# Fans 101

ASHRAE Louisville Chapter

# Fans Make Our World Go Round

- A Fan is the most widely used device to move air
- Almost every building in the world has a fan installed in it
- Fans are one of the most Basic and simplest machines yet are the most misunderstood

# The Basics

- CFM (cubic feet per minute): Fans are rated in CFM for airflow. One CFM is one cubic foot of air being moved through a fan or piece of ductwork in a one minutes time
- SP (static pressure): Is the measure of the amount of pressure required to move a specified amount of air (CFM) through a piece of ductwork
- HP (horsepower): The amount of work a machine can perform in a period of time. One HP equals 33,000 foot/pounds of work per minute also equal to .746 kw
- BHP (brake horsepower): The amount of horsepower required to move a fans wheel to produce a required flow and pressure. This is a consumption rating as compared to the production rating of horsepower itself

# Fan Wheel Designs

- FC (forward curved): Wheel design that is used for low pressures high flows. First mass produced wheel type used mainly in air handlers. 60% efficient
- AF (double thickness airfoil): Wheel design that is good for medium pressures and high airflows. This is the most efficient wheel design. 85% efficient
- BC (backward curved single thickness airfoil): Same basic design as the airfoil but this wheel has only a single surface that is airfoil in shape. 80% efficient
- Radial: This wheel design is used in industrial material handling applications and medium to high pressure applications with lower airflows. 72% efficient
- RT (radial tip): This wheel design is a BC fan that has had the very edge of the blade flattened out. This is used in applications where higher flows are required at medium to high static pressures. Usually in heavy duty industrial application. 78% efficient
- PB (pressure blower): This is a radial bladed very narrow width wheel that is designed for low airflows and high pressures. 75% efficient
- Axial: This wheel design is used in axial fans. These are generally low pressure high flow exhaust applications such as roof exhaust fans. 78% efficient

## Drive Arrangements For Centrifugal Fans AMCA Standard 99-2404-03

**NOTES:**

**SW** - Single Width **DW** - Double Width

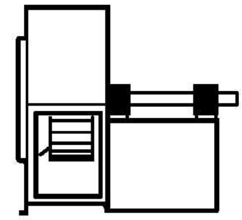
**SI** - Single Inlet **DI** - Double Inlet

Arrangements 1, 3, 7 and 8 are also available with bearings mounted on pedestals or base set independent of the fan housing.

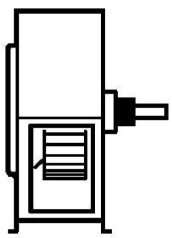
For designation of rotation and discharge, see page 5.

For motor position, belt or chain drive, see page 6.

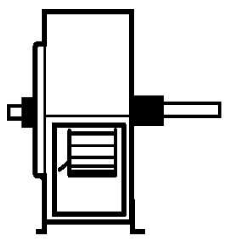
For designation of position of inlet boxes, see page 4.



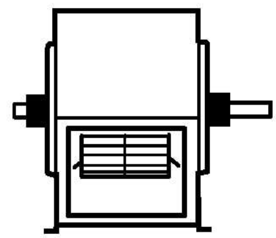
**ARR. 1 SWSI** - For belt drive or direct connection. Impeller overhung. Two bearings on base.



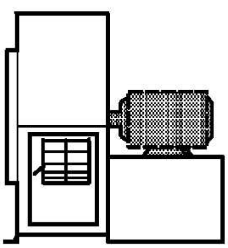
**ARR. 2 SWSI** - For belt drive or direct connection. Impeller overhung. Bearings in bracket supported by fan housing.



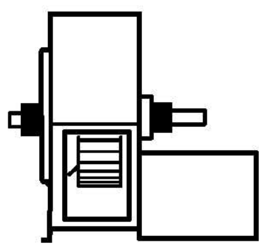
**ARR. 3 SWSI** - For belt drive or direct connection. One bearing on each side and supported by fan housing.



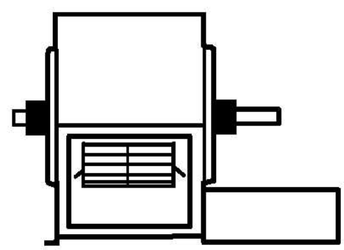
**ARR. 3 DWDI** - For belt drive or direct connection. One bearing on each side and supported by fan housing.



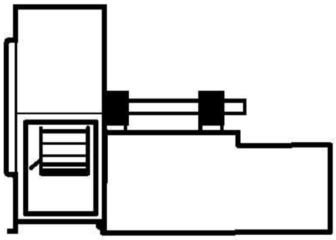
**ARR. 4 SWSI** - For direct drive. Impeller overhung on prime mover shaft. No bearings on fan. Prime mover base mounted or integrally directly connected.



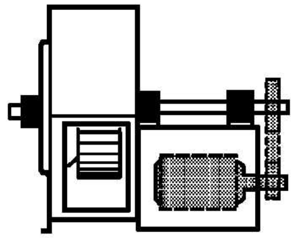
**ARR. 7 SWSI** - For belt drive or direct connection. Arrangement 3 plus base for prime mover.



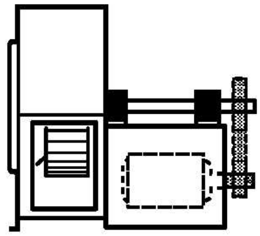
**ARR. 7 DWDI** - For belt drive or direct connection. Arrangement 3 plus base for prime mover.



**ARR. 8 SWSI** - For belt drive or direct connection. Arrangement 1 plus extended base for prime mover.



**ARR. 9 SWSI** - For belt drive. Impeller overhung, two bearings, with prime mover outside base.



**ARR. 10 SWSI** - For belt drive. Impeller overhung, two bearings, with prime mover inside base.

## Drive Arrangements For Centrifugal Fans AMCA Standard 99-2404-03

**NOTES:**

**SW** - Single Width **DW** - Double Width

**SI** - Single Inlet **DI** - Double Inlet

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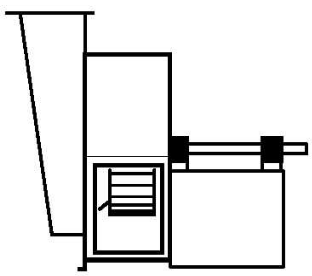
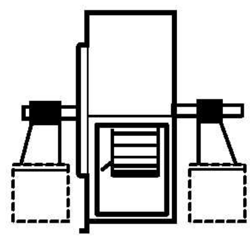
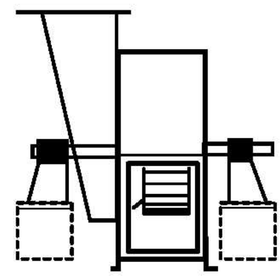


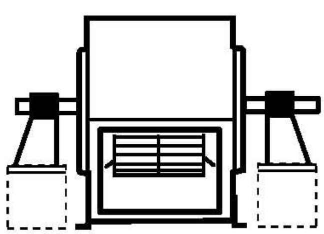
**ARR. 1 SWSI With Inlet Box** - For belt drive or direct connection. Impeller overhung, two bearings on base. Inlet box may be self-supporting.



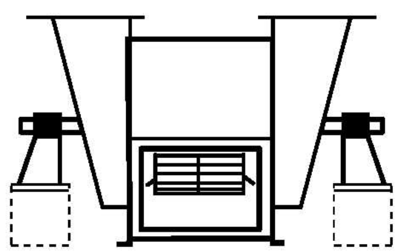
**ARR. 3 SWSI With Independent Pedestal** - For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.



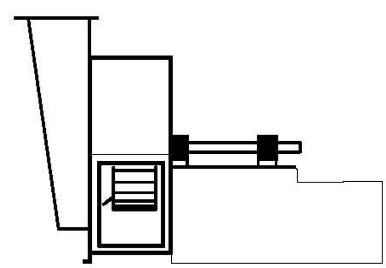
**ARR. 3 SWSI With Inlet Box and Independent Pedestals** - For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.



**ARR. 3 DWDI With Independent Pedestal** - For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals.

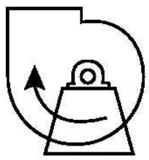


**ARR. 3 DWDI With Inlet Box and Independent Pedestals** - For belt drive or direct connection fan. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.

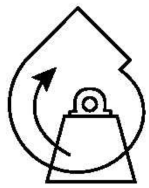


**ARR. 8 SWSI With Inlet Box** - For belt drive or direct connection. Impeller overhung, two bearings on base plus extended base for prime mover. Inlet box may be self-supporting.

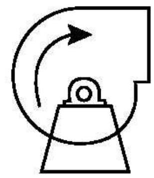
## Designations For Rotation & Discharge Of Centrifugal Fans AMCA Standard 99-2406-03



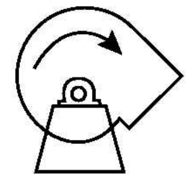
Clockwise
Up Blast
CW 360



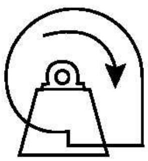
Clockwise
Top Angular Up
CW45



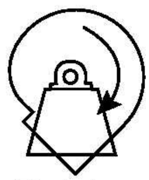
Clockwise
Top Horizontal
CW90



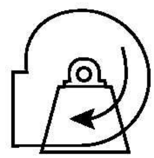
Clockwise
Top Angular Down
CW45



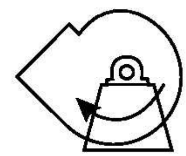
Clockwise
Down Blast
CW 180



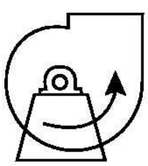
Clockwise
Bottom Angular Up
CW225



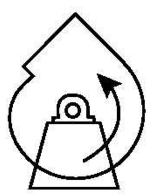
Clockwise
Bottom Horizontal
CW270



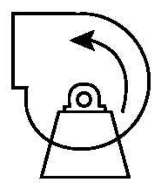
Clockwise
Bottom Angular Up
CW315



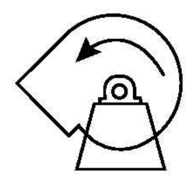
Counterclockwise
Up Blast
CW 360



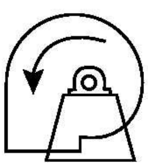
Counterclockwise
Top Angular Up
CW45



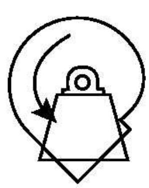
Counterclockwise
Top Horizontal
CW90



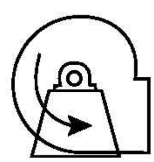
Counterclockwise
Top Angular Down
CW135



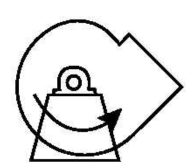
Counterclockwise
Down Blast
CW 180



Counterclockwise
Bottom Angular Down
CW225



Counterclockwise
Bottom Horizontal
CW270

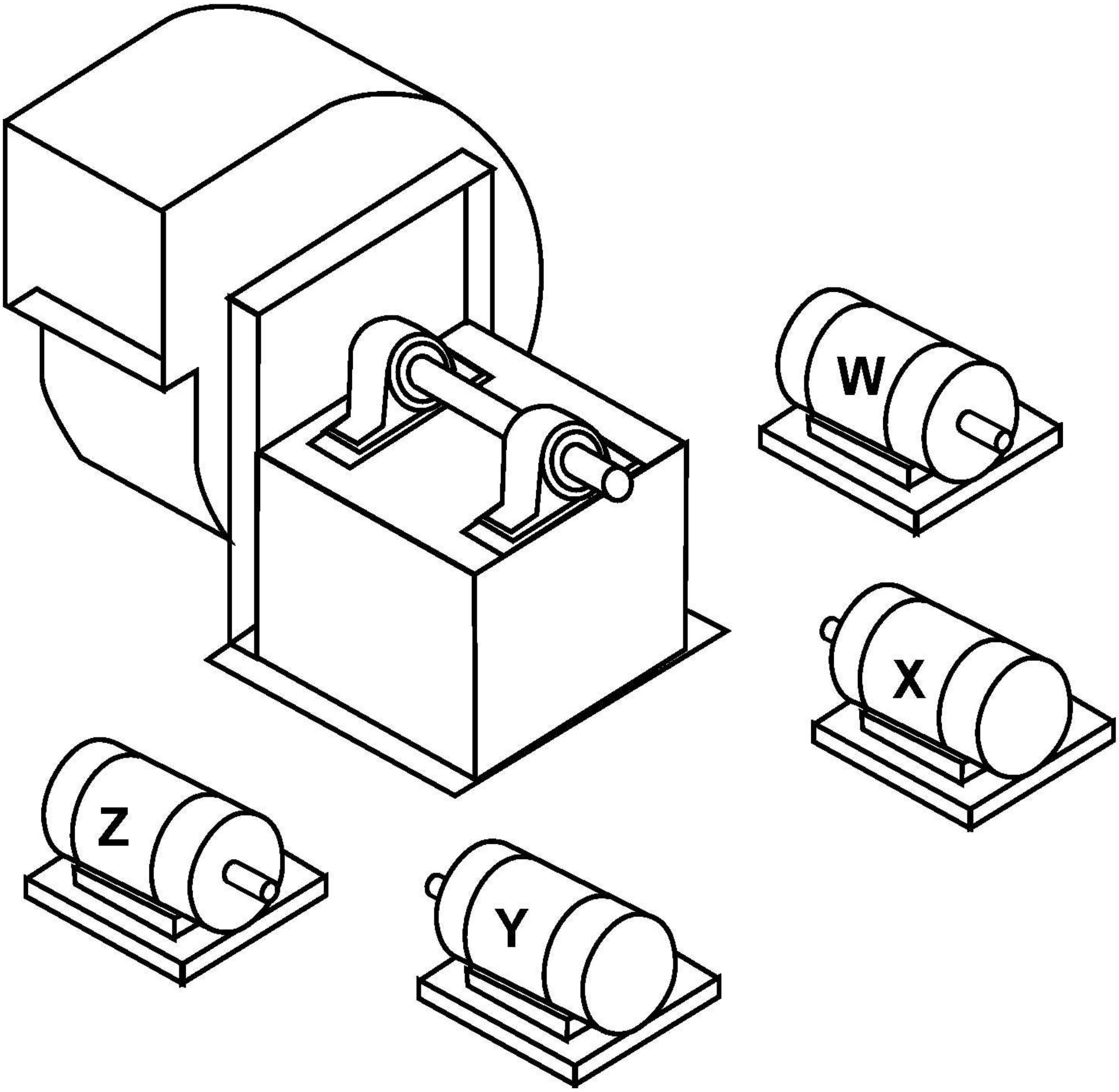


Counterclockwise
Bottom Angular Up
CW315

**NOTES:**

1. Direction of rotation is determined from drive side of fan (see Note 2 and 3).
2. On single inlet fans, drive side is always considered as the side opposite fan inlet (even when driven through fan inlet).
3. On double inlet fans with drives on both sides, drive side is that with the higher powered drive unit.
4. Direction of discharge is determined in accordance with diagrams. Angle of discharge is referred to the vertical axis of fan and designated in degrees from such standard reference axis. Angle of discharge may be any intermediate angle as required.
5. For fan inverted for ceiling suspension, or side wall mounting, direction of rotation and discharge is determined when fan is resting on floor.

## Motor Positions For Belt Or Chain Drive Centrifugal Fans
## AMCA Standard 99-2407-03




**NOTE:**

Location of motor is determined by facing the drive side of fan and designating the motor positions by letters W, X, Y or Z as the case may be.

# FAN LAWS AND SYSTEM CURVES

## INTRODUCTION

The purpose of this Engineering Letter is to explain the basis and application of the rules used to predict fan performance in a given system. With a basic understanding of these rules, the performance of a fan can be quickly calculated for various conditions.

## SYSTEM REQUIREMENTS

The three fundamental rules governing fan performance are commonly called the “fan laws.” These rules are only valid within a fixed system with no change in the aerodynamics or airflow characteristics of the system. For the purpose of this discussion, a *system* is the combination of ductwork, hoods, filters, grills, collectors, etc., through which air is distributed. Therefore, these rules can also be referred to as “system laws.”

## VOLUME AND PRESSURE

The motion of any mass causes friction with its surroundings. The movement of air through a system causes friction between the air molecules and their surroundings (duct walls, filter media, etc.) and any other air molecules. Energy is required to overcome this friction, or resistance. The faster the air moves the greater the resistance to flow and the more energy is required to push or pull the air through the system.

This energy is stated in terms of pressure. The portion of the pressure that results in air velocity is described as velocity pressure (VP). The portion necessary to overcome friction in the air and in the system is described as static pressure (SP). The sum of the two is described as total pressure (TP).

The law of physics, for motion, is expressed algebraically as:

$$V = \sqrt{2gh} \qquad \text{or} \qquad V^2 = 2gh$$

where V = velocity of flow
g = force of gravity
h = pressure causing flow

As can be seen from the equation, the pressure necessary to cause flow is proportional to the square of the velocity. In a system, this means that SP will vary as the square of the change in velocity or volume expressed in cubic feet per minute (CFM). This makes it possible to predict all possible combinations of SP at the corresponding CFM given any one such calculated relationship of SP and CFM for a fixed system.

For example, a system is calculated to require a static pressure equal to 2" water gauge at an airflow rate of 1000 CFM. If it is desired to increase the flow to 1500 CFM without any physical change in the system, the required SP would be:

$$(1500 \div 1000)^2 \times 2'' = 4.5'' \text{ SP}$$

$$\left(\frac{\text{CFM new}}{\text{CFM old}}\right)^2 = \frac{\text{SP new}}{\text{SP old}}$$

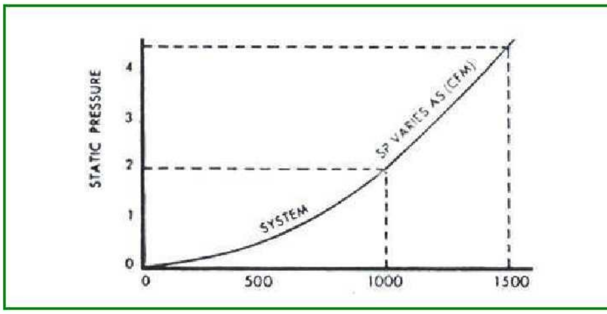



**Figure 1 - System Curve**

The same calculation using any number of varying CFM ratings would result in a plotted curve as shown in Figure 1. Regardless of fan type, fan size, or volume of flow through a system, the relationship of CFM to SP will not change unless the system itself is altered in some way. SP always varies as the square of the change in CFM. The only exception to this rule is found in a laminar flow characteristic where VP is of far greater importance than SP. Such circumstances are not typical of fan systems.

## FAN LAWS

In air movement systems, it is the fan wheel that does the work. In a sense, the fan wheel acts like a shovel. As it revolves, it discharges the same volume of air with each revolution. Working within a fixed system, a fan will discharge the same volume of air regardless of air density, (disregarding the effects of compression at high pressures).

If the fan RPM is increased, the fan will discharge a greater volume of air in exact proportion to the change in speed. This is the first “fan law.”

1. CFM varies in direct proportion to change in RPM

$$\text{CFM (new)} = \frac{\text{RPM (new)}}{\text{RPM (old)}} \times \text{CFM (old)}$$

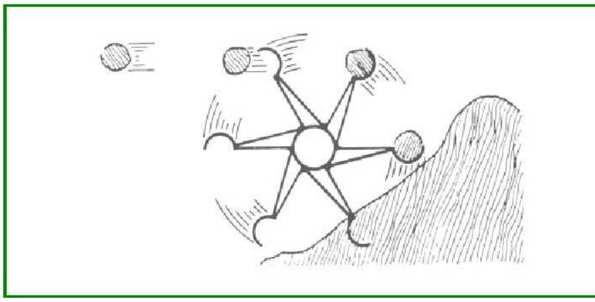


**Figure 2 - A fan wheel is a constant volume device.**

As shown earlier, in a system, the SP varies as the square of the change in CFM. Since CFM varies directly with RPM, RPM can be substituted for CFM in the system equation. Therefore, SP varies as the square of the change in RPM. This is the second "fan law."

2. SP varies in proportion to the change in $(RPM)^2$

$$SP\ (new) = \left(\frac{RPM\ (new)}{RPM\ (old)}\right)^2 \times SP\ (old)$$

The efficiency of a fan is a function of its aerodynamic design and point of operation on its SP/CFM curve (see Figure 3). As the fan speed changes, this relative point of operation remains unchanged as long as the system remains unchanged. Thus, the fan brake horsepower varies proportionally as the cube of the change in RPM. This is the third "fan law."

3. BHP varies in proportion to the change in $(RPM)^3$

$$BHP\ (new) = \left(\frac{RPM\ (new)}{RPM\ (old)}\right)^3 \times BHP\ (old)$$

It is important to remember that each of these "fan law" relationships takes place simultaneously and cannot be considered independently.

## FAN CURVE AND SYSTEM CURVE

As stated previously, a system curve can be plotted to show all possible combinations of SP and CFM for a given fixed system. Any fan used on that system must operate somewhere on that system curve.

Fan performance is determined by laboratory testing and is presented graphically in the form of fan curves. Unless it is physically altered in some way, a fan must operate somewhere on its SP/CFM curve. The relative shape of that curve will not change, regardless of fan speed.

Because the fan and system can each only operate somewhere on their own respective curves, a fan used on a fixed system can only have one point of operation. The point of operation, as shown in Figure 3, is the intersection of the system curve and the fan SP CFM curve.

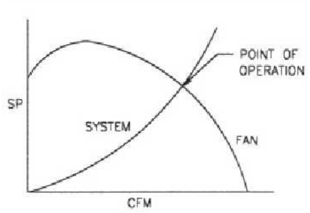
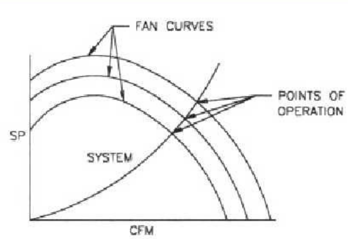



Figure 3

Figure 4




If the fan speed is increased or decreased, the point of operation will move up or down the existing system curve. This is shown in Figure 4.

The following are examples of how the fan curve can be used to calculate changes to flow and pressure requirements.

Example 1: A fan has been selected to deliver 35,530 CFM at 8" SP. The fan runs at 1230 RPM and requires 61.0 BHP.

After installation, it is desired to increase the output 20%. At what RPM must the fan run? What SP will be developed? What BHP is required?

1. CFM varies as RPM
   (1230) (1.20) = 1476 RPM

2. SP varies as $(RPM)^2$
   $(1476/1230)^2$ (8) = 11.52" SP

3. BHP varies as $(RPM)^3$
   $(1476/1230)^3$ (61.0) = 105.4 BHP

Example 2: A fan was originally installed to deliver 10,300 CFM at 2 1/4" SP and to run at 877 RPM, requiring 5.20 BHP.

After installation, it is found that the system only delivers 9,150 CFM at 2 1/2" SP and uses 4.70 BHP. This indicates the original calculations were in error, or that the system was not installed according to plan. What fan RPM and BHP will be necessary to develop the desired 10,300 CFM? What SP should have been figured?

1. CFM varies as RPM
   (10,300/9,150) (877) = 987 RPM

2. SP varies as $(RPM)^2$
   $(987/877)^2$ (2.50) = 3.17" SP

3. BHP varies as $(RPM)^3$
   $(987/877)^3$ (4.70) = 6.70 BHP

## CONCLUSION

Use of the "fan laws" is based on a fixed system and a non-modified fan. Adding or deleting system components such as dampers, or incurring density changes, will create completely new system curves. Changing fan accessories such as inlet boxes, evases, or inlet dampers will alter the fan's performance curve from standard. These variables must be considered before the fan laws can be applied.

During the process of system design, the fan laws can be helpful in determining alternate performance criteria or in developing a minimum/maximum range. If "safety factors" are applied to system calculations, it should be recognized that a 10% factor on volume will result in an increase in horsepower of 33% according to the third fan law. An evaluation should be made weighing the necessity of the safety factor versus the cost penalty incurred.

# UNDERSTANDING FAN PERFORMANCE CURVES

## INTRODUCTION

One of the most important documents customers request from fan manufacturers is performance curves. In addition to graphically depicting the basic fan performance data of CFM, RPM, and SP (on the static pressue curve) and BHP (on the brake horsepower curve), these curves also illustrate the performance characteristics of various fan types, like areas of instability, or the rate of change between flow and pressure. With some basic knowledge of performance curves, decisions can be made concerning fan selection, fan and system alterations, or the advisability of using a fan in a modulating system, for example.

Except for very large fans, performance curve information is generated by connecting the fan to a laboratory test chamber. Very specific test procedures are followed as prescribed in the Air Movement and Control Association's Standard 210 to assure uniform and accurate readings. Data points are collected at a given RPM while the flow is slowly modulated from full closed to full open. The information gathered is then used to develop computer selection programs and published capacity tables for use by system designers and end users.

## STATIC PRESSURE CURVE

The static pressure curve provides the basis for all flow and pressure calculations. This curve is constructed by plotting a series of static pressure points versus specific flow rates at a given test speed. While the static pressure curve depicts a fan's performance at a given speed, it can be used to determine the fan's pressure capability at any volume.

In addition, it is also possible to approximate the fan's performance at other speeds by applying the following fan laws:

1. CFM varies as RPM
2. SP varies as $(RPM)^2$
3. BHP varies as $(RPM)^3$

To locate a fan's point of operation, first locate the required static pressure on the SP scale at the left of the curve. Then draw a horizontal line to the right, to the point of intersection with the SP curve. Next, draw a vertical line from the point of operation to the CFM scale on the bottom to determine the fan's flow capability for that SP at the given speed.

As shown in Figure 1, the performance for this fan is 8750 CFM and 12" SP at 1750 RPM.

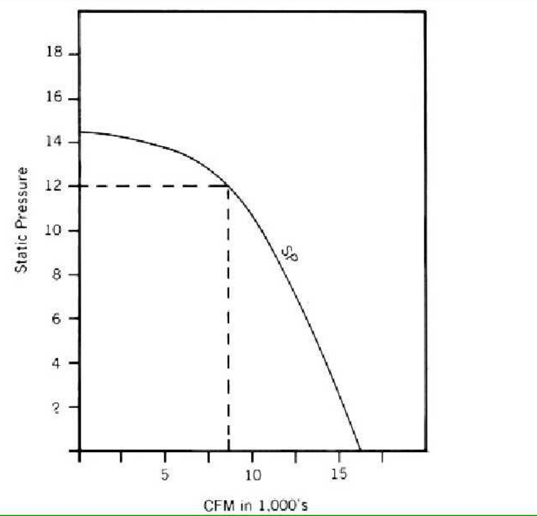



**Figure 1 - Static Pressure Curve**

Assuming this same fan was intended to operate at 1200 RPM, the fan laws can be applied to obtain performance at this lower speed.

1. CFM varies as RPM

$$\frac{\text{CFM (new)}}{\text{CFM (old)}} = \frac{\text{RPM (new)}}{\text{RPM (old)}}$$

Therefore:

$$\text{CFM (new)} = \frac{1200}{1750}\,(8750) \quad = 6000\text{ CFM}$$

2. SP varies as $(RPM)^2$

$$\frac{\text{SP (new)}}{\text{SP (old)}} = \left(\frac{\text{RPM (new)}}{\text{RPM (old)}}\right)^2$$

Therefore:

$$\text{SP (new)} = \left(\frac{1200}{1750}\right)^2 \quad (12) = 5.6\text{'' SP}$$

## BRAKE HORSEPOWER CURVE

Once the CFM and SP have been determined, a BHP rating can be established. An accurate BHP rating is necessary to properly size the motor or to determine the operating efficiency of one fan as compared to another. Performance curves contain a BHP curve from which the BHP rating can be determined for specific capacities. To determine BHP at a specific point of operation, a horizontal line is drawn to the right from the point of intersection of the vertical CFM line and the BHP curve.

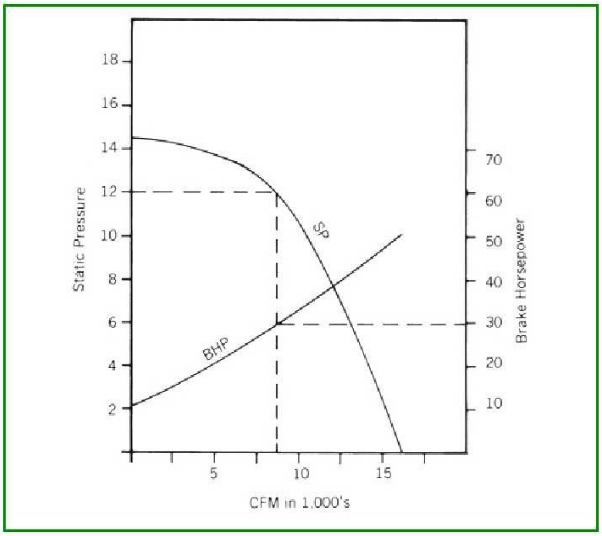



**Figure 2 - Performance Curve**

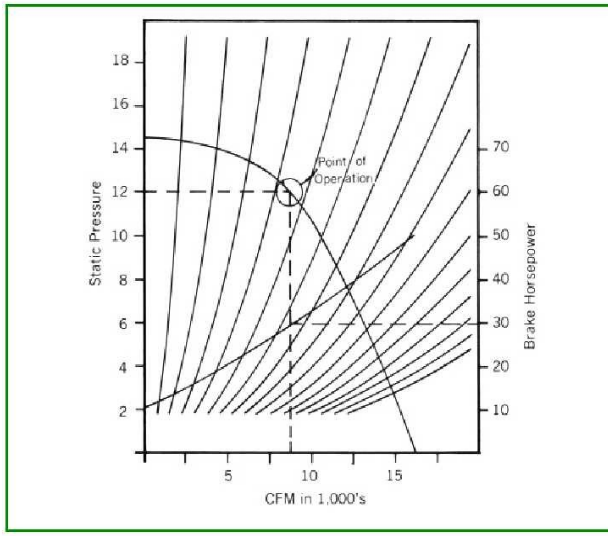



**Figure 3 - Performance Curve with System Lines**

As shown in Figure 2, the fan operating at 8750 CFM and 12" SP at 1750 RPM is rated at 30 BHP. By employing the third fan law, the BHP rating can be determined for operation at 1200 RPM.

3. BHP varies as $(\text{RPM})^3$

$$\frac{\text{BHP (new)}}{\text{BHP (old)}} = \left(\frac{\text{RPM (new)}}{\text{RPM (old)}}\right)^3$$

Therefore:

$$\text{BHP (new)} = \left(\frac{1200}{1750}\right)^3 (30) = 9.67 \text{ BHP}$$

## SYSTEM LINES

Since fans are tested and rated independently from any type of system, a means of determining the fan's capabilities within a given system must be provided. The fan laws apply equally to any system; therefore, CFM and SP variations within the system are predictable. This enables system lines to be superimposed on performance curves to simplify performance calculations. The system line is nothing more than the sum of all possible CFM and SP combinations within the given system. Any combination of fan and system must operate somewhere along that system line.

Because a fan must operate somewhere along its SP curve and since the system has a known system line, their intersection is the point of operation. If the fan speed is changed, the point of operation must move up or down the system line. If the system is changed in some way, the point of operation must move up or down the SP curve. In practice, these principles can be used to check the accuracy of fan performance and system design.

## USING PERFORMANCE CURVES

Figure 3 illustrates the point of operation of a fan selected for 8750 CFM and 12" SP operating at 1750 RPM. Included in Figure 3 are a number of different system lines. If the system does not operate properly upon start-up, measurements can be taken and compared against the available performance curve.

Let's assume that a tachometer reading indicates the fan is running at 1200 RPM instead of 1750 RPM, because of mistakes in motor speed or drive selection, and an airflow check indicates only 6000 CFM. These readings confirm that the system was calculated correctly and that the fan speed must be corrected to 1750 RPM as originally specified to achieve the desired 8750 CFM. If the tachometer reading indicates the proper speed but the airflow reading is down, additional system resistance beyond that originally calculated is indicated. This additional resistance could be caused by partially closed louvers/dampers, changes in duct sizing from the original design, system effect losses, or just an error in the system-resistance calculations. The deficiency can usually be corrected by either altering the system or increasing the fan speed.

Often, performance curves for one speed must be used to determine the performance of a fan for use on systems requiring more air or higher pressures. A performance curve such as Figure 4 can be used to determine fan performance beyond the SP scale shown by using the fan laws to obtain a reference point of operation on the system line. This can be accomplished by applying some suitable factor to the required CFM and the square of that factor to the required SP.

For example, the performance curve shown in Figure 4 can be used to determine fan performance requirements for a system calculated at 15,000 CFM and 23.5" SP, even though that point is beyond the curve. By determining a suitable reference capacity using the fan laws, that falls within the curve data, fan performance requirements can be obtained at the curve speed and then projected up to the system requirements using the fan laws once again.

The required 15,000 CFM and 23.5" SP is on the same system line as 10,000 CFM at 10.4" SP which intersects the fan's SP curve drawn for 1750 RPM and has a corresponding BHP of 33.0 at 1750 RPM. Therefore:

$$\text{RPM (new)} = \frac{15000}{10000} (1750) = 2625 \text{ RPM}$$

$$\text{BHP (new)} = \left(\frac{15000}{10000}\right)^3 (33.0) = 111 \text{ BHP}$$

## FAN PERFORMANCE CHARACTERISTICS

The performance characteristics of a fan can be determined from the performance curve at a glance. These characteristics include such things as stability, increasing or non-overloading BHP, and acceptable points of operation.

Fan performance is based on certain flow characteristics as the air passes over the fan wheel blades. These flow characteristics are different for each generic fan or wheel type, (i.e. radial, forward-curved, backwardly-inclined, radial-tip, and axial). Thus, the performance characteristics will be different for each of these general fan types. Further, these performance characteristics may vary from one manufacturer to the next depending upon the particular design. The characteristics described in this letter are based on **nyb** fan equipment.

The performance curves presented in Figures 1 through 4 are typical of fans with radial-blade wheels. The SP curve is smooth and stable from wide open to closed off. The BHP curve clearly indicates that the BHP increases steadily with the volume of air being handled as shown in Figure 4.

Fans with forward-curved wheels, such as shown in Figure 5, also have a BHP curve that increases with the volume of air being handled. The SP curve differs significantly from the radial since it exhibits a sharp "dip" to the left of the static pressure peak. This sharp dip (shaded area) indicates unpredictable flow characteristics. Though not technically accurate, this region is often referred to as the the "stall" region. It indicates that at these combinations of pressure and relatively low volumes, the airflow characteristics across the wheel blades change or break away so that the fan performance point is no longer stable. Any fan with this characteristic SP curve should not be selected for operation in the unstable area.

As shown in Figure 6, the SP curve for a backwardly-inclined fan has a sharp dip to the left of the static pressure peak. This indicates an area of instability. However, the backwardly-inclined SP curve is generally steeper than that of the forward-curved wheel indicating its desirability for use in higher pressure systems. Therefore, variations in system resistance will result in smaller changes in volume for the BI Fan when compared to the FC Fan.

Even though New York Blower centrifugal fans with AcoustaFoil® wheels are stable in the area left of the peak, the majority of fans with backwardly-inclined wheels exhibit an SP curve similar in appearance to that of the forward-curved fan. The SP curve shown (in Figure 7) for fans using AcoustaFoil (air-foil, backwardly-inclined) wheels exhibits a much smoother depression to the left of the static pressure peak. This indicates that the overall fan design is such that internal flow characteristics remain desirable or predictable even in the region left of peak and that performance in this region is stable.

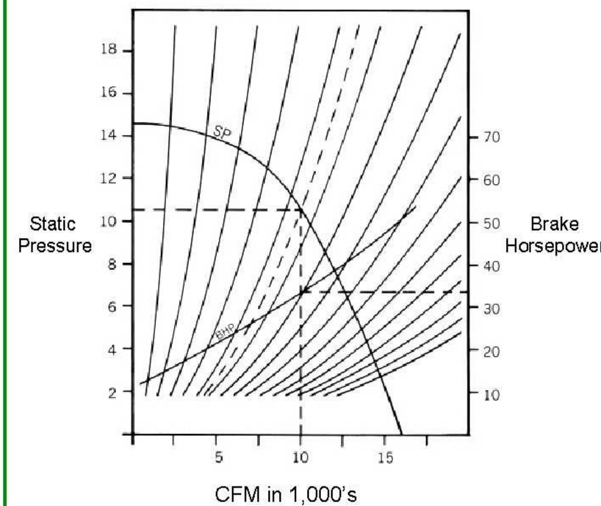



**Figure 4 – Typical Radial-Blade Fan Performance Curve**

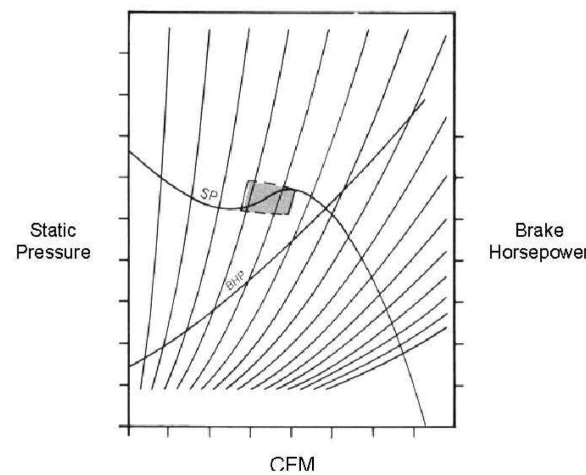



**Figure 5 – Typical FC Fan Performance Curve**

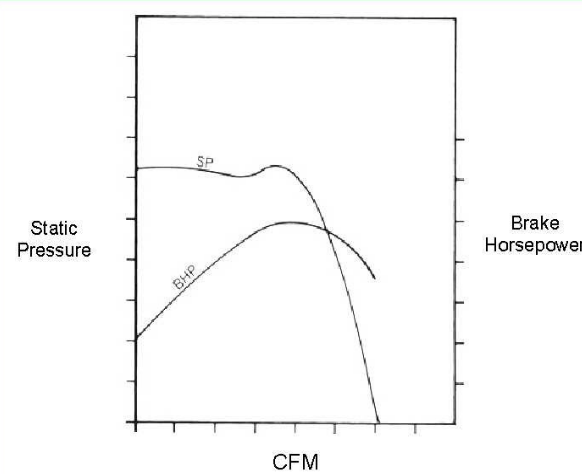



**Figure 6 – Typical BI Fan Performance Curve**

The BHP curve for all backwardly-inclined fans is the major difference between them and all other fan types. As shown in Figures 6 and 7, the BHP curve for backwardly-inclined fans reaches a peak and then drops off as the fan's volume increases. With this "non-overloading" BHP characteristic, it is possible to establish a maximum BHP for a given fan speed and select a motor that can not be overloaded despite any changes or errors in system design. Because BHP varies as $(RPM)^3$ , this non-overloading characteristic does not apply to increases in fan speed, but it is very useful for motor protection against errors or changes in system calculations and installation.

Figures 5 and 6 indicate certain unacceptable selection areas on the SP curve. Although stability or performance may not be a problem, a point of operation down to the far right on the SP curve should be avoided. Selecting a fan that operates far down to the right, eliminates the flexibility to compensate for future system changes. Also, the fan is less efficient in this area as compared to a larger fan operating at a slower speed. Figure 7 shows the best selection area on the SP curve and the area in which the majority of capacity tables are published.

As is evident in Figure 8, the radial-tip fan design combines the backwardly-inclined SP curve characteristics with the radial fan's BHP curve. The radial tip is often more efficient than radial fans and typically best applied in high-pressure applications. As a result of its efficiency and dust-handling capabilities, the radial-tip fan can also be applied to lower pressure material conveying systems.

The term axial fan is used to describe various propeller, vaneaxial, tubeaxial, and duct fans. The performance curves of these fans are characterized by the ability to deliver large volumes of air in relatively low pressure applications. As can be seen in Figure 9, the axial flow fan is distinguished by a drooping BHP curve that has maximum horsepower at no flow or closed-off conditions. The axial fan SP curve exhibits an area of extreme instability to the left of the "hump" in the middle of the curve. Depending upon the severity, axial fans are normally only selected to the right of this area.

## CONCLUSION

A good working knowledge of performance curves is necessary to understand the performance characteristics and capabilities of different fan equipment. Use of performance curves in the selection of fan types and sizing will assure stable and efficient operation as well as future flexibility.

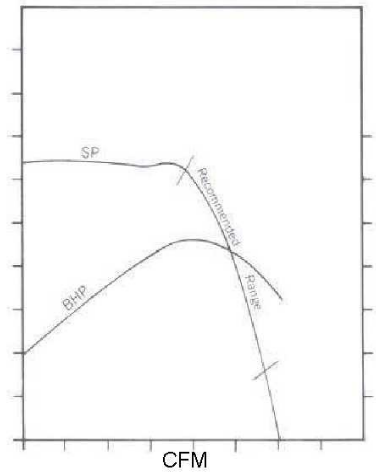



**Figure 7 – Typical AcoustaFoil Fan Performance Curve**

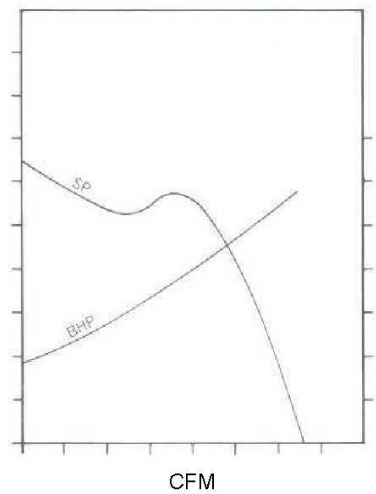



**Figure 8 – Typical Radial-Tip Fan Performance Curve**

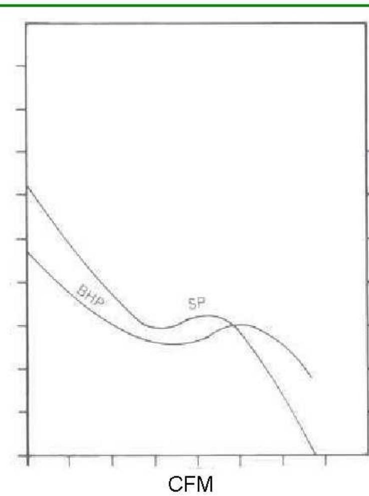



**Figure 9 – Typical Axial Fan Performance Curve**

# Should I Use A VFD On My Fan

- A VFD (Variable Frequency Drive) is used to change the operating speed of a fan by electrical not physical means
- Speed change is required on most commercial and industrial installations when a system is balanced
- Saves money on maintenance and first cost of an installation – Starter versus VFD
- Saves on the operating cost of a fan by removing the need for a damper and allowing for direct drive fans to be used eliminating belt losses

# VFD Economics

Fan Example: 10HP $.06kw/hr 8000 hrs/yr

| % Speed | HP Required VFD | HP Required Damper | Duty Cycle % of time | HP Required (weighted) VFD | HP Required (weighted) Damper |
|---|---|---|---|---|---|
| 100 | 10 | 10 | 20 | 2.00 | 2.00 |
| 80 | 5.1 | 9.6 | 30 | 1.53 | 2.88 |
| 60 | 2.2 | 8.6 | 30 | .66 | 2.58 |
| 40 | .64 | 7.7 | 20 | .13 | 1.54 |
| | | | | 4.32 | 9.00 |

# VFD Economics Summary

- Operating cost per year VFD: 4.32(wthp) x .746(kw/hp) x 8000 hours x .06(cost per kw/hr) = $ 1,546.91
- Operating cost per year Damper: 9.0(wthp) x .746(kw/hp) x 8000 hours x .06(cost per kw/hr) = $ 3,222.72
- Annual Savings bases on VFD control versus control by a Damper would be $ 1,675.81

# Thank You


Brien M Buelow

MK Systems Inc.

1455 Brummel

Elk Grove Village, IL 60007