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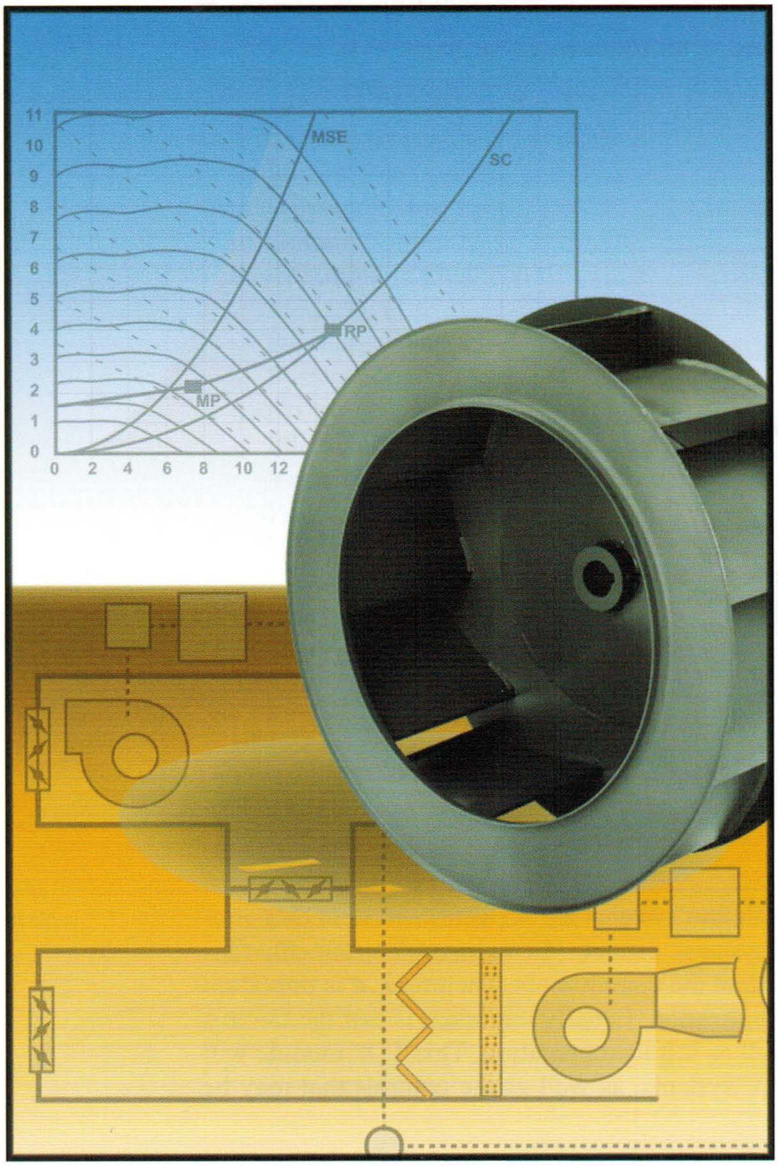


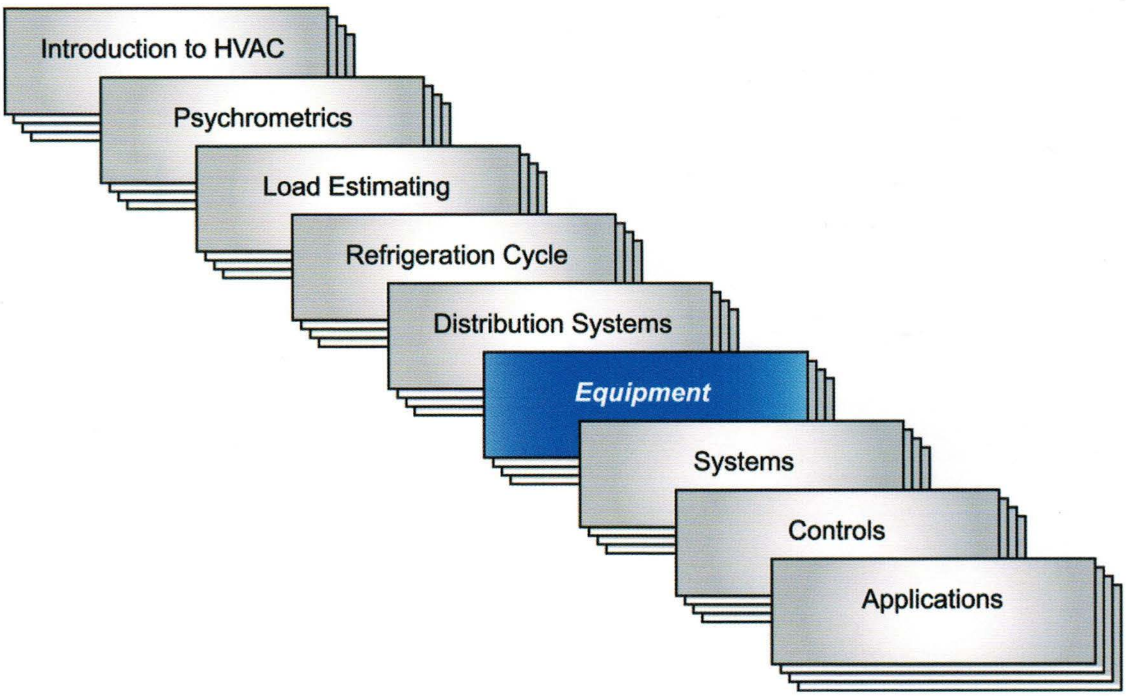
COMMERCIAL HVAC
AIR-HANDLING
EQUIPMENT

Fans in VAV Systems

Technical Development Program

Technical Development Programs (TDP) are modules of technical training on HVAC theory, system design, equipment selection and application topics. They are targeted at engineers and designers who wish to develop their knowledge in this field to effectively design, specify, sell or apply HVAC equipment in commercial applications.

Although TDP topics have been developed as stand-alone modules, there are logical groupings of topics. The modules within each group begin at an introductory level and progress to advanced levels. The breadth of this offering allows for customization into a complete HVAC curriculum – from a complete HVAC design course at an introductory-level or to an advanced-level design course. Advanced-level modules assume prerequisite knowledge and do not review basic concepts.



One of the reasons that VAV (Variable Air Volume) systems are popular is because they provide fan energy savings that constant volume systems cannot. As a general statement, fans consume more energy in a typical HVAC system than the compressors. Therefore, it is important that the correct type of VAV fan be used for the application. Equally important is that the fan in a VAV system is stable at part load operation, as well as full load operation. This TDP module will explain the types of fans that can be used in VAV systems, as well as the controls that may be applied to regulate each.

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Introduction

Building heat loads change throughout the seasons due to variations in outside temperature and shift of solar load patterns. Additionally, building occupant and lighting load patterns change as building space use varies. An air-conditioning system must be able to match these varying load patterns while minimizing the use of energy.

Variable Air Volume Systems (VAV) have the ability to track building load changes and provide fan energy savings that constant volume systems cannot. Since fans may consume as much or possibly even more energy than mechanical refrigeration equipment in a heating, ventilating, and air- conditioning system, VAV systems have become a very popular choice. A typical VAV system is illustrated in Figure 1.

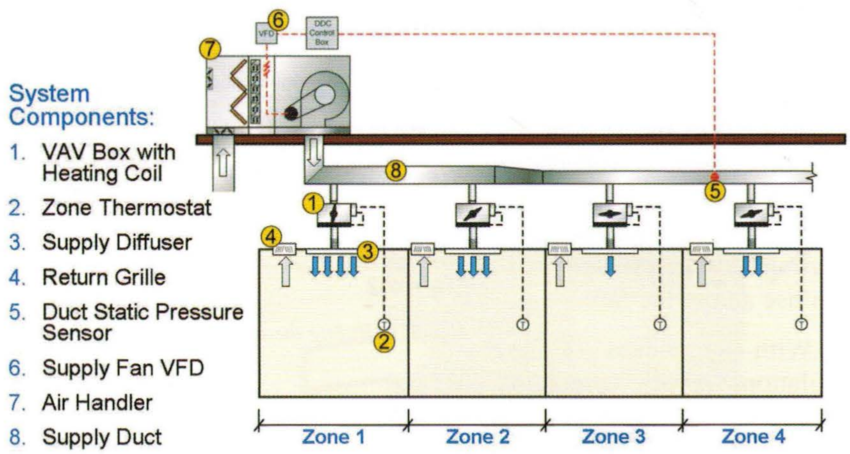


Figure 1

Typical Variable Air Volume System

In a VAV system, a central source such as an air handler or rooftop unit supplies cool air to all the building zones when the building is occupied.

Each building zone is equipped with a VAV terminal. The terminal controls vary the internal damper position to provide just the right volume of air to match the zone cooling load. If any zone should require heat, the zone terminal supply air damper is positioned to the minimum ventilation or heating airflow position, whichever is greater. Typically, the heating airflow position results in about 40 –50 percent of the design cooling cfm. A hot water or electric heater, located on the terminal discharge, is activated upon a call for heat from the thermostat to match the zone heating load.

To complete the VAV air cycle, air exits the zones through return air grilles and flows back to the central unit fan through either a ducted or ceiling plenum type return system.

The opening and closing

of zone terminal dampers to match zone loads has a significant impact on central unit fan energy use and stability, duct system operating pressure, and terminal inlet close-off pressure.

In this TDP, you will learn about the types of fans used in VAV systems and how to predict the part-load impact on the fan/duct/terminal system. You will also learn how to select the proper fan type to match the VAV application, and how to select the proper type of part-load fan volume control to maximize fan energy savings. In addition, this TDP will discuss the application of return and exhaust fans in a VAV system.

Fan Impact

As VAV terminals throttle to match falling zone cooling loads, the pressure drop across the terminals increase. The rise in terminal pressure loss is shown by lines B-B^{1}, C-C^{1}, etc. in Figure 2. This rise in system pressure external to the fan creates a shift in the system curve from A to B to C, etc.

Thus, without changing fan speed, the fan operating point moves from A to B to C, etc. This is called "riding the fan curve." Because of the nature of the fan performance curve, fan static pressure, and thus duct static pressure, a rise at part load in the system resistance causes a decrease in the cfm delivered by the fan. The fan airflow decreases as a function of the increased pressure rise across the fan.

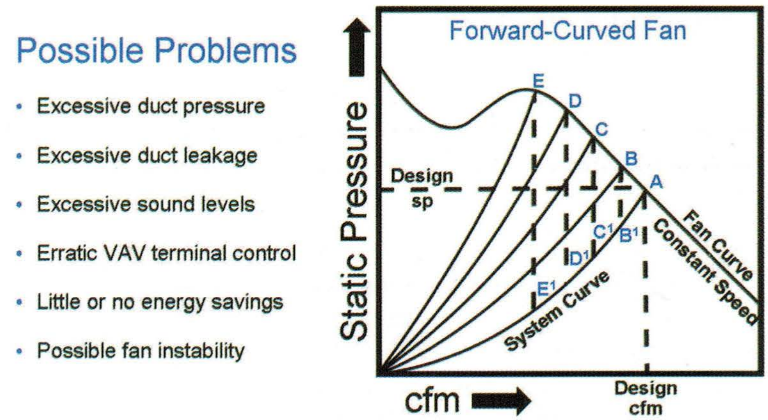


Figure 2

Impact of Riding the Fan Curve

With no means of fan regulation, several problems could develop:

- Excessive duct pressure
- Excessive duct leakage
- Excessive sound levels
- Erratic VAV terminal control
- Little or no energy savings
- Possible fan instability

For a detailed discussion

on the system curve, static pressure, and fan performance curves and their characteristics, see TDP-612, Fans: Features & Analysis.

In order to avoid these problems, and maximize energy savings, some means of fan volume control must be added to the fan. Particular attention needs to be paid to the selection of the fan type and the fan's initial design operating point.

To select the proper type of fan, we must first become acquainted with the characteristics of the types of fans that are most commonly utilized in a VAV system.

Fan Types

A fan is a device used to produce a flow of air. Fans are classified by 2 general types, centrifugal and axial.

Centrifugal Fans

Centrifugal fans are classified according to impeller (wheel) blade design. The most commonly used impeller designs for centrifugal fans for comfort air conditioning are forward- curved, backward-inclined, and airfoil. Impellers and their applications will be covered in this TDP module.

The air is drawn in through one or both sides of the centrifugal impeller and is discharged at a right angle to the fan shaft. A centrifugal fan impeller is usually enclosed in a housing also called a scroll. The air is discharged from the impeller through the outlet in the fan housing. When this housing is mounted inside an insulated cabinet, it comprises the fan section of an air handler. Refer to TDP-611, Central Station Air Handlers for further information.

Figure 3

Centrifugal Fan Configuration

Plenum Fans

When centrifugal airfoil impeller is applied without the housing, and is located inside a cabinet, it is called a plenum fan.

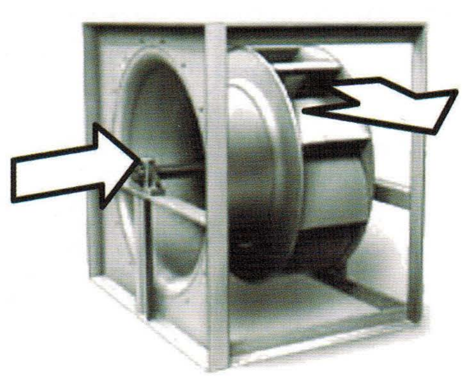


Figure 4

Plenum Fan Configuration.

Axial Fans (In-Line)

In an axial fan, air flows and is discharged parallel to the fan shaft, not at right angles to the fan shaft as with a centrifugal. Axial fans are classified as propeller, tube axial, and vane axial. These fans (with the exception of the propeller) have a tubular configuration, hence the term "in-line." Vane, or tube axial, fans can be driven with an internal direct connected motor or an external shell mounted motor.

There are several variations of the axial fan. These are covered in detail in TDP-612, Fans: Features and Analysis.

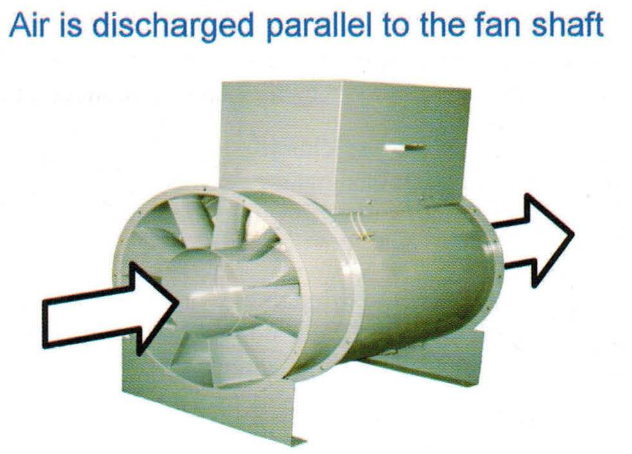


Figure 5

Axial Fan Configuration
Photo courtesy of Barry Blower

Centrifugal Fans

Shown here are the components of a double-width double-inlet (DWDI) fan assembly. This is essentially two single-width fans, side by side, with two inlets and a single outlet or discharge with no partition in the scroll housing. A single width single inlet fan (SWSI) would have a single inlet and take up less space from a width standpoint, but would need to be of greater diameter than the DWDI to move the same volume of airflow. SWSI fans are often applied where it is necessary to mount the fan motor out of the air stream, for example corrosive air. DWDI designs are more common in HVAC equipment.

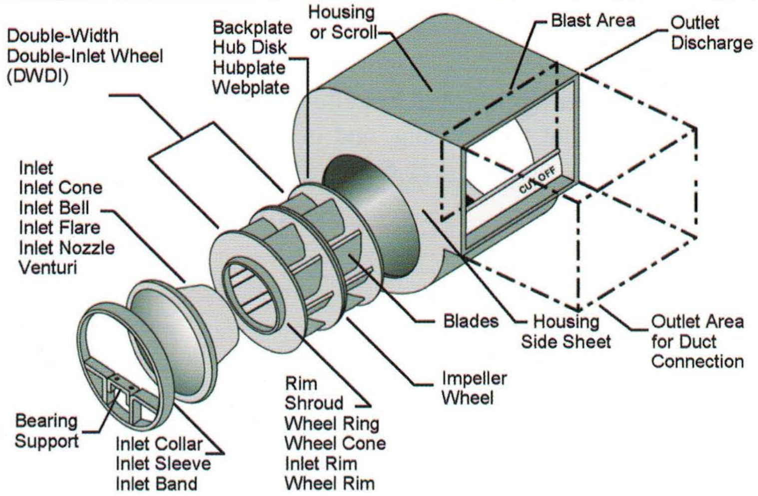


Figure 6

Centrifugal Fan Construction and Terminology

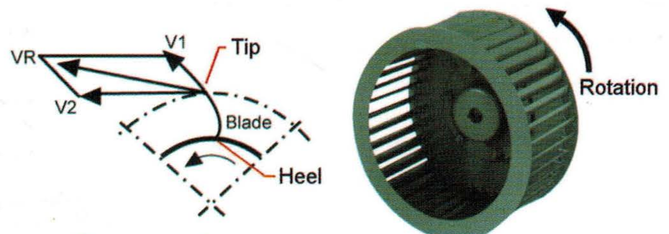
Impeller Design

Forward-Curved

On a forward-curved centrifugal fan, the impeller blades are curved as can be seen here. The air leaves the wheel (VR) at a velocity greater than the tip speed (V2) of the blades. Tip speed is a function of wheel rpm. Since this impeller blade design results in such a large VR, the wheel rpm can be reduced and still produce a comparable airflow to other blade designs. Airfoil and backward inclined, which we will discuss, must be rotated at higher speed. At a given airflow capacity, the forward-curved fan impeller can often utilize a smaller diameter wheel.

> ***Speed***
>
> *The term "speed," when used in this TDP refers to the rpm (revolutions per minute) of the fan impeller.*

Because the forward-curved fan can be rotated at slower speeds and is used for lower static pressures, it is a lightweight design and is therefore less expensive. The fan wheel has 24 to 64 shallow blades with both the heel and the tip of the blade curved forward. This fan is used primarily for low-pressure HVAC applications. Forward-curved fans are best applied operating at static pressures up to 5.0 in. wg.



Characteristics:

- Most commonly used wheel in HVAC
- Light weight – low cost
- Operates at static pressures up to 5 in. wg max
- 24 to 64 blades
- Low rpm (800 to 1200 rpm)

Figure 7

Forward-Curved Wheel Design

Forward-curved centrifugal fans have an overloading horsepower characteristic as the airflow through the fan increases at a constant rpm. This is why they are called overloading type fans.

> ***Because of its appearance,***
>
> *a forward-curved fan wheel is often called a squirrel cage fan.*

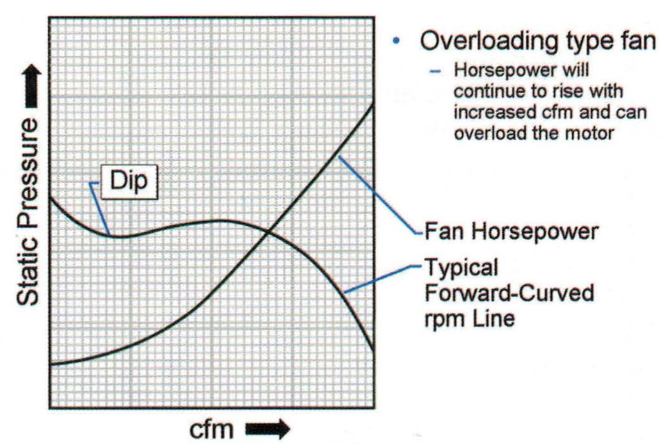


Figure 8

Forward-Curved Fan Characteristics

A typical example of an overloading situation is where a forward-curved centrifugal fan is used for temporary heat duty in an unfinished building. If the ductwork is not completed, the resistance of the duct system may be lower than design, and the fan can deliver more air than required and may eventually overload the motor.

It may be noted that the static pressure-cfm curve of a fan using a forward-curved wheel has a somewhat gradual slope and also contains a "dip." That is how you can recognize a forward-curved application, versus an airfoil or backward-inclined impeller application, which will have a steeper slope and no dip. The dip in the curve of the forward-curved centrifugal fan is to the left of peak pressure. When making fan selection with a forward-curved centrifugal fan, it should be made to the right of the dip to avoid unstable fan operation.

Airfoil and Backward-Inclined

The airfoil impeller is shown below. The airfoil blades have a cross section similar to an airplane wing. Airfoil blades have a thickness that forward-curved and backward-inclined blades do not. A backward-inclined impeller is a thinner (single thickness) bladed airfoil and has an efficiency only slightly less than an airfoil. A backward inclined (BI) impeller will have single thickness blades that are inclined away from the direction of rotation. Fans with airfoil and backward-inclined impellers have the highest efficiency of all centrifugal fans.

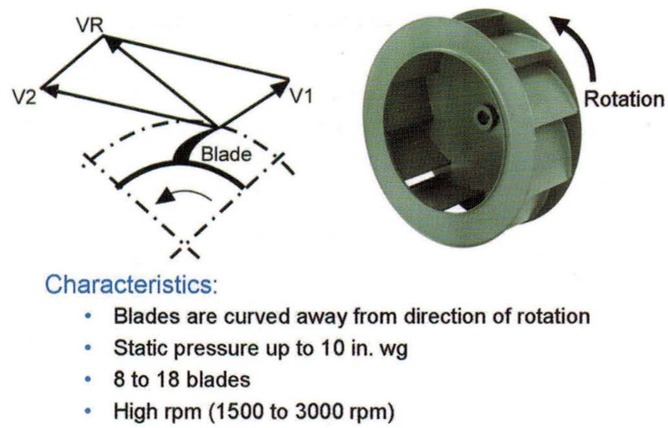


Figure 9

Airfoil Wheel Design

Each airfoil and backward inclined impeller uses approximately 8 to 18 blades inclined backward from the direction of rotation. Because of this, the air leaves the wheel (VR) at a velocity less than the blade tip speed (V2). For a given duty, fans with these impellers will have the highest wheel speed. Fans with airfoil impellers are designed to operate, depending on fan size and manufacturer, at static pressures up to 10 in. wg or higher. Fans with airfoil impellers are not typically used at the static pressures where forward-curved centrifugal fans are the best choice such as less than approximately 5 in wg.

Typically, fans with airfoil impellers are used primarily in large air handlers for systems having relatively high static pressure requirements. Since they are capable of higher static pressures and operate at higher speeds, they are more ruggedly built, which adds to their cost and weight.

Backward-inclined and airfoil fan wheels are considered "non-overloading" because they have the characteristic of almost constant power consumption for the same operating speed (rpm). Some engineers like to use airfoil instead of forward-curved centrifugal fans (when the choice exists) for that reason, even

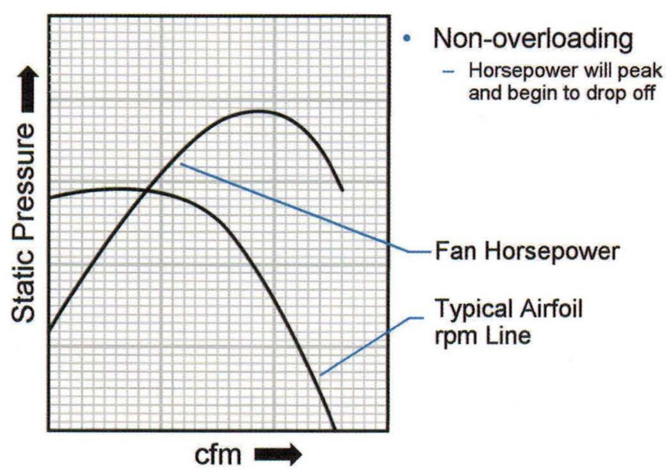


Figure 10

Airfoil Centrifugal Fan Characteristics though they cost more than forward-curved fans. In those areas of applications where either type of fan could be used, it is prudent to make both selections and compare.

Plenum Fan

Plenum fans use non-overloading, single-width single inlet (SWSI) centrifugal airfoil impeller designs constructed of heavy gauge steel with each blade continuously welded to the wheel cone. The fan and its motor operate un-housed within a pressurized plenum or cabinet. When this type of fan utilizes a motor external to the plenum, it is called a plug fan. In a central station air handler, the plenum is the unit casing provided by the manufacturer. Ductwork is connected directly to the plenum without an intermediate transition. In essence, plenum fans use their plenum enclosure as a fan scroll.

Figure 11

Plenum Fan Characteristics
Courtesy of Barry Blower

Plenum fans do not discharge air directly off their impeller and into a discharge duct. The fan pressurizes the plenum it is located in and air is discharged out of the various openings, which are typically field cut into the plenum. For this reason, fan discharge noise is absorbed in the plenum cabinet. This makes the plenum fan ideal for acoustically sensitive fan applications.

Notice the developed inlet cone design to the single inlet airfoil wheel. This allows the fan to efficiently develop static pressure within the wheel.

An important reason that makes plenum fans so popular is that they allow for flexibility in discharge arrangements. The plenum fan may also reduce the space required in the mechanical room for the air-handling unit and the discharge ductwork.

Plenum fans

pressurize a plenum section of the air-handling unit, instead of accelerating the air down a single duct outlet connection. The plenum allows for field-connected duct takeoffs to be attached in multiple directions, saving space and resulting in low discharge sound levels. Any plenum discharge losses resulting from field connections must be added to the external static pressure of the duct system.

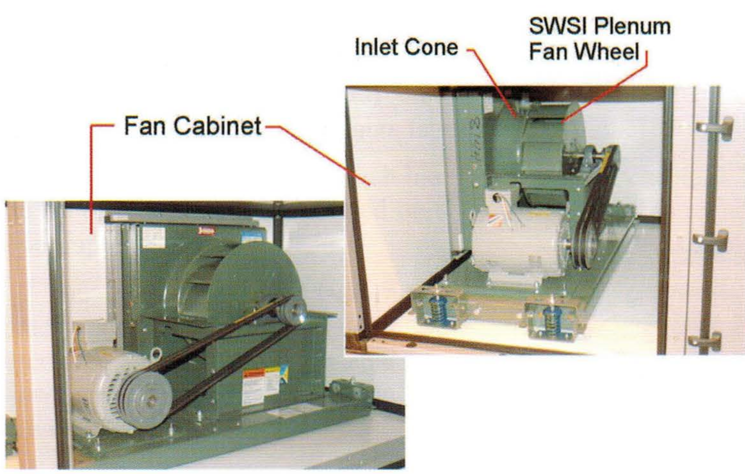


Figure 12

Plenum Fans with Cabinets

Axial (In-line) Fans

Axial (also called in-line) fans are often used for high cfm, low to medium-static applications. The design of the in-line fan allows for direct connection to supply or return ductwork, which can save space in the mechanical room. Axial fans are often applied as return fans as part of a supply-return fan system. They are also be used for exhaust air applications and can even be fitted into factory fabricated air-handling units for supply duty.

- **Use for high cfm applications**
- **In-line space savers with no cabinet**
- **Often used in industrial AC and ventilation applications**
- **Impeller similar to prop fans but blades are more aerodynamic**
- **Often used for return fans in AC applications**

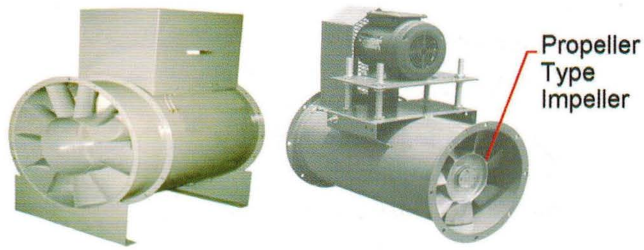


Figure 13

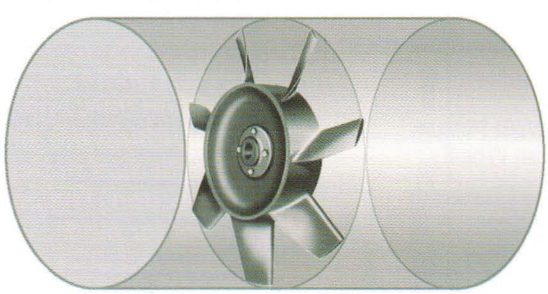
Axial (In-line) Fans

One major difference from centrifugal fans is that air is discharged parallel to the shaft on an axial fan.

Propeller fans are a type of axial fan that is not typically ducted. They are used for moving high volumes of air at very low static pressures. Propeller fans operate at low rpm and are an inexpensive design.

Tube axial fans use a fan design with a propeller type impeller (but with a more aerodynamic configuration) inside a cylindrical tube. They may come with a sound attenuating accessory to help reduce noise levels. Tube axial fans offer a greater efficiency than propeller fans and can be ducted.

Vane axial fan designs are similar to tube axial but incorporate guide (straightening) vanes on the discharge to help redirect the air and improve efficiency. Some vane axial fans have a moveable impeller blade capability. The pitch or angle of the blades can be varied based upon the static pressure and airflow required. The blade angle can be changed manually or automatically.



- Axial Wheel
  - –Air discharged parallel to the shaft
  - –Air is often redirected via straightening vanes making the fan a vane axial

Figure 14

Axial Impeller Design
Photo Courtesy of Barry Blower

The impeller design of an axial fan wheel is similar to a propeller except that the blades are more aerodynamic.

Axial fans are often referred to as in-line or tubular fans. However, not all in-line (or tubular) fans use conventional axial designed impellers.

Fan Volume Control

This section will discuss previous and current methods for fan volume control in VAV systems. Axial fans are used in VAV applications, but the use of centrifugal fans is more common. We will show the current methods of fan volume control using centrifugal fan types as examples.

The ability to control the fan's volume is essential in a VAV system. The goal is to minimize fan static pressure build-up and maximize fan energy savings. Over the years many fan control methods have been utilized and include the following:

- Controllable pitch axial fan
- Modudrive®
- Discharge damper
- System Bypass
- "Riding the fan curve" – no volume control
- Inlet guide vane (IGV)
- Variable frequency drive (VFD)
- Eddy current coupling (also known as eddy drive and eddy current clutch)

Some of these methods are no longer used. However, they are a part of the history of volume control development, and are addressed in this TDP module to provide a historical background to the subject and also to build a foundation for appreciation of the more modern methods. Let's take a look at each of these fan volume control methods, compare their part-load energy-saving characteristics, and see why VFD control has become the dominant choice for HVAC comfort cooling applications.

Controllable Pitch Axial

The CPA (controllable pitch axial) fan varies the pitch of its blades to vary the volume delivered just as a commercial propeller driven aircraft varies the pitch of its blades. While the constant-speed direct-drive motor means that this fan has no drive losses, the complexity in construction of each individual fan blade and its associated linkage resulted in a very expensive fan.

The CPA's fan curve is unlike any other in that it looked like a topographical map with respect to fan efficiency. Its highest efficiency is within a very narrow range, much like the tip of a mountain is depicted on a topographical map.

The fan's varying blade angle generates sound transmitted down the ductwork as the blade pitch changes. However, the CPA fan was an innovative idea.

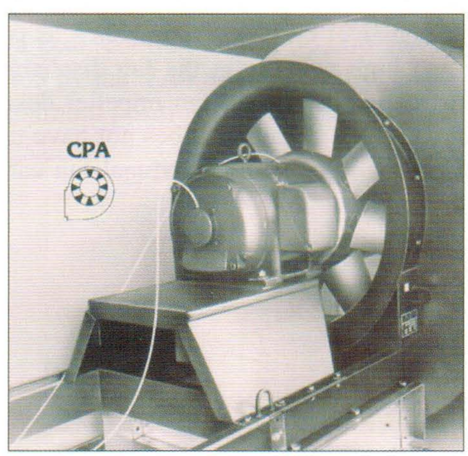


Figure 15

Controllable Pitch Axial Fan

Modudrive®

With Modudrive®, fan speed was altered to match system requirements by varying the pitch diameter of the motor sheave. It could be used with airfoil or forward-curved centrifugal fans. The motor sheave transmitted power to a companion sheave, which then transmitted power to a standard V-belt sheave that transmitted power to the fan sheave. That was a lot of sheaves and mechanical complexity.

The Modudrive® required good maintenance to keep it functioning properly because there were multiple parts and belts in its design.

With all of the belts, companion sheaves, associated idler shafts and bearings, there were higher-than-normal drive losses, especially if everything was not optimally adjusted.

Drive losses

are typically about 3 to 5% in a modern normal belt-driven centrifugal fan arrangement.

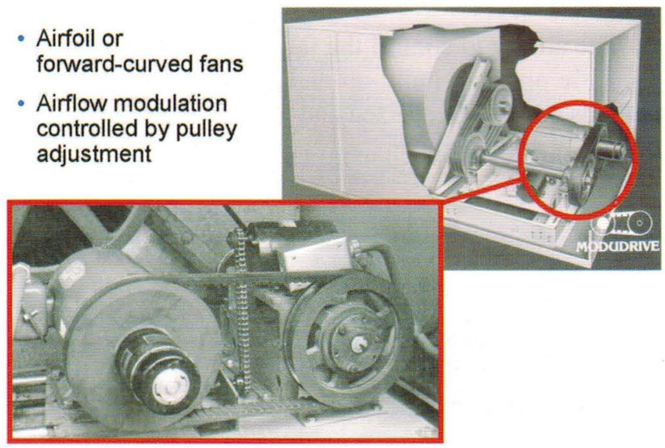


Figure 16

Modudrive®

As a note, both Modudrive® and CPA fans were very competitive from an energy and first cost standpoint and, in their time, they were as reliable as the available alternatives.

Discharge Damper

A discharge damper assembly can be mounted on the discharge of a forward-curved centrifugal fan to accomplish volume control for VAV systems.

By adding a discharge damper to the fan, the build up of excess static pressure is absorbed at the fan rather than at the VAV terminals – as is the case with riding the fan curve. Potential high pressures and high velocity at the fan discharge require the damper components to be of fairly heavy construction. The only fan impeller design that can be used with discharge dampers is a forward-curved type because it does not have the higher static pressure generating capabilities of the airfoil wheel. An airfoil centrifugal fan can generate too much static pressure for discharge dampers to handle.

Control is fairly straightforward in that a static pressure controller will reposition the damper blades as needed to lower the system static pressure, which will result in a build up of static pressure at the fan discharge. The operating point will then ride up the rpm curve to a lower airflow along the constant rpm line. Static pressure on the system side of the discharge dampers in the ductwork will remain at or near the controller's set point.

This is not a very common method of airflow control any longer because the damper tends to create noise and fan instability as the damper blades close while velocity and pressure increase. In addition, the discharge damper will add an additional pressure drop and potential system effect to the system, which will lower overall efficiency.

Dampers are typically mounted close to or right off the fan discharge.

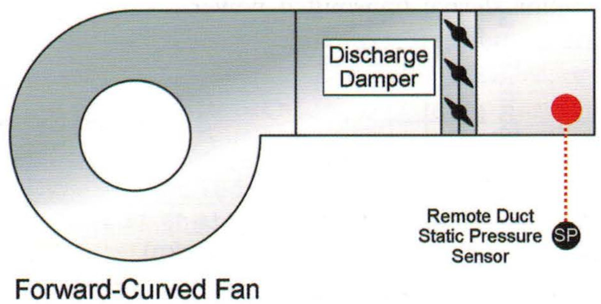


Figure 17

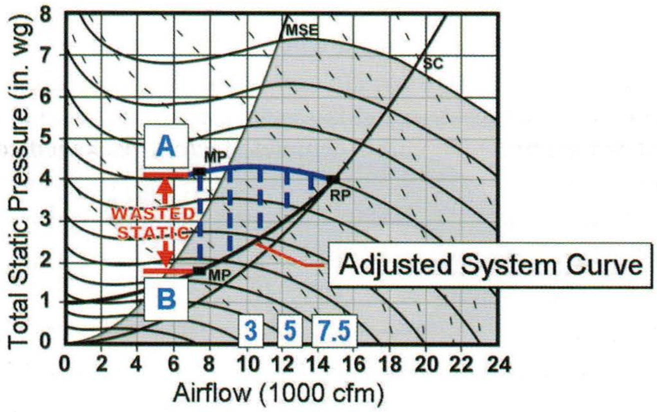
Discharge Damper Installation

A fan discharge damper is available in parallel blade design (shown) or opposed blade design. Parallel blade designs have excellent control near the "top end" of the volume operating range (75 to 100 percent of full volume.) Opposed blade design offers good control over a broader range of airflow than parallel and provides an even distribution of air downstream from the damper.

> *"System effect"*
>
> *is a term that refers to an increase in system static as a result of an inlet or discharge connection that is not uniform. A discharge damper creates system effect due to its close location to the fan discharge. For a complete discussion on "system effect", refer to TDP-612, Fans: Features and Analysis.*

Example: A VAV system has a 15,000 cfm cooling airflow with a 7,500 cfm minimum. A minimum static of 1 in. wg is maintained by the duct static pressure controller in this example. Later in the speed control example, we will use 1.5 in. wg as the set point.

In the examples that follow, we will assume a peak cooling cfm of 15,000 and a minimum cfm of 7,500 that defines the lowest airflow required by the VAV fan. Typically, this lowest cfm occurs when the VAV terminals throttle to their minimum position for ventilation or heating duty.



Legend

\ - rpm \ bhp **MSE** - Max Static Eff. **SC** -System Curve **RP** - Rated Point = 15,000 rpm = 1010 bhp = 17.8 Class II Max. rpm = 1217
rpms (∗100, L to R): 3 4 5 6 7 8 9 10 11 12 13
bhps (L to R): 3 5 7.5 10 15 20 25 30 40
Note: Shaded Area – Recommended Operating Range
MP – Minimum Point (VAV Applications) = 7,500 cfm

Figure 18

Discharge Damper Characteristics

The minimum operating point with discharge dampers requires just over 7.5 bhp to operate (see Point A on chart). If this were a system with a VFD, it would require less than 3 bhp (see Point B on chart).

This method is very inefficient in that it requires the discharge damper to absorb a considerable amount of static pressure at the minimum airflow set point.

On the fan curve here are some important points to understand:

RP (rated point) is the intersection of the system curve and the fan rpm line. The RP defines the resulting cfm. In our examples, RP is at full flow cfm and design static.

MSE (Maximum Static Efficiency) can be thought of as the percentage of input power that is realized as useful work in terms of static pressure. It is best to select the fan to the right of the MSE line (not to the left), especially for a VAV fan that will not be at the peak cfm often. That way, when the cfm is reduced, the fan will still be near the MSE line, which is desirable.

MP (minimum point) means the projected minimum cfm for a VAV fan. This value corresponds to the sum of the individual terminals minimum cfm, often about 40-50 percent of peak. In this TDP module example, the MP value is 50 percent, or 7500 cfm.

Notice the adjusted system curve. VAV systems that utilize a fan modulation device such as a discharge damper, variable inlet vane, or VFD are controlled by duct-mounted static pressure sensor. This sensor is typically set to maintain about a 1 to 1.5 in. wg static pressure at its location regardless of airflow. The adjusted system curve will reflect this valve as the minimum point.

System Bypass

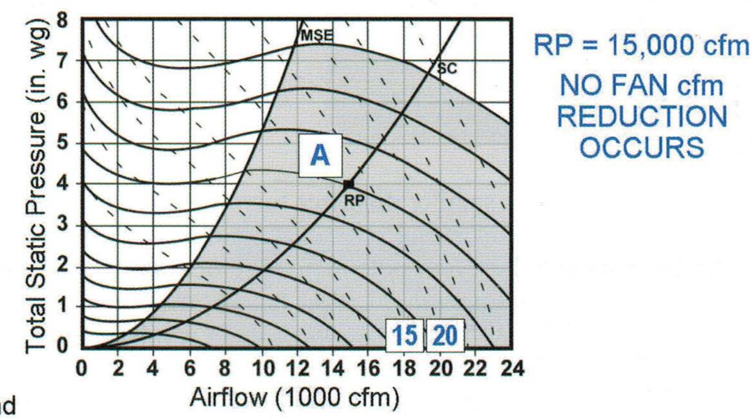
Many smaller multiple zone applications (approximately 20 tons per unit and under) utilize packaged heating and cooling units that incorporate a modulating bypass of the supply air as part of the control strategy. These systems are provided with a complete factory-packaged control system in addition to the bypass designed to provide multiple zones of temperature control using a low cost, single zone, constant volume heating and cooling packaged rooftop unit, VPAC, or split system. Packaged rooftop units (RTUs) are most often used.

Unneeded air at the zone level is sent through the bypass using either a ceiling return air plenum or a ducted return. While these systems are called variable volume and temperature, the fan produces a constant volume of air. For more information on this system, consult TDP 704 Variable Volume and Temperature.

Example: 15,000 cfm cooling with no minimum cfm because of bypassed air.

Notice the operating point with bypass damper(s) always requires between 15 and 20 bhp (Point A) because it is at full fan airflow even at part loads.

> ***With a plenum type bypass,***
>
> *there is often a small reduction in system resistance such that a small increase in fan airflow during bypass mode can occur.*



Legend

- rpm - bhp **MSE** - Max Static Eff. **SC** -System Curve **RP** - Rated Point = 15,000
rpm = 1010 bhp = 17.8 Class II Max. rpm = 1217
rpms (*100, L to R): 3 4 5 6 7 8 9 10 11 12 13
bhps (L to R): 3 5 7.5 10 15 20 25 30 40
Note: Shaded Area – Recommended Operating Range

Figure 19

System Bypass Fan Characteristics

No Volume Control (Riding the Fan Curve)

The simplest form of fan modulation is to ride the fan curve. As the VAV box dampers modulate, the static pressure in the system changes, which varies the airflow produced by the fan. The VAV box dampers absorb the additional static pressure generated and shift the operating point up the rpm curve.

> *"Riding the curve"*
>
> ***for a centrifugal fan means to move backward on the constant rpm line as a result of system pressure build-up. This results in less airflow and lower fan bhp.***

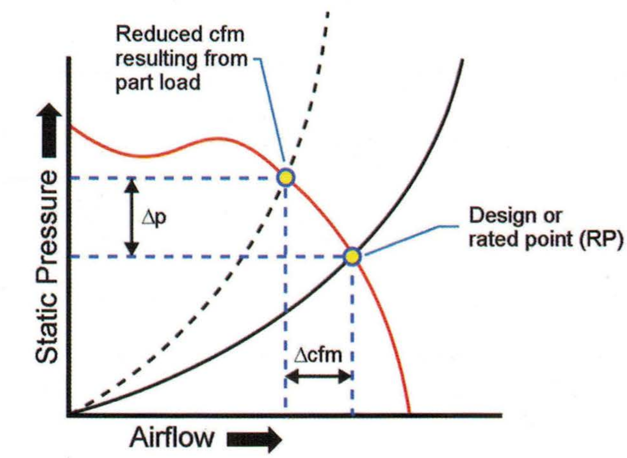


Figure 20

Riding the Fan Curve

It is better to use a smaller diameter forward-curved fan wheel when riding the fan curve because it will place the operating point further to the right on the performance curve and keep any performance point away from potential surge or stall. It is also best to ride the fan curve over a fairly narrow range of operation. Larger ranges of airflows will require a larger static pressure to be absorbed by the VAV terminals, which can result in over pressurization, uncontrollable airflow and/or velocity-related noise issues.

Again, this method is not recommended for backwardly-inclined, airfoil, and plenum fans because of their steep fan curve and large static pressure capability.

Example: 15,000 cfm peak cooling, 7,500 cfm minimum.

The minimum operating point requires just over 7.5 bhp to operate (Point A). If this were a system with a VFD it would require less than 3 bhp (Point B).

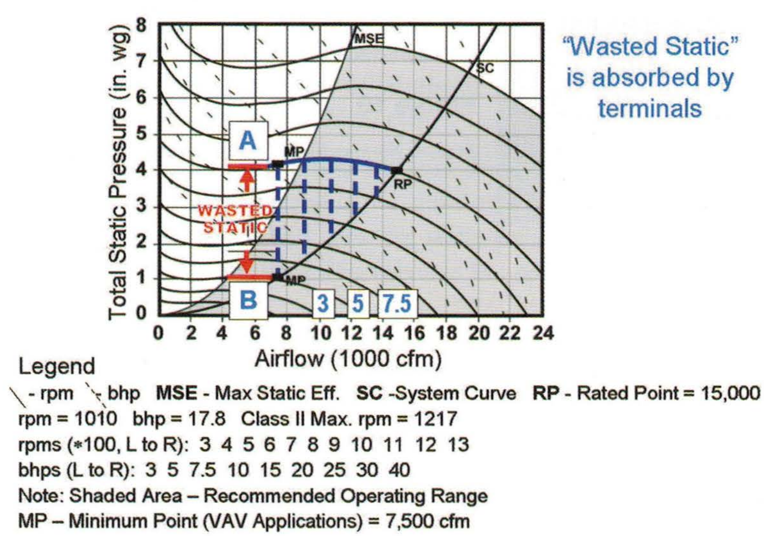


Figure 21

Riding the Fan Curve Characteristics

This method is very inefficient because it requires the VAV box dampers to absorb a considerable amount of static pressure at the minimum airflow set point.

Note

The bypass system is always at or near peak bhp, so it is not represented in this chart.

% Air flow	Airflow (cfm)	bhp	% bhp
100%	15,000	17.8	100%
90%	13,500	15.7	88%
80%	12,000	13.3	75%
70%	10,500	12.0	67%
60%	9,000	9.81	55%
50%	7,500	8.0	45%

Figure 22

Horsepower Reduction Chart

This chart summarizes the bhp reduction from full airflow of discharge dampers and riding the fan curve. While the first cost of those systems will be low, their overall efficiency is poor relative to other fan modulation methods, such as speed control.

Inlet Guide Vanes

Inlet guide vanes

are another method of fan volume control. A set of moveable vanes mounted on the fan inlet reduces airflow by restricting the fan inlet area. IGV design produces a positive whirl to the inlet air, which allows for more efficient fan volume control than previous methods mentioned so far.

Inlet Guide Vanes (IGV) can be applied to most fans including airfoil or backward-inclined. DWDI fans must have inlet vanes on both sides of the fan, which must be equally balanced to prevent unwanted vibration problems.

IGVs also require one or more actuators connected by linkage to the vane assembly.

As the vanes close, reducing the inlet area, fan noise levels will increase due to the higher inlet velocities. Unless properly maintained, they may bind after time. This is the case especially on systems that may have narrow operating bands, such as with forward-curved fans, because they may not have the requirements for much actual vane movement.

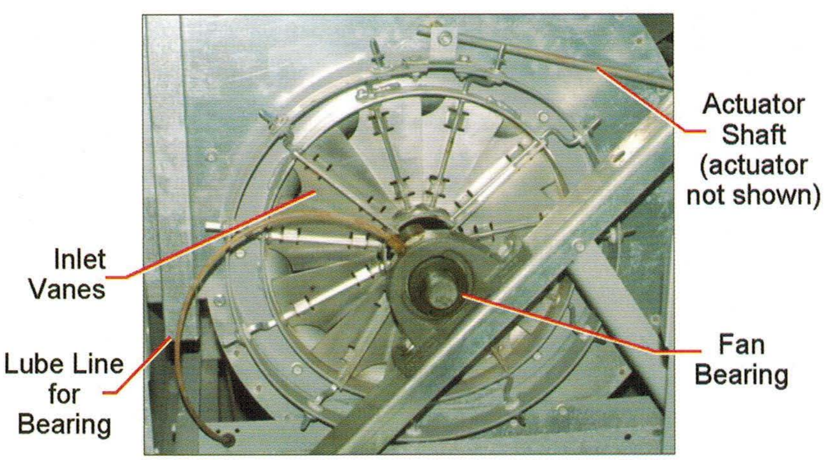


Figure 23

Inlet Guide Vane Configuration

There is also an added resistance that inlet vanes impose on the fan that decreases their overall efficiency.

In effect, the inlet guide vanes create a whole new fan curve for each vane opening position as seen in Figure 24.

Figure 24 shows the effect of inlet guide vanes on bhp. Assuming the design cfm for the job is 80,000, the bhp at 70 percent airflow (56,000 cfm) is about 66 percent.

Effect of Inlet Guide Vane Position on bhp

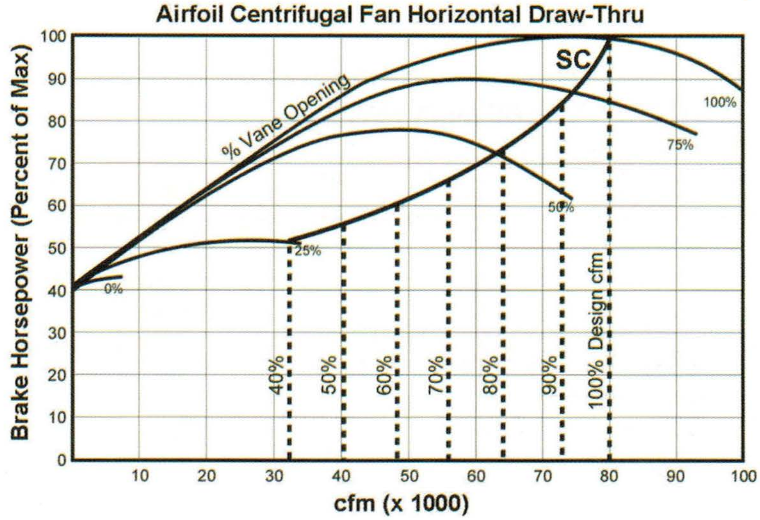


Figure 24

Inlet Guide Vanes bhp Chart

Figure 25 shows that the guide vanes have reduced the static pressure generated by the fan at 70 percent of design airflow from about 87 percent to 30 percent. This avoids unnecessary pressure and noise buildup in the air system in addition to the energy it saves.

Effect of Inlet Guide Vane on Static Pressure

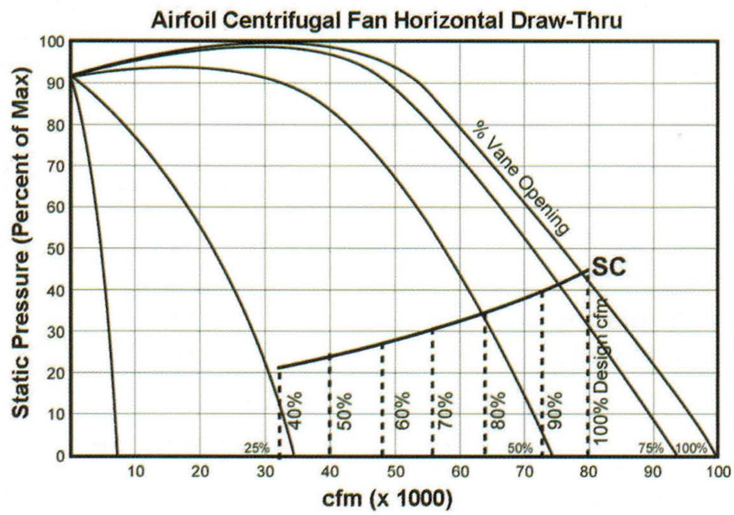


Figure 25

Inlet Guide Vane Static Pressure Chart

Variable Frequency Drives

A Variable Frequency Drive (VFD) is a very popular method of airflow control because drive technology advances have led to increased efficiency and lower costs. Reasonable first cost, soft start, and other VFD attributes such as high power factor (the ratio of active power to apparent power), allow the system to operate as efficiently as possible.

Soft start

is a VFD feature which means the fan is slowly ramped up to full speed. This minimizes mechanical stresses and keeps inrush current lower.

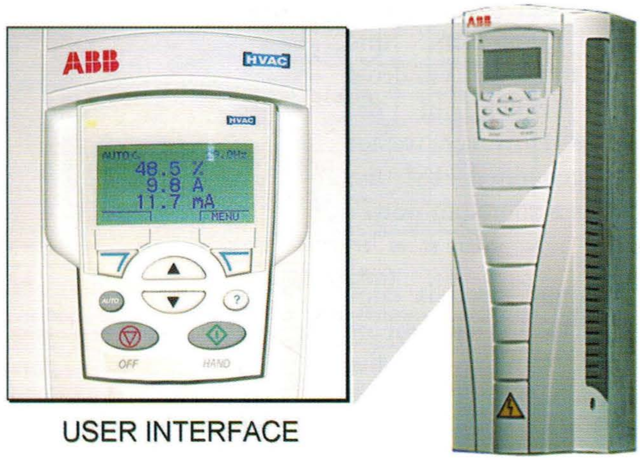


Figure 26

Variable Frequency Drive

The energy savings alone often makes the selection of VFDs the best choice, but sound is another factor. With all of the previous airflow modulation methods discussed, none slowed the fan speed down, and most had some increase in velocity and static pressure at the air terminal in the system. That increase will also dramatically affect the amount of airflow noise in those systems.

With a VFD-controlled fan, there is no increase in velocity or noise because the fan only operates at a point required by the system. Lower horsepower consumption, lower acoustical noise, and soft start capability to reduce demand and drive stresses all make the VFD a superior choice. In addition, energy codes such as ASHRAE 90.1 require the use of VFDs for variable volume systems.

VFDs do have the potential to transmit harmonics back into the power system, which may cause interference with communications or other power- sensitive devices.

A harmonic study

can be performed on the electrical system prior to design completion to determine if harmonics are a concern. If so, proper line filtration or isolation can be specified prior to installation. Many drive manufacturers incorporate these devices directly into the drive itself, thereby eliminating the need to have them field supplied and installed.

Eddy Current Couplings

Eddy current couplings are sometimes used for fan volume control. They can take up extra space inside the fan cabinet, and thus may not be able to fit in some fan cabinets. In essence, the eddy drive varies the rpm of the fan.

The inner component, which contains the electromagnets, is fixed to the shaft to run continuously.

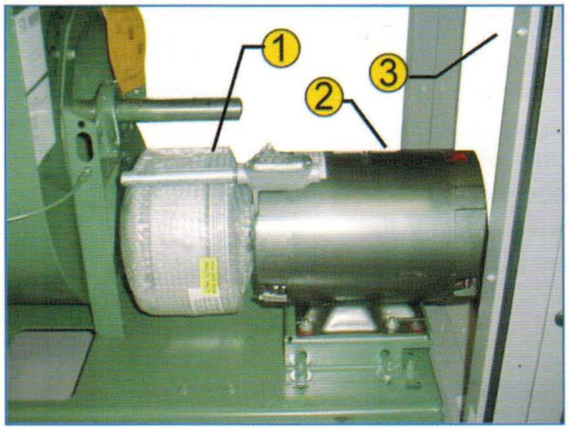


Figure 27

Eddy Current Coupling

As current is applied to the electromagnets, eddy currents are created on the outer drum-shaped component, which causes it to turn. The outer "drum" transfers this power to the fan via drive belt(s) riding in the sheaves. However, waster heat generated by the drum is a source of power loss.

Output speed is controlled by the amount of current applied to the electromagnets. Typically, the shaft can be locked when full drive rpm is required so that constant power is not needed to keep the eddy current coupling engaged.

An eddy current coupling requires a motor starter so it cannot take advantage of a soft start, like a VFD. Just like the brakes on a motor vehicle, energy is required to slow or stop the drum. In this case, it is electrical energy. The braking heat must be dissipated. Heat created by magnetic slip on the outer drum is transferred directly to the airstream.

An eddy current coupling

is also known as an eddy current clutch because the principle of operation is similar to a clutch in a car. They are also called eddy drives.

Eddy current couplings have losses that are equal to output slip times output torque. As output speed decreases, the total losses increase. However, eddy current couplings, when applied to a variable torque load like a fan in a VAV system, are helped by the fact that required torque drops by the square of the speed reduction. The overall efficiency of the VFD is better than the eddy current coupling, however.

Eddy current drives do not create electrical harmonics or require potential harmonic attenuation. However, VFDs are still considered the best method of VAV fan airflow reduction.

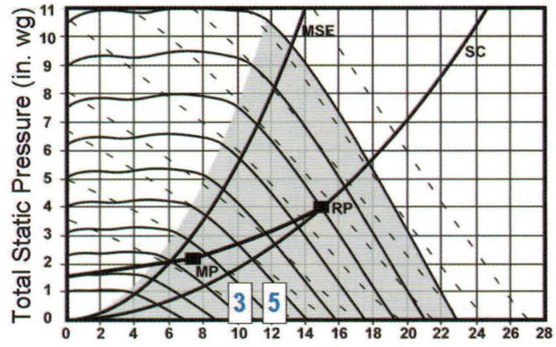
VFDs:

1. Are more efficient
2. Can easily be tied to the building automation system
3. Can report power monitoring information
4. Are more readily available
5. May have a cash payback incentive from the utility company

VFD Example: 15,000 cfm cooling, 7,500 cfm minimum with 1.5 inch minimum static pressure maintained by a static pressure controller in the main duct.

The bhp values

shown in all examples are fan bhp values, i.e., they do not include drive losses. Due to the inefficiency created by the eddy current losses, which are in addition to standard drive losses, the actual motor bhp values (hence energy consumption) at the part-load points for an eddy coupling application are higher than a VFD.



Legend

\ - rpm \ - bhp **MSE** - Max. Static Eff. **SC** -System Curve **RP** - Rated Point
rpm = 2210 bhp = 18.8 Class II Max. rpm = 2442
rpms (*100, L to R): 8 10 12 14 16 18 20 24 26
bhps (L to R): 3 5 7.5 10 15 20 25 30 40
Note: Shaded Area – Recommended Operating Range
MP – Minimum Point (VAV Applications) = 7500 cfm

Figure 28

Speed Control with a VFD

% Air flow	Airflow (cfm)	System Static Pressure*	Fan rpm VFD	bhp** VFD	% bhp*** VFD	% bhp Discharge Dampers and Riding the Fan Curve
100%	15,000	4.0	2210	18.8	100%	100%
90%	13,500	3.525	2000	14.1	75%	88%
80%	12,000	3.1	1810	10.8	57%	75%
70%	10,500	2.725	1625	7.6	40%	67%
60%	9,000	2.4	1475	5.9	31%	55%
50%	7,500	2.125	1300	4.1	22%	45%

* System static is determined using the second fan law
** bhp is from the fan curve
*** % bhp is the percent as compared to the maximum bhp

Figure 29

Fan Modulation Methods and Their Effect on bhp

This chart summarizes the bhp requirements of VFD speed control versus discharge dampers, riding the fan curve. The resulting fan bhp values at various percentages of full airflow are the lowest for the VFD. This results in energy savings at the part load points. The amount of energy saved will be a function of the total run hours, cost of electricity, and the part load profile.

As seen in Figure 30, the most efficient means is with VFD speed control, especially at part load. At 50 percent airflow the discharge damper and riding the curve fan methods resulted in the percent of maximum bhp at 45 percent. For VFD speed control, we see that the percent of maximum bhp is now 22 percent.

As a note, the discharge damper performance line is shown slightly above the riding the fan curve line to represent the system effect of the discharge damper resting directly on the fan discharge.

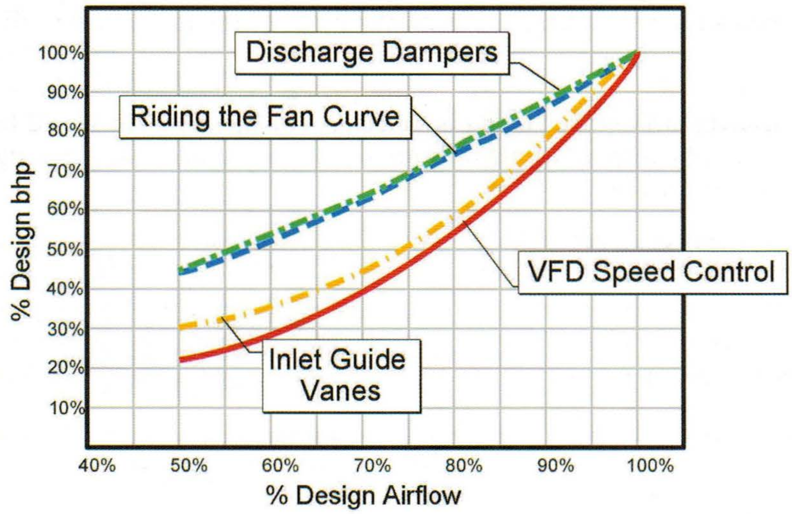


Figure 30

Fan Modulation Methods

Lastly, this example used centrifugal fans to demonstrate part load fan modulation methods. Axial fans are also typically controlled by VFDs when used in VAV systems.

Fan Stability

A stable fan selection is one that is not easily affected by a temporary disturbance or change to the system. A change to the system can be as simple as a damper position change or a door closing in the space that the system serves.

Unstable operation can result in fan surging, which can be heard or felt as pumping or pulsating, or fan stall, which can cause separation of the flow from the fan blades. The result can be the generation of noise and vibration, and possibly erratic fan operation.

The simple solution to avoiding stall or surge is to select a fan so its operating point(s) are on the negatively sloping portion of the pressure/volume curve.

> ***Fan Selection***
>
> *Typically, manufacturers show a valid, recommended operating region with shading or coloring. Not only does the fan selection need to be in the valid region at full load, but on VAV systems at part load as well.*

Fan Selection

When selecting a fan for a VAV system, it is important to consider the system curve and the possibility of fan surge or stall.

At the design VAV system airflow or rated point (RP), the fan should be selected to the right of the maximum static efficiency line on the fan curve. Then, fan operation should be checked at the expected minimum VAV system airflow.

A fan's part-load minimum operating point for a VAV system will be the sum of the minimum air terminal damper cfm positions. A worst-case scenario will occur when all building zones are in the cooling mode and all zone dampers are throttled back to their minimum ventilation positions. If this forces the fan operating point to the left of the fan's stable operating region, then a different airflow fan should be selected. Another solution would be to increase the system minimum airflow to move the minimum operating point back to the right and within the fan's stable operating region.

Some engineers set the minimum air terminal damper cfm positions to match the zone heating needs. This is typically about 50% of the zone cooling cfm needs. Under these conditions, the fan should be checked to see that the minimum fan cfm operating point falls within the fan's stable operating region.

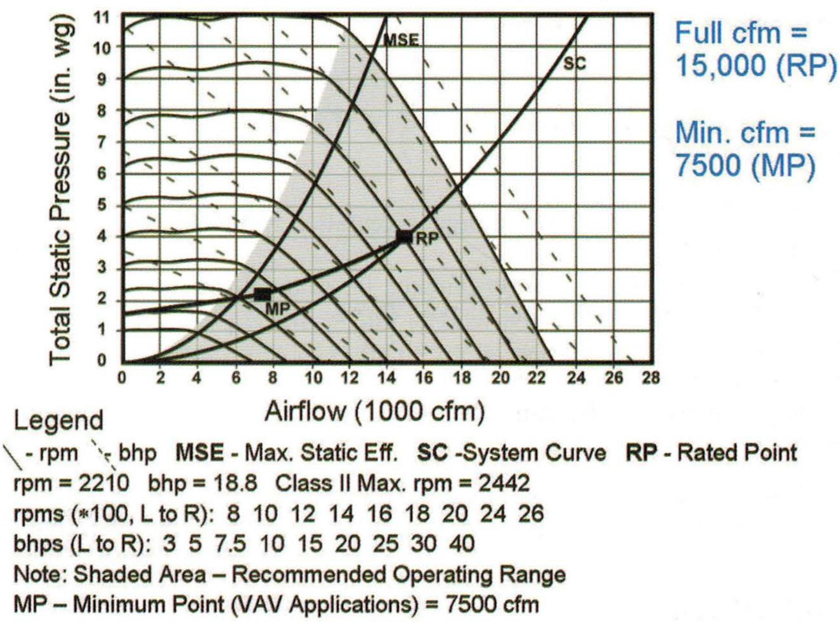


Figure 31

Typical VAV VFD Fan Characteristics

Some manufacturers' software allows you to input the minimum airflow and static pressure set point and plot out the actual VAV system curve. The figure shows the input screen from Carrier's AHUBuilder software program. The box in the lower left highlights where the minimum airflow and minimum static pressure set point can be input.

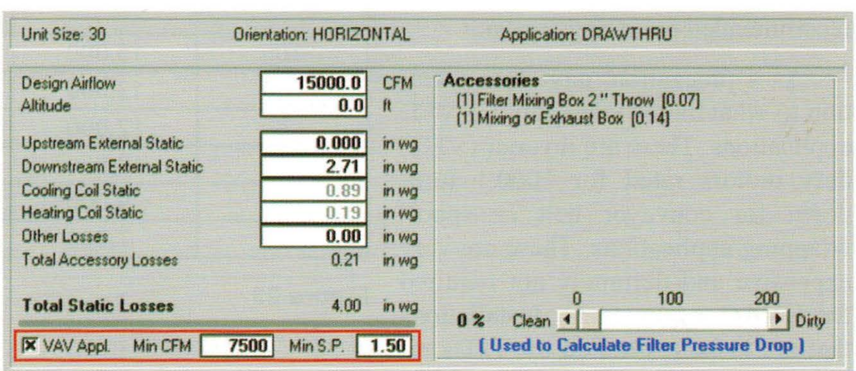


Figure 32

Input Screen – Fan for VAV Duty

It is important to note that the static pressure set point (Min SP) in this example is 1.5 in. wg and the VAV system curve is developed from that point forward, so the adjusted system curve does not start at zero static pressure in. wg, it starts at 1.5 in. wg. This way, the effects of filter loading; coils and ductwork are in addition to the minimum static pressure by the static pressure sensor.

After all of the parameters have been input and the calculation performed, a fan curve with the system curve (labeled SC) indicating the unit's rating point (labeled RP) will be generated.

When the VAV option is selected, a second curve will be plotted indicating the VAV system "adjusted" curve, which includes the minimum operating point (labeled MP) as defined by the user. To avoid part-load instability, the MP operating point should be to the right of the fan curve's minimum operating range.

If a VAV fan curve cannot be calculated in software, then the following manual procedure can be used:

$$\left(\left(\frac{cfm_1}{cfm_{DESIGN}}\right)^2 * \left(P_{S\,DESIGN} - P_{S\,MIN}\right)\right) + P_{S\,MIN} = P_{S1}$$

For example, here is the manual calculation at 50% of design airflow.

$$\left(\left(\frac{7{,}500}{15{,}000}\right)^2 * (4 - 1.5)\right) + 1.5 = 2.125 \text{ in. wg}$$

If we manually calculated the other cfm points, we could reproduce the data generated by the selection software.

This table illustrates the results of a manual calculation method for a system with a centrifugal airfoil fan at a cooling design of 15,000 cfm and a system static pressure of 4 in. wg. The minimum airflow is 7,500 cfm with a minimum system static pressure set point of 1.5 in. wg.

This data can then be plotted on the fan curve to determine if the minimum point will be outside the recommended operating region.

As a precaution, pay close attention to what motor you apply for VFD applications. There are inverter (VFD) duty motors rated for 1000:1 turndown for conveyor belt or spool-wrapping applications. These are very expensive and definitely not required for HVAC applications. Installations that are to be controlled by VFDs require the use of an inverter-duty motor designed to address the increased stresses placed on motors by these drives. These motors can be run continuously at a minimum of 1/6 motor synchronous speed without damage to the motor.

System Cooling Design:	15,000 cfm
System Static Pressure:	4 in. wg
Minimum Airflow:	7,500 cfm
Minimum Static Pressure Set Point:	1.5 in. wg

% cfm	cfm	System and Fan Static Pressure (in. wg)
100	15,000	4.0
90	13,500	3.525
80	12,000	3.1
70	10,500	2.725
60	9,000	2.4
50	7,500	2.125

Figure 33

Manual Plot of System Curve

VFD Energy Savings in VAV Systems

The three fan laws state the following:

- Airflow varies directly as the fan speed
- Static pressure varies as the square of the fan speed
- Horsepower varies as the cube of the fan speed.

For a complete discussion

on fan laws with formulas and examples, refer to TDP-612, Fans: Features and Analysis.

The third fan law shows us how easy it is to save money with a VFD. Because fan speed (rpm) and cfm are directly interchangeable in the equation, then:

$$\frac{hp_1}{hp_2} = \left(\frac{cfm_1}{cfm_2}\right)^3$$

As an example, with a VAV fan turned down to 50 percent, we see that the theoretical fan bhp requirements would be reduced to 12.5 percent of full airflow bhp requirements. The actual value, however, is affected by the minimum static set point of 1.5 in. wg controlling the VFD.

$cfm_1 = 100\%$

$cfm_2 = 50\%$

$$\left(\frac{cfm_2}{cfm_1}\right) = .5$$

$$hp_2 = hp_1 \left(\frac{cfm_2}{cfm_1}\right)^3$$

$$hp_2 = hp_1 (.125)$$

VAV Fan Control

Any VAV system in which the fan airflow is varied by an inlet guide vane, a discharge damper, or a VFD requires an external input signal to the control system.

The air distribution system requires enough static pressure to overcome the ductwork, coils, filters, etc. and a minimum amount of static pressure for the VAV boxes and/or diffusers to operate properly.

A simple closed-loop control starts with measuring the static pressure from a location in the supply duct, usually one-half to two-thirds of the way down the main duct, regardless of the first duct take-off location. This is compared to the required set point and the fan airflow is adjusted as needed to maintain that set point.

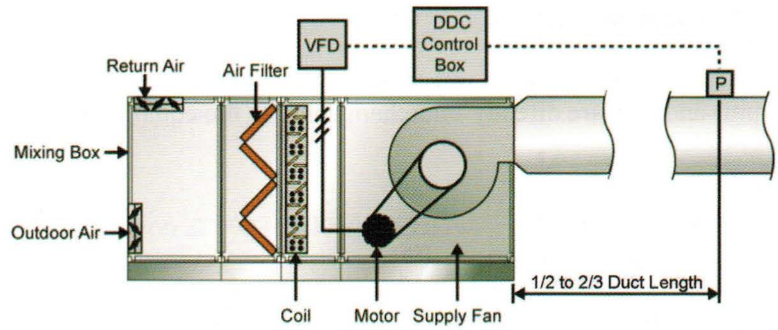


Figure 34

VAV Fan Control

Other things to consider are:

- High static pressure limit controls to protect the air handler, ductwork, and terminal units.
- IGV, discharge damper, and systems riding the fan curve need to be checked at start-up to prevent excessive pressure or possible motor overloading.
- VFDs can be configured to slowly ramp up the motor speed and give the VAV terminals a chance to react. This slow ramp up limits the possibility of start-up over pressurization and also prevents a current surge. In an application where many large fans are in use, this can avoid additional demand surcharges from the utility company.

Fan Tracking and Building Pressurization

Building pressure can become negative, resulting in inward leakage (infiltration) of unconditioned outdoor air unless steps are taken to maintain a positive pressure. Following are two fan arrangement examples that will be reviewed, along with suggested ways to control them.

When a second fan is used, it can be in the return position or in the exhaust position. Return fans run whenever the supply fan runs. Exhaust fans run only upon a build up of space pressure.

Sometimes a fan used for over pressurization relief is called an exhaust fan. Direct exhaust air technically refers to air taken from bathrooms or other areas of the building where the air quality is such that it is not permitted to be returned to the occupied building. In contrast, relief air is air from the conditioned space that needs to be taken out of the building to maintain a proper pressure in the building.

Either way, for purposes of this discussion, we will use the term exhaust fan to designate a second fan that is not in the return position. This is because manufacturers tend to use the term "exhaust" when referring to unit accessories, such as a power exhaust accessory.

Supply and Return Fan Configuration

The primary reason for using return fans is to overcome excessive pressure drop caused by long return air systems. Return air fans are typically needed when the return duct static is greater than 0.375 in. wg. This is a rule of thumb.

On a VAV system, a constant differential in supply and return fan airflow is set up to maintain positive building pressurization at all times. Both fans will have a VFD. The supply fan has a duct static pressure controller that maintains a downstream duct static pressure assuring adequate airflow to the zone terminals. As the supply fan throttles in response to zone loads, the return fan will also.

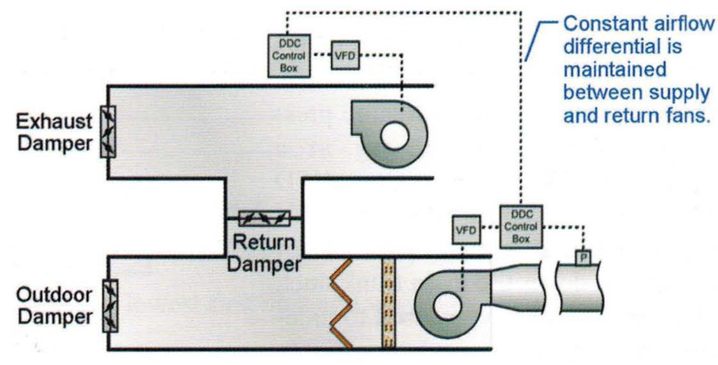


Figure 35

VAV Supply and Return Fans

There are two common ways the return fan is controlled to track the supply fan. The easiest is to maintain a proportional difference in the signal to the return fan VFD. For example, if the supply fan VFD is currently at 80%, the return fan may track at 20 percent less, or 64 percent. As the supply fan VFD modulates to maintain static pressure, the return fan modulates to maintain the 20 percent difference. Another, more accurate but much more costly method involves installing airflow monitors in both the supply and return air ducts and having the return fan VFD control to a differential in airflow of the two airstreams.

As the outdoor damper opens to admit more ventilation air, the return damper will close forcing excess air out of the building through the exhaust damper (which tracks the outdoor damper). Some schemes may utilize a space static pressure controller also.

The decision to use a return fan is not a black-and-white situation. Engineers may also factor in the size of the air handler, measured in cfm, as a further guide when deciding whether or not to use a return fan. Most engineers will not use a return fan for small cfm applications (below 2000 cfm) even if there is an extensive return air duct system (unlikely at that size).

For example, a 2000 cfm job with a return duct loss that is greater than 0.375 in. wg will often not be provided with a return fan. On the other hand, some engineers will use a return fan on larger cfm applications because of standard office design practices. For example, a large 10,000 cfm job might utilize a return fan even if the return duct loss is less than 0.375 in. wg.

Supply and Exhaust Fan Configuration

The primary reason to use an exhaust fan is to assure that building over pressurization can be avoided. During certain times up to 100 percent outside air may be introduced into the building so a means of exhaust is required.

When an exhaust fan is used with a VAV system, volume control must be provided. The most common control of a supply and exhaust fan combination utilizes two VFD drives and DDC controls to maintain a positive static pressure of around 0.05 in. wg in a common building area. One VFD drive is used to control the supply duct static pressure at a point 1/2 - 2/3 of the distance down the main trunk duct. The second VFD drive is used to independently control the exhaust fan to maintain the space static pressure set point. Care should be taken not to locate the space static pressure sensor next to building exterior openings such as foyer doors where the space pressure may fluctuate widely.

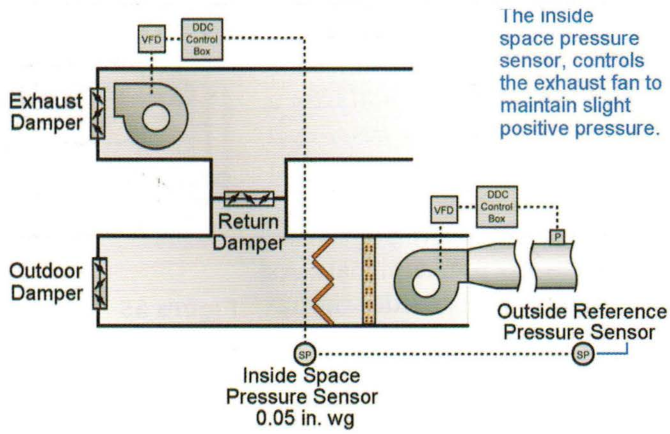


Figure 36

VAV Supply and Exhaust Fan

So, to summarize when to use a return or an exhaust fan:

- Use a return fan when the return duct static is greater than approximately 0.375 in. wg. This is not a hard-and-fast rule, but is a widespread one.
- Use an exhaust fan when space pressurization control is required like may be the case on 100% economizer or make-up air applications.

Keep in mind that the return or exhaust functions do not need to be integrated into the air-handling unit. It is fairly common to have a separate fan for this duty.

Summary

Because buildings operate most of the time at part load, VAV systems remain a popular system choice in today's market place. VAV systems have the inherent ability to significantly reduce fan energy – particularly when the fan is controlled by a variable frequency drive. Thus VFD control has become the dominant volume control method for VAV fans. In order to provide stable and efficient VAV fan operation, care must be taken to select the proper fan type and to select the fan in a manner that avoids part-load fan instability. For centrifugal fan applications with forward-curved wheels, that means staying away from the dip in the fan curve. Also, full and part load stability can be obtained by selecting the fan to the right of the MSE line.

When a second fan is used in addition to the supply fan, it can be in the return or the exhaust location. Return fans are more likely to be used on large cfm applications than on small ones. When a return fan is selected, its part-load volume control must track the supply fan to ensure adequate space pressure control. Finally, exhaust fans should be used for true 100% economizer or make-up air applications.

Work Session

1. True or False? The drive losses in an eddy current coupling are less than that in a VFD.

   ____

2. Which method of fan modulation for VAV applications is most popular and why?

   ____

   ____

3. True or False? VAV systems have the ability to track changes in building loads.

   ____

4. Which method of fan modulation has the lowest first cost and why?

   ____

   ____

5. Discharge dampers are used with which type of fan?

   a. Airfoil
   b. Forward-curved
   c. Plenum
   d. Axial

6. Can you "ride the fan curve" with an airfoil wheel? Why or why not?

   ____

   ____

7. Write the fan law that represents the relationship between horsepower and cfm.____________________

   Why is this important for VFD use?

   ____

8. How do inlet guide vanes (IGVs) work?

   ____

   ____

9. True or False? Part load stability on a VFD-equipped fan can be checked on manufacturer's computerized fan selection software. ____________________

10. Which fan volume control method must absorb the excess static pressure that is generated during part load operation?

    a) Discharge damper
    b) VFD
    c) Eddy current clutch
    d) Inlet guide vane

11. What method of fan volume control is used most often with axial fans?

____

____

____

12. The best overall method of fan modulation in VAV systems is

a) VFD
b) Eddy current coupling
c) Riding the fan curve
d) Discharge dampers

13. True or False? An airfoil type centrifugal fan is a non-overloading fan. ____________

14. True or False? Inlet guide vanes impose an additional external resistance on the fan operation.

____

15. When should a return fan be used?

____

____

____

____

Appendix

Fan Static hp Equation

$$\text{Static hp} = \frac{\text{cfm} * P_s}{6350}$$

Where :

cfm = Standard Air (density = 0.075 lb/ft^3)

P_S = Static Pressure (in wg)

$$\text{Fan Static Efficiency } \% = \frac{\text{Static hp}}{\text{Static bhp}} * 100 = \frac{\text{cfm} * P_S}{6350 * \text{bhp}}$$

Fan Heat Equation

Motor in Airstream

$$\text{btuh} = \frac{\text{Fan bhp} * 3.413 \dfrac{\text{btuh}}{\text{Watt}} * 746 \dfrac{\text{watt}}{\text{hp}}}{\text{Motor Efficiency}}$$

$$= \frac{\text{Fan bhp} * 2545 * \dfrac{\text{btuh}}{\text{hp}}}{\text{Motor Efficiency}}$$

Motor outside of Airstream

$$\text{btuh} = \text{Fan bhp} * 2545 \frac{\text{btuh}}{\text{hp}}$$

Work Session Answers

1. False

2. The VFD (variable frequency drive) is the most popular method of fan modulation because VFDs are most efficient at part load, they can easily report power monitoring information to the building automation system, and their use may result in a rebate from the utility company.

3. True

4. Riding the fan curve has the lowest first cost because there is no additional hardware involved.

5. b

6. No! The steep pressure volume curve and static pressure capability will quickly damage ductwork and VAV terminals.

7. $$\frac{hp_1}{hp_2} = \left(\frac{cfm_1}{cfm_2}\right)^3$$

   VFDs save energy based on this fan law.

8. IGVs are curved in the direction of fan rotation to produce positive whirl, which reduces theoretical head and power. This means that as IGV position changes, the fan curve itself is altered.

9. True

10. a

11. VFD

12. a

13. True

14. True

15. Use a return fan when the return duct static is greater than approximately 0.375 in. wg. This is not a hard-and-fast rule, but it has been in use in the industry for many years.

Prerequisites:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-102	796-026	797-026	ABCs of Comfort
TDP-103	796-027	797-027	Concepts of Air Conditioning
TDP-201	796-030	797-030	Psychrometrics, Level 1: Introduction
TDP-612	796-050	797-050	Fans: Features and Analysis

Learning Objectives:

After reading this module, participants will be able to:

- Understand the impact of fan regulation in a VAV system
- Identify the types of fans used in VAV designs
- Discuss previous and current methods of fan volume control
- Utilize fan curves to demonstrate stable fan selections
- Compare fan volume control methods for energy savings
- Discuss fan static pressure control, tracking, and pressurization

Supplemental Material:

Form No.	Book Cat. No.	Instructor Presentation Cat. No.	Title
TDP-632	796-057	797-057	Rooftop Units, Level 2: Variable Volume
TDP-611	796-049	797-049	Central Station Air Handlers

Instructor Information

Each TDP topic is supported with a number of different items to meet the specific needs of the user. Instructor materials consist of a CD-ROM disk that includes a PowerPoint™ presentation with convenient links to all required support materials required for the topic. This always includes: slides, presenter notes, text file including work sessions and work session solutions, quiz and quiz answers. Depending upon the topic, the instructor CD may also include sound, video, spreadsheets, forms, or other material required to present a complete class. Self-study or student material consists of a text including work sessions and work session answers, and may also include forms, worksheets, calculators, etc.

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