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A CASE STUDY OF CENTRIFUGAL PUMP WITH VARIOUS LOSSES IN FAN ROTATIONS

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Abstract: In this paper a detail case study about the working principle of centrifugal pump, Nowadays, the centrifugal pumps became very popular because of recent development of high speed electric motors, steam turbines etc. Centrifugal pumps can be single-stage or may be multistage pumps. It depends upon the number of impellers used in the pump. Single stage pump consists of only one impeller while in multistage pumps the impellers are mounted in the series in pumps.

Keywords: Six phase transmission line; fault analysis.

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INTRODUCTION

CHAPTER: 1.

A **centrifugal fan** is a mechanical device for moving air or other gases. The terms "blower" and "squirrel cage fan", (because it looks like a hamster wheel), are frequently used as synonyms. These fans increase the speed and volume of an air stream with the rotating impellers.
Centrifugal fans use the kinetic energy of the impellers to increase the volume of the air stream, which in turn moves them against the resistance caused by ducts, dampers and other components. Centrifugal fans displace air radially, changing the direction (typically by 90°) of the airflow. They are sturdy, quiet, reliable, and capable of operating over a wide range of conditions.

Centrifugal fans are constant displacement devices or constant volume devices, meaning that, at a constant fan speed, a centrifugal fan moves a relatively constant volume of air rather than a constant mass. This means that the air velocity in a system is fixed even though the mass flow rate through the fan is not.

Centrifugal fans are not positive displacement devices. Centrifugal fans have certain advantages and disadvantages when contrasted with positive-displacement blowers.
The centrifugal fan is one of the most widely used fans. Centrifugal fans are by far the most prevalent type of fan used in the HVAC industry today. They are often cheaper than axial fans and simpler in construction. They are used in transporting gas or materials and in ventilation system for buildings. They are also well-suited for industrial processes and air pollution control systems.
The centrifugal fan is a drum shape composed of a number of fan blades mounted around a hub. As shown in the animated figure, the hub turns on a driveshaft mounted in bearings in the fan housing. The gas enters from the side of the fan wheel, turns 90 degrees and accelerates due to centrifugal force as it flows over the fan blades and exits the fan housing.

CHAPTER: 2

CONSTRUCTION

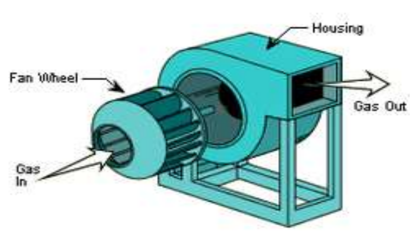


Figure 1: Components of a centrifugal fan

Main parts of a centrifugal fan are:
1. Fan housing
2. Impellers
3. Inlet and outlet ducts
4. Drive shaft
5. Drive mechanism

Other components used may include bearings, couplings, impeller locking device, fan discharge casing, shaft seal plates etc.

Drive mechanisms

The fan drive determines the speed of the fan wheel (impeller) and the extent to which this speed can be varied. There are three basic types of fan drives.

Direct

The fan wheel can be linked directly to the shaft of an electric motor. This means that the fan wheel speed is identical to the motor's rotational speed. With this type of fan drive mechanism, the fan speed cannot be varied unless the motor speed is adjustable. Air conditioning automatically provides faster speed because colder air is denser.

Some electronics manufacturers have made centrifugal fans with external rotor motors (the stator is inside the rotor), and the rotor is directly mounted on the fan wheel (impeller).

Belt

A set of sheaves is mounted on the motor shaft and the fan wheel shaft, and a belt transmits the mechanical energy from the motor to the fan.

The fan wheel speed depends upon the ratio of the diameter of the motor sheave to the diameter of the fan wheel sheave and can be obtained from this equation:

Variable

Variable drive fans may use hydraulic or magnetic couplings (between the fan wheel shaft and the motor shaft) that allow variable speed. The fan speed controls are often integrated into automated systems to maintain the desired fan wheel speed.

An alternate method of varying the fan speed is to use an electronic variable-speed drive to control the speed of the motor driving the fan. This offers better overall energy efficiency than mechanical couplings, especially at greatly-reduced speeds.

Bearings

Bearings are an important part of a fan. Sleeve-ring oil bearings are used extensively in fans. Some sleeve-ring bearings may be water-cooled. Water-cooled sleeve bearings are often used when the fan moves hot gases. Heat is conducted through the shaft and into the oil, which must be cooled to prevent overheating the bearing. Lower-speed fans have bearings in hard-to-reach spots, so they use grease-packed bearings.

Many turbo blowers use either an air bearing or a magnetic bearing.

Fan dampers and vanes

Fan dampers are used to control gas flow into and out of the centrifugal fan. They may be installed on the inlet side or on the outlet side of the fan, or both. Dampers on the outlet side impose a flow resistance that is used to control gas flow. Dampers on the inlet side (inlet vanes) are designed to control gas flow by changing the amount of gas or air admitted to the fan inlet. Inlet dampers (inlet vanes) reduce fan energy usage due to their ability to affect the airflow pattern into the fan.

Fan blades

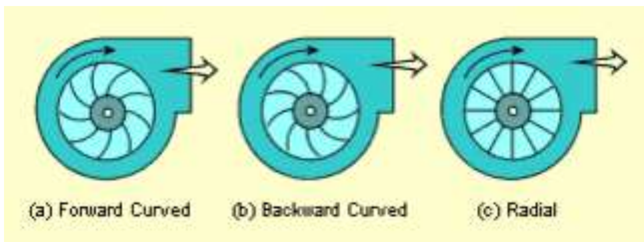


Figure 2: Centrifugal fan blades

The fan wheel consists of a hub with a number of fan blades attached. The fan blades on the hub can be arranged in three different ways: forward-curved, backward-curved or radial.

Forward-curved

Forward-curved blades, as in Figure 3(a), curve in the direction of the fan wheel's rotation. These are especially sensitive to particulates. Forward-curved blades provide a low noise level and relatively small air flow with a high increase in static pressure.

Backward-curved

Backward-curved blades, as in Figure 3(b), curve against the direction of the fan wheel's rotation. Smaller blowers may have **backward-inclined** blades, which are straight, not curved. Larger backward-inclined/-curved blowers have blades whose backward curvatures mimic that of an airfoil cross section, but both designs provide good operating efficiency with relatively economical construction techniques. These types of blowers are designed to handle gas streams with low to moderate particulate loadings They can be easily fitted with wear protection but certain blade curvatures can be prone to solids build-up. Backward curved wheels are often heavier than corresponding forward-curved equivalents, as they run at higher speeds and require stronger construction.

Backward curved fans can have a high range of specific speeds but are most often used for medium specific speed applications—high pressure, medium flow applications.

Backward-curved fans are much more energy efficient than radial blade fans and so, for high power applications may be a suitable alternative to the lower cost radial bladed fan.

Straight radial

Radial blowers, as in Figure 3(c), have wheels whose blades extend straight out from the centre of the hub. Radial bladed wheels are often used on particulate-laden gas streams because they are the least sensitive to solid build-up on the blades, but they are often characterized by greater noise output. High speeds, low volumes, and high pressures are common with radial blowers, and are often used in vacuum cleaners, pneumatic material conveying systems, and similar processes.

CHAPTER:3

PRINCIPLE OF OPERATION

The centrifugal fan uses the centrifugal power supplied from the rotation of impellers to increase the kinetic energy of air/gases. When the impellers rotate, the gas particles near the impellers are thrown-off from the impellers, then moves into the fan casing. As a result, the kinetic energy of gas is measured as pressure because of the system resistance offered by the casing and duct. The gas is then guided to the exit via outlet ducts. After the gas is thrown-off, the gas pressure in the middle region of the impellers decreases. The gas from the impeller eye rushes in to normalize this. This cycle repeats and therefore the gas can be continuously transferred.

Table 1

Differences between fans and blowers

Equipment	Pressure Ratio	Pressure rise (mm H2O)
Fans	Up to 1.1	1136
Blowers	1.1 to 1.2	1136-2066

Velocity triangle

Mainarticle: Velocity triangle

A diagram called a velocity triangle helps us in determining the flow geometry at the entry and exit of a blade. A minimum number of data are required to draw a velocity triangle at a point on blade. Some component of velocity varies at different point on the blade due to changes in the direction of flow. Hence an infinite number of velocity triangles are possible for a given blade. To describe the flow using only two velocity triangles, we define mean values of velocity and their direction. Velocity triangle of any turbo machine has three components as shown:

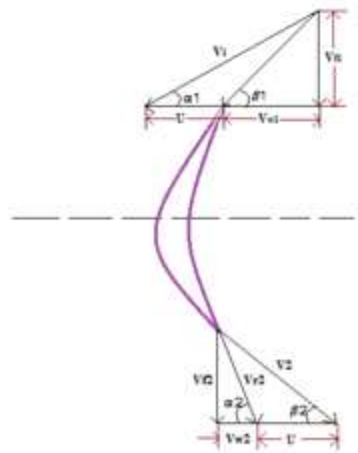


Figure.3 Velocity triangle for forward facing blade

- U Blade velocity
- V_r Relative Velocity
- V Absolute velocity

These velocities are related by the triangle law of vector addition: This relatively simple equation is used frequently while drawing the velocity diagram. The velocity diagram for the forward, backward face blades shown are drawn using this law. The angle α is the angle made by the absolute velocity with the axial direction and angle β is the angle made by blade with respect to axial direction.

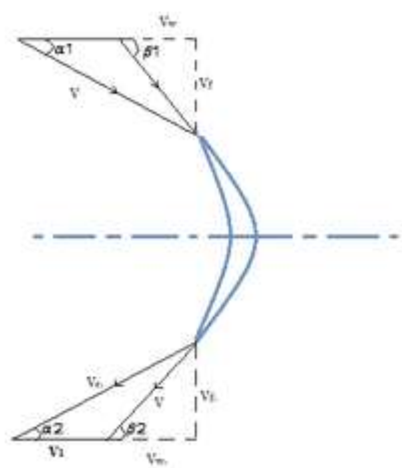


Figure. 4 Velocity triangle for backward-facing blade

Difference between fans and blowers

The property that distinguishes a centrifugal fan from a blower is the pressure ratio it can achieve. In general, a blower can produce a higher pressure ratio. As per American Society of

Mechanical Engineers (ASME) the specific ratio - the ratio of the discharge pressure over the suction pressure – is used for defining the fans and blowers.

CHAPTER:4

RATING

Ratings found in centrifugal fan performance tables and curves are based on standard air SCFM. Fan manufacturers define standard air as clean, dry air with a density of 0.075 pounds mass per cubic foot (1.2 kg/m³), with the barometric pressure at sea level of 29.92 inches of mercury (101.325 kPa) and a temperature of 70 °F (21 °C). Selecting a centrifugal fan to operate at conditions other than standard air requires adjustment to both static pressure and power.

At higher-than-standard elevation (sea level) and higher-than-standard temperature, air density is lower than standard density. Air density corrections must accounted for for centrifugal fans that are specified for continuous operation at higher temperatures. The centrifugal fan displaces a constant volume of air in a given system regardless of air density.

When a centrifugal fan is specified for a given CFM and static pressure at conditions other than standard, an air density correction factor must be applied to select the proper size fan to meet the new condition. Since 200 °F (93 °C) air weighs only 80% of 70 °F (21 °C) air, the centrifugal fan creates less pressure and requires less power. To get the actual pressure required at 200 °F (93 °C), the designer must multiply the pressure at standard conditions by an air density correction factor of 1.25 (i.e., 1.0/0.8) to get the system to operate correctly. To get the actual power at 200 °F (93 °C), the designer must divide the power at standard conditions by the air density correction factor.

Air Movement and Control Association (AMCA)

The centrifugal fan performance tables provide the fan RPM and power requirements for the given CFM and static pressure at standard air density. When the centrifugal fan performance is not at standard conditions, the performance must be converted to standard conditions before entering the performance tables. Centrifugal fans rated by the Air Movement and Control Association (AMCA) are tested in laboratories with test setups that simulate installations that are typical for that type of fan. Usually they are tested and rated as one of four standard installation types as designated in AMCA Standard 210.

AMCA Standard 210 defines uniform methods for conducting laboratory tests on housed fans to determine airflow rate, pressure, power and efficiency, at a given speed of rotation. The purpose of AMCA Standard 210 is to define exact procedures and conditions of fan testing so that ratings provided by various manufacturers are on the same basis and may be compared. For this reason, fans must be rated in standardized SCFM.

CHAPTER:5
LOSSES

Centrifugal fans suffer efficiency losses in both stationary and moving parts, increasing the energy input required for a given level of airflow performance.

Impeller entry
Flow at the intake and its turning from axial to radial direction causes losses at the intake. Friction and flow separation cause impeller blade losses since there is change in incidence angle. These impeller blade losses are also included in the category.

Leakage
Leakage of some air and disturbance in the main flow field is caused due to the clearance provided between the rotating periphery of the impeller and the casing at the entry.

Impeller
Passage friction and flow separation cause impeller losses that depend on relative velocity, rate of diffusion, and blade geometry. Impeller dynamic balancing usually is done on a precision balancing machine, because all energy of vibrational imbalance is lost (for example, this can easily amount to 50% of air-flow loss in poorly-maintained home AC units).

Diffuser and volute
Friction and flow separation also causes losses in the diffuser. Further losses due to incidence occur if the device is working beyond its design conditions. Flow from the impeller or diffuser expands in the volute, which has a larger cross section leading to the formation of eddy, which in turn reduces pressure head. Friction and flow separation losses also occur due the volute passage.

Disc friction
Viscous drag on the back surface of the impeller disc causes disc friction losses.

CHAPTER:6

AFFINITY LAWS
The **affinity laws** (Also known as the "Fan Laws" or "Pump Laws") for pumps/fans are used in hydraulics, hydronics and/or HVAC to express the relationship between variables involved in pump or fan performance (such as head, volumetric flow rate, shaft speed) and power. They apply to pumps, fans, and hydraulic turbines. In these rotary implements, the affinity laws apply both to centrifugal and axial flows.

The laws are derived using the Buckingham π theorem. The affinity laws are useful as they allow prediction of the head discharge characteristic of a pump or fan from a known characteristic measured at a different speed or impeller diameter. The only requirement is that the two pumps or fans are dynamically similar, that is the ratios of the fluid forced are the same.

Law 1. With impeller diameter (D) held constant:

Law 1a. Flow is proportional to shaft speed:

$$\frac{Q_1}{Q_2} = \left(\frac{N_1}{N_2}\right)$$

Law 1b. Pressure or Head is proportional to the square of shaft speed:

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$$

Law 1c. Power is proportional to the cube of shaft speed:

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

Law 2. With shaft speed (N) held constant:

Law 2a. Flow is proportional to the cube of the impeller diameter:

$$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^3$$

Law 2b. Pressure or Head is proportional to the square of the impeller diameter:

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2$$

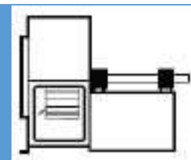
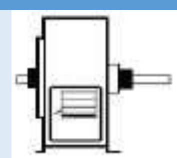
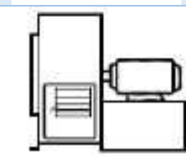
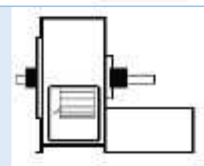
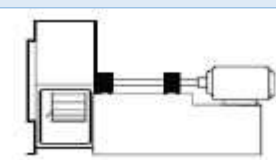
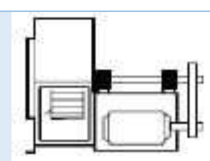
Law 2c. Power is proportional to the fifth of impeller diameter:

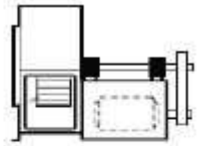
$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^5$$

Where

- Q is the volumetric flow rate (e.g. CFM, GPM or L/s),
- D is the impeller diameter (e.g. in or mm),
- N is the shaft rotational speed (e.g. rpm),
- H is the pressure or head developed by the fan/pump (e.g. psi or Pascal), and
- P is the shaft power (e.g. W).

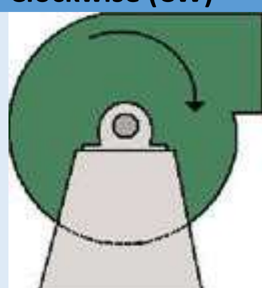
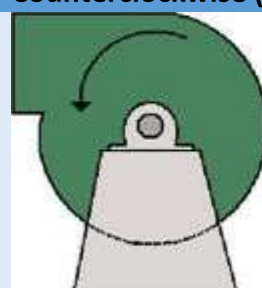
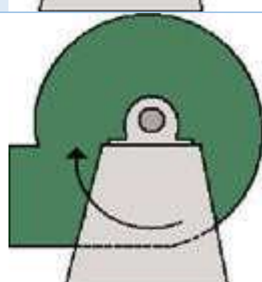
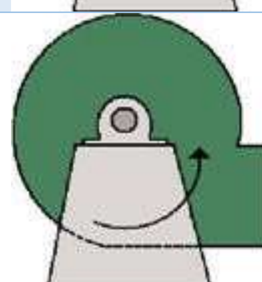
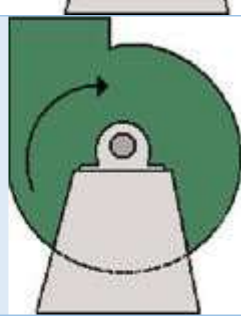
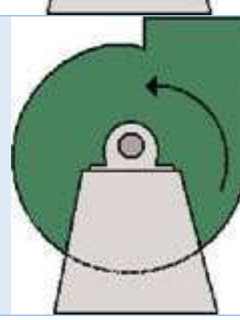
CHAPTER:7
FAN ARRANGMENT

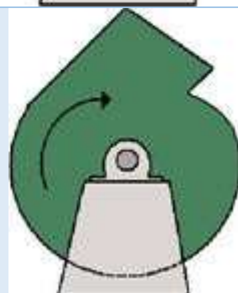
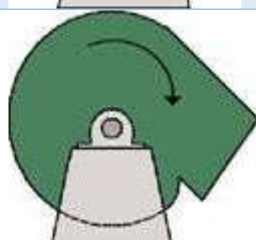
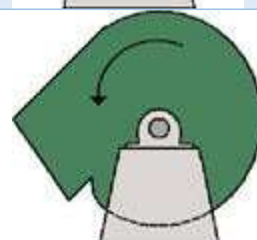
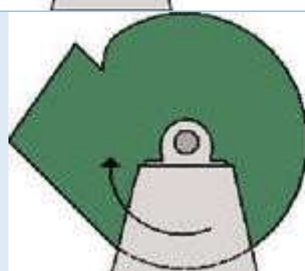
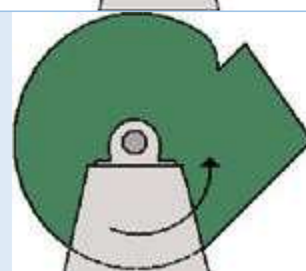
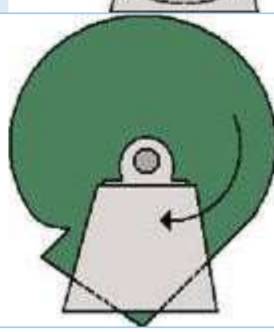
Arrangement #1: This arrangement is very popular for most applications due to the flexibility and the versatility of discharge positions at time of manufacture. Overhung wheel on shaft-and bearing assembly isolates fan bearings from airstream. Normally this arrangement is used for V-belt-drive fans which provides flexibility in fan performance. Motor mounts independently from fan. See Motor Positions Diagram.	
Arrangement #3: Wheel supported between bearings is compact and suitable for clean, dry-air service. Arr. 3 fans are usually sold for V-belt drive fan applications.	
Arrangement #4: Wheel mounted directly on motor shaft to provide the most compact design. Elimination of shaft and bearings for minimum maintenance.	
Arrangement #7: Wheel supported between bearings is compact and suitable for clean, dry-air service. Arr. 7 fans are usually sold for coupling drive fan applications.	
Arrangement #8: Integral pedestal furnished for the motor and coupling. Most flexible of the direct drive arrangements allowing for larger motors, fan sizes and accessories.	
Arrangement #9: An adjustable motor mount featured on this arrangement provides a compact grouping of motor, drive belt and fan. The v-belt drive permits a wide variety of fan speeds. Similar to Arrangement 1, but with motor mounted on side of fan pedestal reducing overall size and field-installation costs.	

Arrangement #10: Compact, packaged design with motor mounted within the fan pedestal. Minimum field installation labor required.	

CHAPTER:8
FAN ROTATION

Fan rotation is determined by looking at the fan from the drive end. The drive end is is the side of the fan housing through which the fan or motor shaft protrudes. For DWDI or Arrangement 3 SWSI fans, it is the side of the fan housing at which the belt drive sheave is mounted.
For centrifugal fans, Discharge is the direction the fan outlet points.
Specification of the rotation and discharge when ordering a housed centrifugal fan is required so that the fan will fit in your system properly.

Discharge Direction	Clockwise (CW)	Counterclockwise (CCW)
Top Horizontal (TH)		
Bottom Horizontal (BH)		
Up Blast (UB)		

Down Blast (DB)		
Top Angular Up (TAU)		
Top Angular Down (TAD)		
Bottom Angular Up (BAU)		
Bottom Angular Down (BAD)		

CHAPTER: 9
FAN PERFORMANCE CURVE

PRESSURE VOLUME CURVE

The most important characteristic of a fan or a system is the relationship that links the primary variables associated with its operation. The most commonly used fan characteristic is the relationship between pressure rise and volume flow rate for a constant impeller speed (RPM). Similarly the relationship between pressure loss and volume flow rate is the most commonly used system characteristic. Fan pressure rise characteristics are normally expressed in either total pressure (TP) or static pressure (SP), with static pressure being the units most commonly used in the United States. The fan volume flow rate (airflow) is commonly expressed in cubic feet per minute, or CFM. Therefore, the system pressure loss and volume flow rate requirements are typically expressed as a certain value of static pressure (SP) at some CFM. The fan "pressure-volume" curve is generated by connecting the fan to a laboratory test chamber. By following very specific test procedures as outlined in the Air Movement and Control Association (AMCA) Standard 210, data points are collected and plotted graphically for a constant RPM, from a "no flow" block off condition to a "full flow" or wide condition.

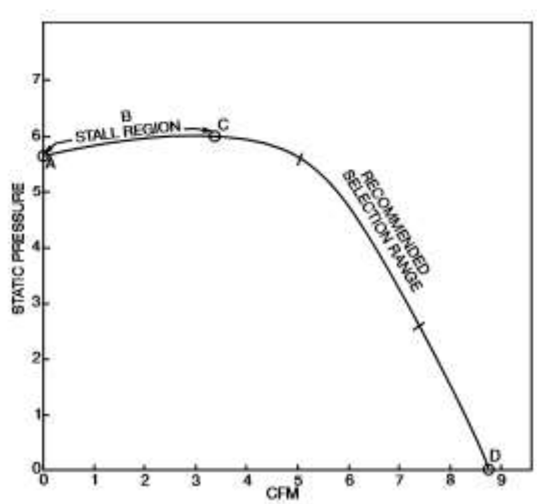


Figure.5 Static Pressure Curve

OPERATING POINT CUVRE

The operating point (point of operation or design point) is defined as the fan pressure rise (SP)/volumetric flow rate (CFM) condition where the fan and system are in a stable equilibrium. This corresponds to the condition at which the fan SP/CFM characteristic intersects the system pressure loss/flow rate characteristics. Figure 6 illustrates this fan/system operating point using the centrifugal fan performance curve.

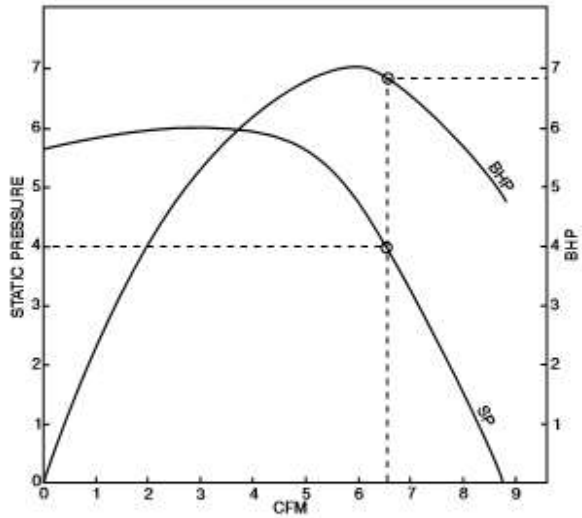


Figure.6 Static Pressure/Horsepower Curve Backward Inclined Centrifugal

The system line is simply a parabolic curve made up of all possible SP and CFM combinations within a given system and is determined from the fan law that SP varies as RPM2. Another fan law states that CFM varies as the RPM. Therefore, we can also say that SP varies as CFM2. Note: Some systems have modulating dampers which will not follow this parabolic curve. Sometimes a fan system does not operate properly according to the design conditions. The measured airflow in the fan system may be deficient or it may be delivering too much CFM. In either case, it is necessary to either speed the fan up or slow it down to attain design conditions. Knowing that the fan must operate somewhere along the system curve, and knowing that it is possible to predict the fan performance at other speeds by applying the following fan laws: 1. CFM varies as RPM 2. SP varies as RPM2 3. BHP varies as RPM3 We can now graphically present an "operating line" between various fan speeds using the fan/system operating point data from Figure 6. This results in new SP curves and BHP curves as shown in Figure 7.

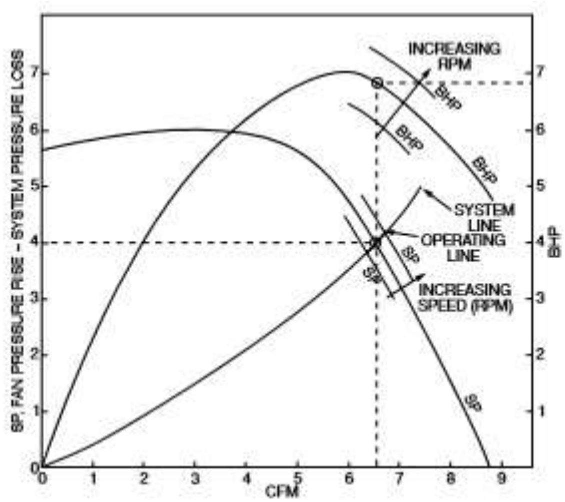


Figure 7. Variable Fan Characteristic Curve, Backward Inclined Centrifugal

These speed changes represent an example of fan control that can be accomplished through drive changes or a variable speed motor. Another way to present an "operating line" is to add a damper, making the system the variable characteristic. By modulating the damper blades, new system lines are created resulting in an operating line along the fan curve. This can be seen graphically in Figure 8.

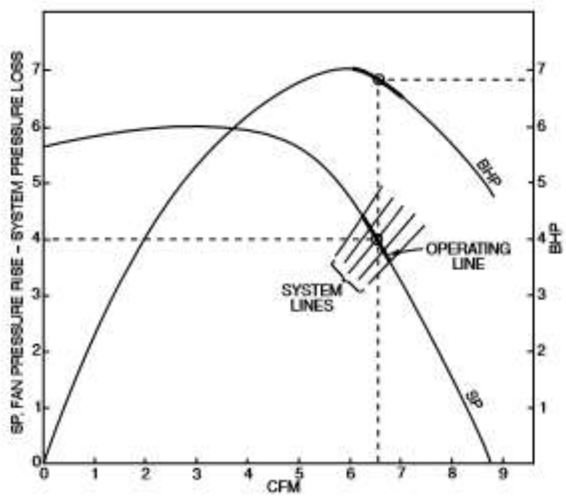


Figure 8. Variable System Characteristic Curve, Backward Inclined Centrifugal

Combining the fan control curve (Figure 7) with the system controlled curve (Figure 8) results in a fan/system controlled curve having an "operating region" as shown in Figure 9.

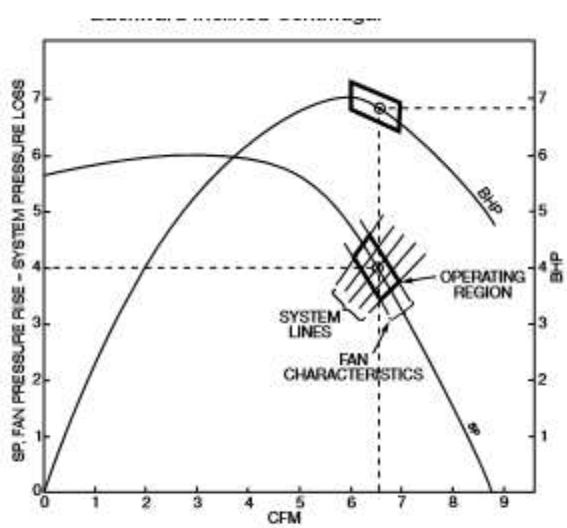


Figure 9. Variable Fan/System Characteristic Curve Backward Inclined Centrifugal

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