable of operating safely at every point of rating on or below the “minimum performance” limit for that class. For further information, consult AMCA Standard 99-2408-69.
FACTORS DETERMINING FAN SELECTION

Often the physical arrangement of components in a piece of air conditioning equipment determines the type of fan used. Compact window air conditioners may incorporate propellers, mixed-flow impellers, and/or centrifugal impellers. Sometimes they are found with a condenser fan and an evaporator fan driven from opposite ends of a double-shafted motor. In some units, both fans are propellers. In others, both fans are forward-curved centrifugal models. In still others, there may be one of each.

The basic design of some air conditioning equipment makes it advantageous to use a fan that turns the leaving air 90° to the incoming air. In such cases, a centrifugal or mixed-flow fan is used. When it is necessary to move air straight through the unit, propeller fans work best. Sometimes a manufacturer’s design calls for evaporator condensate water to be thrown onto the condenser. A slinger ring on a propeller or mixed-flow fan can serve this purpose.

The shape of the condenser coil also can affect the choice of fan type. Large, relatively thin condenser coils work best when the air flow is evenly distributed. One large propeller fan or several small ones are often chosen to accomplish this when space is at a minimum.

One of the most important factors in selecting the proper fan for a particular application is the resistance of the system in which it operates. Restricted air passages, thick coils, and long or winding ducts all add up to greater resistance. It is this resistance that creates the static pressure (or static head) when air moves in the system. The greater the air flow rate (cfm), the greater the static pressure.

GRAPHING SYSTEM CHARACTERISTICS

Look at the graph shown in Figure 5. Note that its horizontal (x) axis represents air volume in cubic feet per minute (cfm), and its vertical (y) axis represents static pressure in inches, water gauge (in. w.g.). The characteristics of any air distribution system can be drawn as a curve by plotting known coordinates on a graph of this kind. For example, assume that the air flow and the static pressure of a given system have been measured, and are found to be 1,000 cfm at 1 in. w.g. Plot this point on the graph.

A system curve is always a parabola—that is, a curve whose y coordinate increases as the square of the x coordinate. In other words, if you wish to determine the static pressure at 2,000 cfm, you would calculate as follows: The cfm has doubled (2 x the original 1,000 cfm), and therefore the static pressure must
increase by $2^2$, or $4 \times$ the original 1 in. w.g. You can plot a second point, then, at 2,000 cfm and 4 in. w.g. At 3,000 cfm, the air volume has become $3 \times$ the original 1,000 cfm. Since $3^2 = 9$, the new static pressure is $9 \times$ the original 1 in. w.g. Plot a third point at 3,000 cfm and 9 in. w.g. Now draw a curve, starting at 0-0 and passing through these three points. You can use this curve to estimate the expected static pressure created by the system for any cfm wanted.

Every air distribution system has its own curve. Some systems have steep parabolic curves, others have curves that are more shallow. If you plotted curves for several different systems on the same graph, you would have many different lines. Each line would pass through 0-0, but otherwise the curves would never cross or touch each other.

The more obstructions there are to the flow of air, the more sharply the system curve will rise. Thus, smaller, more restrictive systems have steep curves, while larger, more “open” systems have shallow curves. In fact, if a system is completely sealed off, so that any amount of pressure can be created without producing the slightest flow, the system “parabola” becomes a straight vertical line along the left edge of the graph.

At the other extreme, imagine a desk fan blowing in an open room, with no restrictions. It can move any volume of air without producing any static pressure. The “parabola” then becomes a straight horizontal line along the bottom of the graph.

**FAN LAWS**

As a service technician, you may be called on to change the rate of air flow in an air conditioning system. Usually it is an increase that is wanted. You may be able to increase the fan speed, to move more air at an increased static pressure. You may be able to decrease the system resistance, to allow more air flow at a reduced static pressure. You may even do both.

Reducing the system resistance can mean anything from opening balancing dampers to adding branch ducts to replacing an entire duct system with a larger one. Replacing dirty filters with clean ones is another method of reducing resistance. Predicting the results of these changes is sometimes no better than an educated guess. However, increasing the overall cfm by increasing the fan speed is predictable.

Why? Because all fans operate according to a set of mathematical relationships called fan laws. These laws show how fan speed, air volume, static pressure, density, and power are related to each other. Fan speed typically is expressed in revolutions per minute (rpm), air volume in cfm, static pressure (SP) in in. w.g., density (D) in lb/ft$^3$, and power in brake horsepower (bhp). The fan laws can be written as a series of mathematical equations, as follows:

\[
\frac{\text{cfm}_2}{\text{cfm}_1} = \frac{\text{rpm}_2}{\text{rpm}_1}, \\
\frac{\text{SP}_2}{\text{SP}_1} = \left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^2, \\
\frac{\text{bhp}_2}{\text{bhp}_1} = \left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^3, \\
\frac{\text{D}_2}{\text{D}_1} = \left(\frac{\text{rpm}_2}{\text{rpm}_1}\right)^2.
\]

These equations are very helpful when you must determine the effect of changing the speed of a fan. As you can see from the first equation above, any change in air flow is directly proportional to the change in fan speed that caused it. Doubling the rpm, for example, doubles the cfm.

The second equation states that a change in static pressure is proportional to the square of the corresponding change in fan speed.
in fan speed. Using simple substitution, you can reason that, since cfm and rpm are proportional, static pressure is also proportional to the square of the cfm. You will recall that this is exactly the relationship illustrated by the parabolic system curves explained in the previous section.

Note in the third equation above that the fan’s brake horsepower increases as the cube of the fan speed increases (or as the cube of the air volume increases, by combining the first and third equations). This means that if you try to increase the air flow in a system without making any other changes, the brake horsepower (and, therefore, the energy consumption) will increase dramatically.

The density of the air does not affect the volume moved. A fan is known as a “constant-volume” device—that is, it produces the same cfm, regardless of how dense the air is. However, as you can see by combining the second and fourth equations above, the static pressure developed by the fan varies in direct proportion to the density. In other words, the heavier the air is, the greater the pressure will be.

You can use the fan laws to predict how a change in one variable will affect the others. Since volume is the variable most commonly changed in an air distribution system, let’s look at an example based on a change from an existing cfm to a new cfm. Follow these steps:

1. The first step is to calculate the ratio of the new cfm to the existing cfm, as follows:

   \[
   \text{ratio} = \frac{\text{new cfm}}{\text{existing cfm}}
   \]

2. To determine the new fan speed, multiply the ratio in Step 1 times the existing rpm:

   \[
   \text{new rpm} = \text{ratio} \times \text{existing rpm}
   \]

3. To determine the new static pressure, square the ratio in Step 1 and multiply the result times the existing SP:

   \[
   \text{new SP} = (\text{ratio})^2 \times \text{existing SP}
   \]

4. To determine the new hp required, first cube the ratio in Step 1 and then multiply the result times the existing hp:

   \[
   \text{new hp} = (\text{ratio})^3 \times \text{existing hp}
   \]

Assume that you have the following existing conditions: A fan operating at 850 rpm is driven by a 0.6-hp motor. The fan produces an air flow of 2,500 cfm and a static pressure of 0.2 in. w.g. You want to increase the air flow to 3,000 cfm. Use the four-step procedure explained above to calculate the resulting fan speed, static pressure, and horsepower.

1. Calculate the ratio of new cfm to existing cfm:

   \[
   \text{ratio} = \frac{3,000 \text{ cfm}}{2,000 \text{ cfm}} = 1.2
   \]

2. Calculate the new rpm:

   \[
   \text{new rpm} = 1.2 \times 850 \text{ rpm} = 1,020 \text{ rpm}
   \]

3. Calculate the new static pressure:

   \[
   \text{new SP} = (1.2)^2 \times 0.2 \text{ in. w.g.} = 1.44 \times 0.2 \text{ in. w.g.} = 0.29 \text{ in. w.g.}
   \]
4. Calculate the new hp:

\[
\text{new hp} = (1.2)^3 \times 0.6 \text{ hp} = 1.73 \times 0.6 \text{ hp} = 1.04 \text{ hp}
\]

The calculations show that under the new conditions, the cfm would increase 20%, the rpm also would increase 20%, the static pressure would increase 45%, and the horsepower would increase 73%.

**FAN EFFICIENCY**

The efficiency of a mechanical device is defined as the ratio of output power to input power. It is usually expressed as a percentage. A given blower or fan generally works best when used with a system that has a particular resistance. That is, the fan is more efficient with this particular system resistance than with any other. (System resistance is the sum of all the pressure losses through elbows, filters, dampers, registers, coils, etc.) To put it mathematically, the fan’s power output divided by its power input is greater for this particular system or its equivalent than for any other.

You might think that by “opening up” a system (by opening dampers, removing filters, or enlarging ducts) and thereby decreasing resistance, you could increase a fan’s efficiency. But system specifications are usually stated in terms of the volume of air (cfm) to be moved. In a well-balanced system, the fan’s capacity matches the cfm requirement at a given static pressure. Reducing the resistance reduces the static pressure. And as shown by the fan laws above, if you change the static pressure, other conditions also will be affected.

Consider, for example, an application in which the fan is well-matched to the original system specifications. It does its work efficiently, but you wish to reduce horsepower in some way without reducing air flow. If you enlarge the duct system, you will decrease the resistance, and thus the static pressure. As a result, you will have to slow down the fan in order to maintain the original cfm. Now the cfm is the same, static pressure is less, and fan speed is lower. The fan’s power output has been reduced, so the power input is also less. But if you divide the new power output by the new power input, you will probably find that the fan is less efficient in the new system. Why? Because it doesn’t match the new system as well as it did the old system. It would be possible at this point to reduce horsepower even further by replacing the original fan with one that better matches the new system resistance. The lesson to be learned from this is that the original system designer must estimate the static pressure that the fan will have to overcome when delivering the required cfm. When system specifications are calculated accurately, a fan that will work efficiently can be selected.

**FAN PERFORMANCE CURVES**

As you have seen, you must know what the capacity of a fan is before you consider opening a damper or increasing the fan speed. Doing so may not be safe—it could result in an overloaded motor, for example, or an unusually high power consumption—so you should be able to predict the results of any changes you may decide to make. Fan and blower manufacturers provide important information about their products in the form of performance curves. A performance curve is a graph that represents a fan’s volume flow rate (along the horizontal axis), plotted against either the horsepower required or the static pressure (along the vertical axis). Figure 6 shows typical performance curves for two forward-curved centrifugal fans.

![Figure 6. Typical performance curves for centrifugal fans with forward-curved blades](image)
Look at the top curve in Figure 6. It plots the performance of Fan A, operating at 1,000 rpm. Note that when the cfm is 0 (at the far left of the graph), the fan develops its maximum static pressure—in this case, a little more than 0.9 in. w.g. This is the pressure produced when the ductwork is entirely closed off. By the time the air flow has reached 500 cfm, the static pressure has dropped nearly 0.2 in. w.g. This is because registers are being opened to allow air to flow through the system. The lowered resistance means a lower static pressure and a higher volume flow rate.

Between about 500 and 1,000 cfm, the static pressure temporarily rises as the air flow continues to increase. This is known as the “unstable” portion of a fan performance curve. Then, after reaching a second high point of 0.8 in. w.g. at about 1,000 cfm, the static pressure begins to drop and continues to drop as the system gradually opens. When the static pressure reaches 0 (at the bottom of the graph), all restrictions have been removed. When there are no restrictions to air flow and no static pressure, a fan is said to be in a state of free delivery.

The bottom curve in Figure 6 plots the performance of a smaller fan rated at 800 rpm. Note that the two curves have the same shape. This is because Fan A and Fan B are geometrically similar. Fans that are geometrically similar—that is, all of their wheel dimensions have the same proportionate ratios—produce performance curves with the same shape. (Of course, the two curves also could represent the same fan operating at two different speeds.) Remember that a system’s design characteristics, as represented by its parabolic curve, are independent of the fan used in that system. If a system characteristic curve and a fan performance curve are plotted on the same graph, the two curves will intersect. The point at which they intersect is called the point of operation. The point of operation tells you the cfm and the corresponding static pressure that you can expect for that particular fan operating in that particular system.

Figure 7 shows the two fan performance curves from Figure 6 (Fan A and Fan B) plotted on the same graph with two system characteristic curves (the dashed lines labeled “System 1” and “System 2”). This produces four points of operation (the numbered circles 1, 2, 3, and 4). Ideally, the point of operation should fall between the pressure “peak” and free delivery, on the right-hand slope of the fan performance curve. Selecting a fan with a point of operation in the “unstable” portion of the curve (to the left of the pressure peak) may result in “hunting” and noisy operation. As you can see from the location of the points of operation in Figure 7 below, neither...
Fan A nor Fan B would work well in System 1. Both fans would be more suitable for operation in System 2.

Sometimes fan performance curves are plotted to show brake horsepower along the vertical axis, instead of static pressure. Figure 8 below shows brake horsepower curves for both Fan A and Fan B. Figure 9 shows the same graph with the brake horsepower curves superimposed over the original fan performance curves. Note that brake horsepower and static pressure share the same axis. This is frequently done to conserve space. Figure 9 also includes a system characteristic curve. What can you learn from this graph? You can tell, for example, that if Fan B, operating at 800 rpm, is installed in this particular system, a volume flow rate of 840 cfm and a static pressure of 0.52 in. w.g. can be expected. Now follow the 840-cfm line straight down until it crosses the brake horsepower curve for Fan B. From the point of intersection, follow a horizontal line to the left to meet the brake horsepower scale. You will find that 0.15 hp will be required to drive Fan B in this application.

On some graphs, selected horsepowers are plotted as shown in Figure 10. This eliminates the need for a horsepower scale. The points of intersection indicate the performance of a particular fan when driven by a motor operating at the given horsepower. Another alternative method of graphing fan performance plots fan speed (rpm) along the vertical axis, instead of static pressure. Selected static pressure values are represented by the individual performance curves. An advantageous feature of this type of graph is that system resistances are drawn as straight lines, instead of curves.

Regardless of how fan performance is represented, it is important to realize that fans are tested and rated under strictly controlled conditions. Fan performance data are generally based on dry air at “standard” conditions—14.696 psi and 70°F (0.075 lb/ft³). However, when a fan is installed in a working system, it may be required to handle air at some other density. Restrictions at either the inlet or the discharge of the fan also can affect its performance—and, as you know, when one variable changes, all the others change. Likewise, system characteristic curves are based on design conditions. But filters soon begin to load with dirt, coils start condensing moisture, and the positions of dampers are changed. If the actual system resistance is not the same as the design system resistance, the system curve changes. This means that the point of operation along the fan’s performance curve also changes, and the static pressure and volume delivered will not be as calculated. When these variations from “ideal” conditions occur, the fan laws will help you calculate new values.
POINTS OF RATING

Occasionally you may hear the term “point of rating” in discussions of fan performance. A point of rating may be any point on a fan performance curve. Every fan performance curve has an infinite number of points of rating, each of which designates a specific flow rate and the corresponding static pressure. Look back at Figure 9 and note that the system curve passes through the peaks of the two fan performance curves. These peaks are characteristic of forward-curved fans, and help to illustrate a point. Imagine that the two fan performance curves, instead of representing two geometrically similar fans, represent a single fan operating at two different speeds. You can see that if a system curve passes through a particular point of rating at one speed, it passes through the same point of rating at all speeds. In other words, a single system curve passes through the same point of rating on all the performance curves of a given fan.

It was stated earlier that a fan is usually most efficient when used in a system that has a particular system resistance. Since a system is represented by a parabolic curve that passes through a specific point of rating on the fan performance curve, then there is a point of rating to represent each fan’s maximum efficiency. In the case of a forward-curved centrifugal fan, this point of rating is usually near the peak of its performance curve.

Sometimes it is necessary to discuss or compare fan performance in general terms, without referring to definite values of speed, volume, and static pressure. You may want to compare the location of the peak point of rating of one fan with that of another, for example. This point is usually expressed as a certain percentage of “free delivery” cfm. That is to say, the cfm at the peak of any fan performance curve is a certain percentage of the cfm at free delivery (when the static pressure is 0). This percentage remains the same, regardless of the rpm used.

FAN PERFORMANCE TABLES

Although graphs present fan performance data clearly, catalog space does not always permit their use. Some manufacturers use tables instead to show selected points of operation. Most fan tables relate all four variables—rpm, cfm, static pressure, and horsepower. There are two forms of tables in common use. The first is shown in Table 1. Selected cfm values are listed in the first column (at the far left). Selected static pressure values are listed across the top. If you turn this table so that the cfm column is horizontal, you can see how it resembles a standard fan performance graph. The proper rpm and horsepower values are found in the columns that make up the main body of the table. Each of these values represents a point of operation on the fan performance curve. Bold-face values indicate maximum operating efficiencies. Note that the table also includes a column showing outlet velocity in feet per minute (next to the cfm column).

<table>
<thead>
<tr>
<th>Air flow, cfm</th>
<th>Outlet velocity, ft/min</th>
<th>Table 1. Typical fan performance table (maximum operating efficiencies shown in bold type)</th>
<th>Static pressure, in. w.g.</th>
</tr>
</thead>
<tbody>
<tr>
<td>600</td>
<td>718</td>
<td>563</td>
<td>0.050</td>
</tr>
<tr>
<td>800</td>
<td>956</td>
<td>590</td>
<td>0.080</td>
</tr>
<tr>
<td>1000</td>
<td>1198</td>
<td>630</td>
<td>0.120</td>
</tr>
<tr>
<td>1200</td>
<td>1436</td>
<td>680</td>
<td>0.170</td>
</tr>
<tr>
<td>1400</td>
<td>1675</td>
<td>733</td>
<td>0.225</td>
</tr>
</tbody>
</table>

A second form of fan performance table is shown in Table 2. This type of table is frequently used by distributors of general-purpose blowers that are sold complete with motor, belts, and a set of pulleys. It lists maximum horsepower values and rpm values in the first two columns at the left. Selected static pressure values listed across the top, as in Table 1. The body of the table contains the corresponding cfm values. Normally, fan performance tables provide only values that fall within the recommended operating range.

<table>
<thead>
<tr>
<th>hp</th>
<th>rpm</th>
<th>Static pressure, in. w.g.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>1/8</td>
</tr>
<tr>
<td>1</td>
<td>293</td>
<td>7200</td>
</tr>
<tr>
<td>1 1/2</td>
<td>350</td>
<td>9300</td>
</tr>
<tr>
<td>2</td>
<td>400</td>
<td>9200</td>
</tr>
</tbody>
</table>
PERFORMANCE OF VARIOUS FAN TYPES

Several different types of fans were described at the beginning of this chapter. With an understanding of fan performance graphs and tables, you can now look at how their performance characteristics differ from each other.

FORWARD-CURVED CENTRIFUGAL FANS

A centrifugal impeller with forward-curved blades produces more cfm and more static pressure at a given rpm than either radial or backward-curved impellers. Consequently, forward-curved centrifugal fans are popular in domestic equipment, where space is often at a premium and noise must be kept to a minimum.

Forward-curved centrifugal fans display a typical “dip” in their performance curves. Remember from Figure 7 that the area to the left of the peak is the so-called “unstable” portion of the curve, where operation is to be avoided. If you are considering a particular fan for a given system, and the point of operation falls in this unstable area, you may be able to solve the problem by selecting a narrower model with the same impeller diameter (and nearly the same rpm). This is to say that the efficient operating range of a narrower centrifugal fan falls to the left of a wider one’s.

Look back at Figure 8, which plots cfm against horsepower for two forward-curved centrifugal fans. As you can see, for a given fan speed, the required horsepower goes up as the system resistance goes down (and cfm delivery increases). This can be demonstrated by removing an air conditioning unit from the air distribution system and running it in free air. Often the motor will overheat to the point of cutting out on overload.

The performance curve of a backward-curved centrifugal fan usually does not display a “dip” to the left of the peak cfm/static pressure point. As a result, a backward-curved centrifugal fan may not have an unstable range. Figure 11 shows cfm plotted against both static pressure and horsepower for a typical backward-curved centrifugal fan.

A backward-curved centrifugal fan may be designed to be non-overloading. That is, for a given rpm, the required horsepower reaches a maximum at some point to the left of free delivery. Further decreases in system resistance do not require an increase in horsepower.

This can be quite advantageous in large commercial installations, where the added capacity needed to ensure freedom from mechanical breakdown can represent a large investment in an oversized motor. The necessarily higher speed, larger size, and extra cost of backward-curved centrifugal fans is offset by the higher efficiencies achieved over a wider range of system resistance. Greater efficiency, in turn, can mean smaller motors and therefore lower operating costs.

Figure 12 shows performance curves for a typical propeller fan. The shaded areas show maximum operating efficiency ranges for propeller fans and comparably sized centrifugal fans and backward-curved centrifugal fans.

Figure 11. Typical performance curves for centrifugal fans with backward-curved blades

Figure 12. Typical propeller performance curves

Figure 13. Typical mixed-flow fan performance curves (shaded area shows maximum operating efficiency)
fans. As you can see, the normal operating range of the propeller fan (the area in which the point of operation should be located) falls to the right of that of the centrifugal fan. This means that propeller fans perform better when system resistance is low—which is one of the reasons why this type of fan is found in air-cooled condensers and cooling towers. In such systems, ductwork normally is not used, and large volumes of air are required in proportion to the static pressure created.

Attempts to make a propeller fan do a job that requires a centrifugal fan can, at best, be only partially successful. The additional rpm required often creates objectionable noise, excessive power consumption, and structural failure.

**MIXED-FLOW FANS**

As you might expect, the operating range of a mixed-flow fan fills a gap on a fan performance graph between the working ranges of equally sized propeller fans and centrifugal fans. You can see such a gap in Figure 12. Frequently, this gap can be successfully eliminated simply by varying the size of either the centrifugal fan or the propeller fan.

Most of the mixed-flow impellers in use today resemble a high-pitched propeller, and may go unnoticed for what they really are. Many of them can be found in the condenser side of window air conditioners. The air discharged radially from the fan blades is distributed over the face of a coil much larger than the diameter of the impeller. Typical mixed-flow fan performance curves are shown in Figure 13. Note that the horsepower increases with decreasing static pressure in this example, although in some cases the performance curves may be nearly flat.