# Building Airflow System Control Applications

**Contents**

- **Introduction** ................................................................. 3
- **Definitions** ...................................................................... 3

**Airflow Control Fundamentals**

- Need For Airflow Control .................................................. 5
- What Is Airflow Control ..................................................... 5
- Types Of Airflow Systems .................................................. 5
  - Variable Air Volume ....................................................... 5
  - Constant Air Volume .................................................... 6
- Variable Versus Constant Air Volume .................................... 7
- Ventilation ......................................................................... 7
- Pressurization ..................................................................... 8
  - Building Pressure Balance ........................................... 8
  - Containment Pressurization ......................................... 9
  - Wind Pressure Effects ............................................... 9
  - Stack Effect .................................................................... 9
- Characteristics Of Fans And Fan Laws ......................... 10
  - General ........................................................................... 10
  - Fan Types ...................................................................... 10
  - Fan Performance Terms ............................................. 11
  - Fan Laws ...................................................................... 11
  - Fan Horsepower .......................................................... 11
  - Duct System Curves ..................................................... 12
  - Fan Curve And System Curve Comparison ................... 12
- Characteristics Of Airflow In Ducts ............................ 12
  - General ........................................................................... 12
  - Pressure Changes Within A Duct ............................... 12
  - Effects Of Fittings ........................................................ 14
  - Effects Of Dampers ..................................................... 15
  - Effects Of Air Terminal Units ..................................... 15
- Measurement Of Airflow In Ducts .............................. 15
  - General ........................................................................... 15
  - Pressure Sensors .......................................................... 15
  - Pitot Tube Sensors ........................................................ 16
  - Total And Static Pressure Sensors ............................... 16
- Airflow Measuring Devices ........................................... 18
Airflow Control Applications

Central Fan System Control ................................................................. 19
Supply Fan Control For VAV Systems .............................................. 19
General ...................................................................................... 19
Duct Static High-Limit Control ..................................................... 20
Return Fan Control For VAV Systems ........................................... 21
Open Loop Control ....................................................................... 21
Direct Building Static Control ...................................................... 21
Airflow Tracking Control ............................................................... 21
Duct Static Control ...................................................................... 22
Relief Fan Control For VAV Systems ............................................ 22
Return Damper Control For VAV Systems ..................................... 23
Sequencing Fan Control ................................................................. 24
Other Control Modes ................................................................... 24
Warm-Up Control ......................................................................... 24
Smoke Control ............................................................................. 24
Night Purge Control ...................................................................... 24
Zone Airflow Control .................................................................... 24
Airflow Tracking/Space Static Pressure ........................................ 24
Multiple Fan Systems .................................................................... 26
Exhaust System Control ................................................................. 26
Types ........................................................................................... 26
Fume Hoods .................................................................................. 26
Laboratory Pressurization .............................................................. 28

References ...................................................................................... 30
INTRODUCTION

This section explains the need for airflow control in a central air handling system, describes the various means of airflow measurement, provides fan and duct characteristics, and discusses suggested means of airflow control. The final control system design depends, of course, on specific job requirements.

There are several types of airflow control that relate directly to the control of airflow in a central air handling system. These types of airflow control include space pressurization, zone pressurization, and exhaust air control. Space or zone pressurization is used when an enclosed area within a building (e.g., a clean room, hospital space, laboratory, fire and smoke control area) must be kept at a positive or negative pressure so contaminated air does not migrate to unwanted areas. Basic types of space pressure control are static pressure, airflow tracking, and constant airflow. Exhaust air control regulates the amount of air exhausted to keep it at the minimum safe level. Space pressure control is generally required with exhaust air control, and control of airflow in a central air handling system is generally required with space pressure control and/or exhaust air control.

For information on air terminal units used in building airflow control system applications, refer to the Individual Room Control Applications section.

DEFINITIONS

Airflow: The rate at which a volume of air moves through a duct. In this section, airflow is denoted $Q$ and is measured in cubic feet per minute (cfm). Airflow is derived:

$$ Q = A \times V_{AVG} $$

Where:
- $Q =$ Airflow in cfm
- $A =$ Cross-sectional area of duct in square feet ($ft^2$)
- $V_{AVG} =$ Average velocity

Axial fan: A propeller type fan where airflow within the wheel is substantially parallel to the shaft and in-line with the duct. Axial fan airflow can be controlled by speed, variable inlet vanes, or variable pitch blades depending on the fan type.

Centrifugal fan: A fan where airflow within the wheel is substantially radial to the shaft and the air must make two turns before being expelled from the fan housing. Centrifugal fan airflow can be controlled by speed, variable inlet vanes, or less commonly by dampers.

Constant Air Volume (CAV) system: A central fan system in which airflow in the duct is maintained at a constant volume.

Differential: The difference between supply and return airflows necessary to maintain a positive or a negative pressure in an area. For example, if supply airflow is 1800 cfm and return airflow is 1500 cfm, the differential (positive) is 300 cfm. The 300 cfm surplus leaves the building through exhaust fans or vents and exfiltration.

Duct: A circular or rectangular tube for conveying air.

Duct cross-sectional area: For round ducts, the duct cross-sectional area is $\pi r^2$, where $r$ is the radius. For rectangular ducts, the duct area is $X \times Y$, where $X$ and $Y$ are the height and width dimensions. In this section, a duct cross-sectional area is measured in square feet ($ft^2$).

NOTE: If duct dimensions are in inches (in.) and the result of the duct area is in square inches ($in^2$), divide the result by 144 $in^2/ft^2$ to obtain square feet ($ft^2$).

Duct diameter: For round ducts, the diameter is twice the radius ($2r$). For rectangular ducts, an equivalent diameter is derived: $2XY \div (X + Y)$, where $X$ and $Y$ are the height and width.

Fan surge: A condition that occurs when air passing over the fan blades causes a stall. A fan surge causes a fluctuation in duct static pressure and an increase in noise level.

Flow Measuring Station (FMS): A device containing multiple static pressure sensors and multiple total pressure sensors manifolded separately for instantaneously measuring average pressures across the face of a duct.

Impact tube: A sensing device with a single opening that points directly into the airstream for measuring total pressure.

Manometer: An instrument for measuring low pressure such as static pressure.

Pitot tube: A sensing device containing both an impact tube and a static pressure tube in a single probe.
**Static pressure:** The pressure created by air (whether in motion or not) confined in an enclosed area such as a duct or building due to its potential energy. Static pressure, denoted SP, is exerted perpendicularly on all interior walls of the enclosure (duct or building) with respect to a reference pressure outside the enclosure. When static pressure is above atmospheric pressure it is positive and when below atmospheric pressure it is negative.

**Static pressure sensor or tube:** A sensing device with several holes perpendicular to an airstream for measuring static pressure.

**Total pressure:** The algebraic sum of Velocity Pressure (VP) plus Static Pressure, denoted TP. Total pressure is derived:

\[ TP = VP + SP \]

**Turndown:** The relationship, in percent, between the maximum minus the minimum airflow to the maximum airflow.

\[ \text{Turndown \%} = \left( \frac{\text{Max Flow} - \text{Min Flow}}{\text{Max Flow}} \right) \times 100 \]

For example, in a system with a maximum airflow of 2000 cfm and minimum airflow of 400 cfm, the turndown is 80 percent.

**Variable Air Volume (VAV) system:** A central fan system in which airflow in the duct varies depending on the instantaneous load requirements of the connected VAV terminal units.

**Velocity:** The speed or rate of flow of the air stream in a duct. In this section, velocity is denoted V and is measured in feet per minute (fpm). See General Engineering Data section.

- **Average Velocity**—The sum of the air velocities from equal area increments of a duct cross-section divided by the number of increments. Average velocity, denoted \( V_{AVG} \), is derived:

\[ V_{AVG} = \frac{\sum (V_1 + V_2 + V_3 + \ldots + V_N)}{N} \]

Where

\[ N = \text{Number of duct increments} \]

- **Peak Velocity**—The greatest air velocity occurring in an increment of a duct cross-section. Peak velocity is denoted \( V_{PK} \).

- **Velocity Pressure:** The pressure created by air moving at a velocity due to its kinetic energy. Velocity pressure, denoted VP, is always exerted in the direction of airflow and is always a positive value. Velocity pressure and velocity are related by the equation:

\[ V = \left( \sqrt{2G \times \frac{VP \times Dw}{Da} \times \frac{1 \text{ ft}}{12 \text{ in.}}} \right) \times \frac{60 \text{ sec}}{1 \text{ min}} \]

Where:

- \( V \) = Velocity in fpm
- \( G \) = Gravitational acceleration in feet per second squared (ft/sec\(^2\))
- \( VP \) = Velocity pressure in inches of water column (in. wc)
- \( Dw \) = Density of water at a specified temperature measured in pounds per cubic foot (lb/ft\(^3\))
- \( Da \) = Density of the air flowing in the duct measured in pounds per cubic foot (lb/ft\(^3\))
- 1ft/12 in. = Conversion factor to convert inches to feet
- 60 sec/1 min = Conversion factor to convert seconds to minutes

The density of air (Da) is 0.075 lb/ft\(^3\) (at 70°F, 29.92 in. wc atmospheric pressure, and 50 percent relative humidity) and gravity (G) is equal to 32.2 ft/sec\(^2\). At 70°F, the density of water (Dw) is 62.27 lb/ft\(^3\). With this data, the relationship of velocity to velocity pressure is simplified:

\[ V = \left( \sqrt{2 \times 32.2 \text{ ft/sec}^2 \times VP \text{ in. wc} \times \frac{62.27 \text{ lb/ft}^3}{0.075 \text{ lb/ft}^3} \times \frac{1 \text{ ft}}{12 \text{ in.}}} \right) \times \frac{60 \text{ sec}}{1 \text{ min}} \]

This equation reduces to:

\[ V = 4005 \sqrt{VP} \]

See General Engineering Data section for Velocity vs. Velocity Pressure table.
AIRFLOW CONTROL FUNDAMENTALS

NEED FOR AIRFLOW CONTROL

Proper control of airflow is important to physiological principles including thermal and air quality considerations. Air distribution systems, containment pressurization, exhaust systems, and outdoor air dilution are examples of airflow control systems used to meet ventilation requirements. Life safety requirements are also met with fire and smoke control systems using airflow control functions. Therefore, an understanding of airflow control is required to provide the various locations in a building with the necessary conditioned air.

One means of maintaining indoor air quality is to dilute undesirable materials (e.g., carbon dioxide, volatile organic compounds) with outdoor air. It is important to understand the control of outdoor air airflow rates in order to:

- Increase outdoor airflow rates when needed for dilution ventilation
- Prevent excessive building and space pressurization
- Minimize outdoor airflow rates when possible to limit energy costs

WHAT IS AIRFLOW CONTROL

In HVAC systems, a well designed combination of fans, ducts, dampers, airflow sensors, static pressure sensors, air terminal units, and diffusers is necessary to provide conditioned air to the required spaces. The function of airflow control is to sense and control the static pressures and airflows of the building. The static pressures occur in ducts and building spaces; airflows occur in ducted air supplies, returns, and exhausts.

TYPES OF AIRFLOW SYSTEMS

An air handling system can provide heating, cooling, humidification, and dehumidification as well as variable quantities of outdoor air. Air handling systems can be classified as single-path or dual-path. The single-path system has all heating and cooling coils in series in a duct. The single duct supplies all terminal equipment. The dual-path system has a cooling coil in one duct and a heating coil (or just return air) in another duct. Both ducts supply dual-duct terminal equipment or multizone dampers.

These systems are further classified (ASHRAE 1996 HVAC Systems and Equipment Handbook) as follows:

Single-Path Systems:
- Single duct, constant air volume
- Single zone systems
- Reheating, single duct-variable air volume
- Simple variable air volume
- Variable air volume, reheat

Single duct, variable air volume-induction
Single duct, variable air volume-fan powered
Constant fan, intermittent fan

Dual-Path Systems:
- Dual duct, single fan-constant air volume
- Single fan, constant air volume reheat
- Variable air volume
- Multizone

The more common types of air handling systems are:
- Single duct, variable air volume
- Single duct, constant air volume

VARIABLE AIR VOLUME

A Variable Air Volume (VAV) system controls primarily the space temperature by varying the volume of supply air rather than the supply air temperature (Fig. 1). The interior zones of most large buildings normally require only cooling because of occupancy and lighting loads. Air terminal units serve these zones and operate under thermostatic control to vary the airflow in the individual spaces to maintain the required temperature. The perimeter zones can have a varying load depending on the season and exposure. Heating may be supplied via reheat coils that operate under thermostatic control while air terminal units maintain minimum airflow.

Airflow in the supply duct varies as the sum of the airflows through each VAV terminal unit varies. In light load conditions, the air terminal units reduce the airflow. As more cooling is required, the units increase airflow. Air terminal units typically have controls to limit maximum and minimum airflow and compensate for variations in supply duct static.

To ensure that all air terminal units have sufficient pressure to operate, a supply airflow control system is required. To monitor duct static, static pressure sensors are installed near the end of the supply duct. When VAV terminal unit dampers open, the static drops in the supply duct. The static pressure sensor detects the static pressure drop and the airflow control system increases the supply fan output. The opposite occurs when the VAV terminal unit dampers close.

Another VAV system feature is that the difference in airflow between the supply and return fans (and not the position of the outdoor air damper) determines the amount of minimum outdoor air ventilation delivered by the supply system. For example, Figure 1 shows a single duct VAV system for a building with the fans on and outdoor air dampers at minimum position. The air terminal units reduce the airflow. As more cooling is required, the units increase airflow. Air terminal units typically have controls to limit maximum and minimum airflow and compensate for variations in supply duct static.
are fully open, the outdoor air volume will be 5,000 cfm. To increase outdoor air volume, it is necessary to modulate the return air damper. This airflow control provides a slightly positive building static pressure with respect to outdoor air in a properly designed system.

As supply air volume is reduced, so is return air volume. The return air fan is normally sized smaller than the supply fan. Flow measuring stations are located in both the supply and return ducts so that the return air fan can track the airflow of the supply fan with a constant flow differential. Thus, as airflow through the supply fan reduces, the control system reduces the airflow of the return fan. This control system can maintain either a fixed airflow difference or a percentage of supply airflow difference. For more information on VAV systems, see AIRFLOW CONTROL APPLICATIONS.

CONSTANT AIR VOLUME

A Constant Air Volume (CAV) system controls space temperature by altering the supply air temperature while maintaining constant airflow. Since the airflow is constant, the system design provides sufficient capacity to deliver supply air to the space for design load conditions. In many systems, reheat coils allow individual space control for each zone and provide heating when required for perimeter zones with different exposure. The CAV system shown in Figure 2 has the same interior and perimeter zone load requirements as the VAV system in Figure 1. The CAV system, however, does not use static pressure sensors and flow measuring stations since the airflow is constant.

In a CAV system, the supply and return fans are manually set to meet the total airflow needs. The cooling coil discharge temperature can be reset as a function of the zone having the greatest cooling load. This improves operating efficiency. The reheat coils on the constant volume boxes are controlled by individual space thermostats to establish the final space temperatures.

Fig. 1. Single Duct Variable Air Volume System.
VARIABLE VERSUS CONSTANT AIR VOLUME

Sizing of central equipment is based on climate conditions which indicate heating and cooling loads. In a CAV system, typically outdoor air is supplied, cooled, distributed to the various zones, and then reheated for the individual needs of each space. The sizing of the central equipment is based on the sum of the loads from all air terminal units served by the CAV system. In Figure 2 for a single duct CAV system, the maximum load from all air terminal units is 40,000 cfm. Therefore, the supply fan is sized for 40,000 cfm and, with 5,000 cfm exhaust, the return fan is sized for 35,000 cfm.

The diversity of the heating and cooling loads in a VAV system permits the use of smaller central equipment. If a building with a VAV system has glass exposures on the east and west sides, the solar load peaks on the two sides at different times. In addition, due to building use, offices and conference rooms on the east and west sides are never fully occupied at the same time. The interior spaces do not use reheat and receive only at minimum airflow. Therefore, the instantaneous load of the VAV system central equipment is not the sum of the maximum loads from all terminal units. Instead, the maximum instantaneous load on the central equipment of a VAV system is a percentage of the sum of all maximum individual loads. This percentage can vary for different buildings.

In Figure 1 for a single duct VAV system, the maximum air terminal unit load is 40,000 cfm with a diversity of 75 percent. This means that the supply fan is sized for only 30,000 cfm. Similarly the return fan is sized for 25,000 cfm instead of 35,000 cfm and the coils, filters, and ducts can also be downsized.

VENTILATION

Care must be used to assure that the AHU ventilation design complies with relevant codes and standards which are frequently revised. Ventilation within a constant air volume system is often a balancing task. During occupied, non-economizer periods of operation, the mixing dampers are positioned to bring in the required OA. The system balancing person determines the specific minimum ventilation damper position.

If this is done on a VAV system at design load, as the VAV boxes require less cooling and less airflow, the supply fan capacity reduces, and the inlet pressure to the supply fan becomes less negative as the fan unloads. As the filter inlet pressure becomes less negative, less OA is drawn into the system which is unacceptable from a ventilation and IAQ perspective. VAV systems therefore require design considerations to prevent non-economizer occupied mode ventilation from varying with the cooling load. This may be accomplished in several ways.

The dampers may be set at design load as for a constant air volume system, and the filter inlet pressure noted. Then the noted filter inlet negative pressure can be maintained by modulating the return air damper. Keeping this pressure constant keeps the OA airflow constant. This method is simple but requires good maintenance on the OA damper and linkage, positive positioning of the OA damper actuator, and the balancing person to provide the minimum damper position and the pressure setpoints.

Another positive method is to provide a small OA injection fan set to inject the required OA into the AHU mixing box during occupied periods (the OA damper remains closed). The fan airflow quantity may be controlled by fan speed adjustment or inlet damper setting and sensed by an airflow measuring station for closed loop modulating control. This basic method is positive and relatively maintenance free, but like the pressure control method, it requires a balancing person to make adjustments. This closed loop control method is more costly and requires keeping the airflow pickup/sensor clean, but it allows simple setpoint entry for future adjustments.

A minimum OA damper may be provided for the occupied OA airflow requirement. An airflow measuring station in the minimum OA duct is required to modulate the minimum OA damper in sequence with the RA damper to maintain a constant volume of OA. This method is more costly than the first method, but it allows convenient software setpoint adjustments.

Theoretically, the mixing dampers may be modulated to maintain a constant OA volume during occupied periods using an OA duct airflow measuring station. Since, in these examples, the OA duct is sized for 100 percent OA, the minimum is usually 20 to 25 percent of the maximum. The airflow velocity at minimum airflow is extremely low, and velocity pressure measurement is usually not practical. Hot wire anemometer velocity sensing at these velocities is satisfactory, but costly, and requires that the sensing element be kept clean to maintain accuracy. (Filtering the entering OA is helpful.) A smaller, minimum OA damper and duct may also be used to assure adequate airflow velocity for velocity pressure measurement.

If a return fan and volumetric tracking return fan control are used, and no relief/exhaust dampers exist, or if the RA damper is closed and the MA damper is open during occupied non-economizer modes, the OA volume equals the SA volume minus the RA volume. This method is simple and low cost but is only applicable when building exhaust and exfiltration meets minimum ventilation requirements.

Where minimum OA only is provided (no economizer dampers), variations of any of these methods may be used. See the Air Handling System Control Applications section for further information.
PRESSURIZATION

Building pressurization and ventilation are important aspects of airflow control. A building or areas within a building can be pressurized for positive, negative, or sometimes neutral static pressure to control the flow of air from one area to another. A building can use positive static pressure to eliminate infiltration drafts, possible dirt, and contamination from outside sources. Areas within a building such as laboratories can use negative pressure to prevent contamination and exfiltration to adjacent spaces and zones. Proper building pressurization must also consider the effects of outdoor wind pressure and stack effect or vertical air differences.

BUILDING PRESSURE BALANCE

The pressures in a building must be balanced to avoid airflow conditions which make it difficult to open and close doors or conditions which cause drafts. Buildings have allowable maximum and minimum static pressures that control these conditions. A force of 30 through 50 pounds is considered the maximum reasonable force to open a door. The equation used to calculate the force to overcome the pressure difference across a door as well as to overcome the door closer is as follows:

\[ F = F_{DA} + \frac{Kd \times W \times A \times \Delta p}{2 \times (W - D)} \]

Where:
- \( F \) = Total door opening force applied at the knob in pounds (lb)
- \( F_{DA} \) = Force to overcome the door closer, applied at the knob, in pounds (lb). This force is normally between 3 and 20 lb.
- \( Kd \) = A coefficient, 5.20
- \( W \) = Door width in feet (ft)
- \( A \) = Door area in square feet (ft²)
- \( \Delta p \) = Differential static pressure across the door in inches of water column (in. wc)
- \( D \) = Distance from the door knob to the knob side of the door in feet (ft)
Rearranging the equation to calculate the differential pressure results in the following:

\[
\Delta p = \left(\frac{F - F_{DA}}{K_d \times W \times A}\right) \times [2 \times (W - D)]
\]

EXAMPLE:
Calculate the differential pressure for a 3 ft wide x 7 ft high door that has a 30-lb opening force, a 10-lb force to overcome the door closer, and 0.25 ft between the door knob and the door edge.

\[
\Delta p = \left(\frac{(30 - 10)}{5.20 \times 3 \times 21}\right) \times [2 \times (3 - 0.25)]
\]

\[
\Delta p = 0.3358 \text{ in. wc}
\]

Similarly, the maximum pressure difference which overcomes a 10 lb door closer is as follows:

\[
\Delta p = \left(\frac{(0 - 10)}{5.20 \times 3 \times 21}\right) \times [2 \times (3 - 0.25)]
\]

\[
\Delta p = 0.1679 \text{ in. wc}
\]

CONTAINMENT PRESSURIZATION

In an airflow system for a building that requires containment pressurization, the direction of infiltration is toward the space with the contaminants. The direction is controlled by regulating the pressure differentials between the spaces. For example, a laboratory is typically kept at a negative pressure relative to surrounding spaces so that any infiltration is into the laboratory. The static pressure difference required for containment varies with the specific application. In buildings with areas that require smoke control, a minimum static pressure difference of 0.02 through 0.04 in. wc is suggested to control cold smoke. (Cold smoke is generated when water spray from a sprinkler system cools smoke from a building fire.)

WIND PRESSURE EFFECTS

Wind effects generate surface pressures which can change supply and exhaust fan capacities, infiltration and exfiltration of air, and interior building pressure. Wind can affect environmental factors (e.g., temperature, humidity, and air motion), dilution ventilation, and control of contaminants from exhausts.

The pressure exerted by wind on a building surface can be calculated from the following equation:

\[
P_w = C_w \times K_w \times V^2
\]

Where:

- \(P_w\) = Wind pressure in inches of water column (in. wc)
- \(C_w\) = Dimensionless pressure coefficient ranging from –0.8 for leeward walls through 0.8 for windward walls
- \(K_w\) = A coefficient, 4.82 x 10^{-4}, for air density of 0.075 lb/ft^3
- \(V\) = Wind velocity in miles per hour (mph)

EXAMPLE:
For a wind velocity of 25 mph and a pressure coefficient of –0.8 (leeward side), the wind pressure on the building is as follows:

\[
P_w = -0.8 \times (4.82 \times 10^{-4}) \times (25)^2 = -0.241 \text{ in. wc}
\]

STACK EFFECT

Stack effect or thermal buoyancy is the difference between indoor and outdoor temperature which causes a pressure difference that affects air movement in a building. Whenever outdoor air is colder than indoor air, the building draws in air near the base (infiltration) and expels air near the top (exfiltration). Conversely, when outdoor air is warmer than indoor air, the building draws in air near the top (infiltration) and expels air near the base (exfiltration).

The level of the building at which the differentials of indoor and outdoor static pressures are equal (with zero wind speed) is called the neutral pressure level or neutral plane (Fig. 3). Location of the neutral plane depends on the distribution of outdoor openings. If one large opening dominates building leakage, the neutral plane is found close to that opening. For vertical openings uniformly distributed, the neutral plane is at midheight. In general, the neutral plane for tall buildings varies from 0.3 through 0.7 of total building height.

Stack effect can be calculated from the following equation:

\[
\Delta p = K_s \times \left(\frac{1}{T_o} - \frac{1}{T_i}\right) \times h
\]

Where:

- \(\Delta p\) = Pressure difference in inches of water column (in. wc)
- \(T_o\) = Outdoor absolute temperature in degrees Rankin (°R) (460 + °F)
- \(T_i\) = Indoor absolute temperature in degrees Rankin (°R) (460 + °F)
- \(h\) = Height of building in feet (ft)
- \(K_s\) = Coefficient, 7.46
CHARACTERISTICS OF FANS AND FAN LAWS

NOTE: This text provides an overview of fan characteristics and fan laws. For more information, see the Trane Air Conditioning Manual listed in REFERENCE.

GENERAL

In airflow systems, a fan converts mechanical, rotative energy into fluid energy. This is basically accomplished by a wheel or propeller which imparts a forward motion to the air. For HVAC applications, fans rarely exceed a total pressure of 12 in. wc.

Fans must be properly installed to achieve smooth control and correct performance. In general, manufacturer recommendations should be followed and the following noted (from Engineering Fundamentals of Fans and Roof Ventilators, Plant Engineering, Copyright 1982):

— Fans should be located so the discharge of one does not enter the intake of another fan.
— Intake area should be at least 20 percent greater than the fan wheel discharge area.
— Fans located opposite from each other should be separated by at least six fan diameters.
— Elbows or other abrupt duct transformations on the discharge side of the fan should not be closer than one diameter from the fan wheel.
— Direction of fan discharge and rotation should be selected to match duct or mounting requirements.

FAN TYPES

Two main types of fans are used in airflow systems centrifugal and axial:

Centrifugal Fans: A centrifugal fan (Fig. 4) has airflow within the wheel that is substantially radial to the shaft (or away from the axis of the shaft). The air from an in-line centrifugal fan does not have to turn before being expelled from the fan housing. Some centrifugal fan designs are differentiated by the inclination of the blades. Each blade design has a peculiar advantage:

— Backward inclined blades are generally larger and more quiet than forward inclined blades. They are more suitable for larger sizes.
— Forward inclined blades are suitable in small packaged units and operate at a lower static pressure.
— Air foil blades are backward inclined, and are efficient and quiet due to an air foil shaped blade. Generally these are used on the largest fans.

Axial Fans: An axial fan has airflow through the wheel that is substantially parallel to the shaft (or along the axis of the shaft). Various designs of axial fans are available (Fig. 5), mainly differentiated by the duty of the fan. Each design has a peculiar advantage:

— Propeller fans are low pressure, high airflow, noisy fans. They work up to a maximum static pressure of 0.75 in. wc.
— Tubeaxial fans are heavy-duty propeller fans arranged for duct connection. They discharge air with a motion that causes high friction loss and
noise. They work up to a maximum static pressure of 3 in. wc.
— Vaneaxial fans are basically tubeaxial fans with straightening vanes added to avoid spiraling air patterns. They are space efficient, quieter than tubeaxial fans, and work at static pressures up to 10 in. wc.

**FAN PERFORMANCE TERMS**

The following are terms used when discussing fan performance:

**Fan volume**: The airflow passing through the fan outlet. Generally this fan outlet value is only slightly less than the airflow at the fan inlet because specific volume changes due to air compression are small.

**Fan outlet velocity**: The fan volume divided by the fan outlet area. This velocity is a theoretical value because, in reality, the velocity pattern at the outlet of a fan is not easy to measure.

**Fan Static Pressure (FSP)**: The fan total pressure minus the fan velocity pressure (FSP = FTP – FVP). It can be calculated by subtracting the total pressure at the fan inlet from the static pressure at the fan outlet.

**Fan Total Pressure (FTP)**: The difference between the total pressure at the fan inlet and the total pressure at the fan outlet. The FTP value measures the total mechanical energy added to the air by the fan.

**Fan Velocity Pressure (FVP)**: The velocity pressure corresponding to the fan outlet velocity.

**FAN LAWS**

Fan laws (Table 1) are simple and useful when dealing with changing conditions. Three important laws deal with speed changes:

1. Airflow varies directly with the fan speed. For example, doubling the fan speed (rpm) doubles the airflow (cfm) delivery.
2. Static pressure varies as the square of the fan speed. For example, doubling the fan speed (rpm) develops four times the static pressure (in. wc).
3. Power varies as the cube of the fan speed. For example, doubling the fan speed (rpm) requires eight times the fan power (hp).

### Table 1. Fan Laws

<table>
<thead>
<tr>
<th>Variable</th>
<th>When Speed Changes</th>
<th>When Density Changes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Airflow</td>
<td>Varies DIRECT with Speed Ratio</td>
<td>Does Not Change</td>
</tr>
<tr>
<td>CFM2</td>
<td>$\frac{\text{ CFM}_2}{\text{ CFM}_1} = \left( \frac{\text{ RPM}_2}{\text{ RPM}_1} \right)^2$</td>
<td></td>
</tr>
<tr>
<td>Pressure</td>
<td>Varies with SQUARE of Speed Ratio</td>
<td>Varies DIRECT with Density Ratio</td>
</tr>
<tr>
<td>P2</td>
<td>$P_2 = P_1 \left( \frac{\text{ RPM}_2}{\text{ RPM}_1} \right)^2$</td>
<td>$P_2 = P_1 \left( \frac{D_2}{D_1} \right)$</td>
</tr>
<tr>
<td>Horsepower</td>
<td>Varies with CUBE of Speed Ratio</td>
<td>Varies DIRECT with Density Ratio</td>
</tr>
<tr>
<td>HP2</td>
<td>$\text{ HP}_2 = \text{ HP}_1 \left( \frac{\text{ RPM}_2}{\text{ RPM}_1} \right)^3$</td>
<td>$\text{ HP}_2 = \text{ HP}_1 \left( \frac{D_2}{D_1} \right)$</td>
</tr>
</tbody>
</table>

**FAN HORSEPOWER**

The theoretical horsepower (hp) required to drive a fan is the horsepower required if there were no losses in the fan (100 percent efficiency). The horsepower formula is:

\[
\text{Theoretical hp} = \frac{\text{cfm} \times \text{FTP}}{6356}
\]

Where:
- cfm = Quantity of air
- 6356 = A constant for English units
- FTP = Fan total pressure.

Brake horsepower (bhp) is the actual horsepower required to drive the fan.

\[
\text{bhp} = \frac{\text{Theoretical hp} \div \text{Fan efficiency}}{6356 \times \text{Fan efficiency}}
\]

The brake horsepower is always larger than the theoretical horsepower due to inefficiencies. The actual brake horsepower of the fan can be determined only by testing.
DUCT SYSTEM CURVES

Fan unit duct systems have a certain amount of friction, or resistance, to the flow of air. Once this resistance of the duct system is known for a specific volume of airflow, a curve can be drawn based on the relationship:

\[ \frac{P_2}{P_1} = \left(\frac{\text{cfm}_2}{\text{cfm}_1}\right)^2 \]

This formula is merely another way of stating that pressure (P) changes as the square of the airflow (cfm).

The system curve (also called system resistance, duct resistance, or system characteristic) is similar to Figure 6.

FAN CURVE AND SYSTEM CURVE COMPARISON

In order to deliver the required air quantity, a fan must be selected that can overcome the duct resistance. However, because of dampers repositioning and other equipment changes, resistance of the duct may change. The results of such conditions can be seen in Figure 7.

The fan curves shown are for a fan running at two speeds, 400 rpm and 600 rpm. Also, two system curves, A and B, have been plotted. The intersection of the system curves and the fan curves indicate the quantities of air the fan will provide. With System Curve A, if the fan is running at 600 rpm, it will deliver 20,500 cfm at 0.70 in. wc (Point 1). With the same system curve (A), if the fan is running at 400 rpm, it will deliver 14,400 cfm at 0.30 in. wc (Point 2).

System Curve B shows increased resistance of the duct system due to dampers throttling or filters clogging. With System Curve B, if the fan is running at 600 rpm, it will deliver 15,600 cfm at 1.53 in. wc (Point 3). With the same system curve (B), if the fan is running at 400 rpm, it will deliver 10,500 cfm at 0.70 in. wc (Point 4).

CHARACTERISTICS OF AIRFLOW IN DUCTS

GENERAL

Supply and return ducts can be classified by application and pressure (ASHRAE 1996 Systems and Equipment Handbook). HVAC systems in public assembly, business, educational, general factory, and mercantile buildings are usually designed as commercial systems. Air pollution control systems, industrial exhaust systems, and systems outside the pressure range of commercial system standards are classified as industrial systems.

Classifications are as follows:

- **Residences**—±0.5 to ±1.0 in. of water.
- **Commercial Systems**—±0.5 to ±10 in. of water.
- **Industrial Systems**—Any pressure.

The quantity of air flowing in a duct can be variable or constant, depending on the type of system. See TYPES OF AIRFLOW SYSTEMS.

PRESSURE CHANGES WITHIN A DUCT

For air to flow within a duct, a pressure difference must exist. The fan must overcome friction losses and dynamic (turbulent) losses to create the necessary pressure difference. Friction losses occur due to air rubbing against duct surfaces. Dynamic losses occur whenever airflow changes velocity or direction. The pressure difference (also called pressure head) required to move air must be sufficient to overcome these losses and to accelerate the air from a state of rest to a required velocity.

In HVAC systems, the air supplied by the fan includes two types of pressures: velocity pressure and static pressure. Velocity pressure is associated with the motion of air and is kinetic energy. Static pressure is exerted perpendicularly to all walls of the duct and is potential energy. Velocity and static
Pressure are measured in inches of water column (in. wc). Total pressure is the sum of the static and velocity pressure and, therefore, is also measured in inches of water column.

In an airflow system, the relationship between velocity pressure and velocity is:

\[ V = 4005 \sqrt{VP} \]

NOTE: See Velocity Pressure in DEFINITIONS for a derivation of this formula.

If the velocity and duct size are known, the volume of airflow can be determined:

\[ Q = AV \]

Where:
- \( Q \) = Airflow in cubic feet per minute (cfm)
- \( A \) = Cross-sectional area of duct in square feet (ft²)
- \( V \) = Velocity in feet per minute (fpm)

Examples of the relationships between total, velocity, and static pressures are shown in Figure 8A for positive duct static pressures and Figure 8B for negative duct static pressures. When static pressure is above atmospheric pressure it is positive and when below atmospheric pressure it is negative. The examples use U-tube manometers to read pressure. The sensor connected to the U-tube determines the type of pressure measured.

In a theoretical duct system without friction losses, the total pressure is constant along the entire duct (Fig. 9). The static and velocity pressures, however, change with every change in the duct cross-sectional area. Since the velocity decreases in larger duct sections, the velocity pressure also decreases, but the static pressure increases. When theoretical ducts change size, static pressure is transformed into velocity pressure and vice versa.
An actual duct system (Fig. 10) encounters a phenomenon called pressure loss or friction loss. Pressure loss is caused by friction between the air and the duct wall. Dynamic losses also occur due to air turbulence caused by duct transitions, elbows, tees, and other fittings. At the open end of the duct in Figure 10, the static pressure becomes zero while the velocity pressure depends solely on the duct size. The pressure loss due to friction appears to be a static pressure loss. However, in reality the total pressure decreases because the pressure loss due to friction also indirectly affects the air velocity in the duct. When the duct inlet and outlet sizes are identical, the velocity pressures at both places are equal and the difference in static pressure readings actually represents the pressure loss due to friction.

Fig. 10. Actual Changes in Pressure with Changes in Duct Area.

In most applications, the duct outlet is larger than the duct inlet (velocity is lower at the outlet than at the inlet). When the duct size increases, a small part of the initial velocity pressure is converted into static pressure and lost as friction loss (Fig. 11). This concept is called static regain. Similar to water flow through a pipe, a larger airflow through a given duct size causes a larger pressure loss due to friction. This pressure drop or friction loss cannot be regained or changed to static or velocity pressure.

Fig. 11. Pressure Changes in a Duct with Outlet Larger than Inlet.

The size of a duct required to transport a given quantity of air depends on the air pressure available to overcome the friction loss. If a small total pressure is available from the fan, the duct must be large enough to avoid wasting this pressure as friction loss. If a large total pressure is available from the fan, the ducts can be smaller with higher velocities and higher friction losses. Reducing the duct size in half increases the velocity and the friction loss increases.

In most low pressure airflow systems, the velocity component of the total pressure may be ignored because of its relative size. For example, if a supply fan delivers 10,000 cfm at 2 in. wc static pressure in a supply duct that is 3 ft x 4 ft (or 12 ft²), the Velocity \( V = \frac{Q}{A} \) is \( 10,000 \text{ cfm} \div 12 \text{ ft}^2 \) or 833 fpm. The Velocity Pressure \[ VP = \left( \frac{V}{4005} \right)^2 \] is \( \left( \frac{833}{4005} \right)^2 = 0.043 \) in. wc. The velocity pressure is 2.2 percent of the static pressure at the fan \( \left( \frac{0.043}{2.0} \right) \times 100 = 2.2\% \).

In most high pressure airflow systems, the velocity pressure does become a factor. For example, if a supply fan delivers 10,000 cfm at 6 in. wc static pressure in a round supply duct that is 24 in. in diameter or 3.14 ft², the Velocity \( V = \frac{Q}{A} \) is \( 10,000 \text{ cfm} \div 3.14 \text{ ft}^2 \) or 3183 fpm. The Velocity Pressure \[ VP = \left( \frac{V}{4005} \right)^2 \] is \( \left( \frac{3183}{4005} \right)^2 \) or 0.632 in. wc. The velocity pressure is 10.5 percent of the static pressure at the fan \( \left( \frac{0.632}{6.0} \right) \times 100 = 10.5\% \).

EFFECTS OF FITTINGS

Ducts are equipped with various fittings such as elbows, branch takeoffs, and transitions to and from equipment which must be designed correctly to prevent pressure losses.

In elbows, the air on the outside radius tends to deflect around the turn. The air on the inside radius tends to follow a straight path and bump into the air on the outer edge. This causes eddies in the air stream and results in excessive friction losses unless prevented. Turning vanes are often used in elbows to reduce the friction loss. In addition, they provide more uniform and parallel flow at the outlet of the elbow.

In transitions to and from equipment an attempt is made to spread the air evenly across the face of the equipment. If the diverging section into the equipment has too great an angle, splitters are often used. The splitters distribute the air evenly and reduce friction losses caused by the air being unable to expand as quickly as the sides diverge. In converging sections friction losses are much smaller, reducing the requirement for splitters.
EFFECTS OF DAMPERS

Dampers are often used in ducts for mixing, for face and bypass control of a coil, for volume control, or for numerous other air volume controls. Figure 12 shows the velocity profile in a straight duct section. Opposed blade dampers are recommended where there are other system components downstream of the damper such as coils or branch takeoffs as they do not adversely distort the air velocity profile. Parallel blade damper can be used where the airflow discharges into a free space or a large plenum.

Fig. 12. Velocity Profile of Parallel Blade vs Opposed Blade Damper.

EFFECTS OF AIR TERMINAL UNITS

A variety of air terminal units are available for air handling systems. For VAV systems, the single duct, Variable Constant Volume (VCV), throttling type air terminal unit (Fig. 13) is typically used. With this device, the space thermostat resets the setpoint of an airflow controller, varying the volume of conditioned air to the space as required. Since a number of these units are usually connected to the supply duct, it is the collective requirements of these units that actually determines the airflow in the main supply duct. In this type of system, the supply fan is controlled to maintain a constant static pressure at a point in the duct system so there is sufficient supply air for all of the air terminal units.

Fig. 13. Single Duct, Variable Constant Volume Air Terminal Unit.

MEASUREMENT OF AIRFLOW IN DUCTS

GENERAL

Total pressure and static pressure can be measured directly; velocity pressure cannot. Velocity pressure is found by subtracting the static pressure from the total pressure. This subtraction is typically done by differential pressure measuring devices.

PRESSURE SENSORS

Some applications require only the measurement of static pressure. To obtain an accurate static pressure measurement, a static pressure sensor (Fig. 14A) is used. This sensor has a closed, bullet-shaped tip followed by small peripheral holes that are perpendicular to the airflow for measuring air pressure. The total pressure sensor (Fig. 14B) is similar except there is an opening in the end of the tube and no openings along the sides.

Fig. 14. Pressure Sensors.
PITOT TUBE SENSORS

A pitot tube measures both total pressure and static pressure. This device combines the total pressure sensor and the static pressure sensor tubes into one device (Fig. 15).

Fig. 15. Pitot Tube.

To obtain accurate velocity pressure readings, the pitot tube tip must point directly into the air stream. Figure 16 shows the error in static and total pressure readings when the pitot tube does not point directly into the air stream. Misaligning a pitot static pressure tube causes the static readings to first increase and then drop off rapidly as the angle of inclination ($\theta$) increases. The total pressure reading drops off gradually and then more rapidly as $\theta$ increases. (Modern Developments in Fluid Dynamics, Volume 1, Dover Publications, Copyright 1965.)

Fig. 16. Error Caused by Improperly Mounting Pitot Tube in Airstream.

The following are guidelines for using pitot tubes:

1. A pitot tube can be used to measure either static or total pressure.
2. A pitot tube measures the velocity at one point. However, several readings across the duct (a traverse) are normally needed to obtain an accurate average velocity.
3. Accurate pressure readings cannot occur in locations with varying swirls, eddies, or impinged air. Because of this:
   — The pitot tube must be inserted into the air stream at least 10 duct diameters downstream from elbows, and five duct diameters upstream of bends, elbows, or other obstructions which cause these effects. (See DEFINITIONS for a description of round and rectangular duct diameters.)
   — Air straightening vanes must be located upstream from a pitot tube location to ensure accurate readings. Rotating airflow can occur in long, straight, duct sections. If air straightening vanes are not used, this nonparallel airflow can strike a pitot tube at an angle and produce false total and static pressure measurements.
4. It is impractical to use the pitot tube (or devices using its principles) at velocities below about 700 fpm (velocity pressure of 0.0305 in. wc).

Even with an inclined manometer, velocity pressures below 0.0305 in. wc are too small for reliable measurement outside of a well equipped laboratory. According to the Industrial Ventilation Manual 17th Edition, 1982, “A carefully made and accurately leveled 10:1 inclined manometer calibrated against a hook gauge can read to approximately ±0.005 in. wc. A standard pitot tube with an inclined manometer can be used with the following degree of accuracy:

<table>
<thead>
<tr>
<th>Velocity (fpm)</th>
<th>Percent Error (±)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4000</td>
<td>0.25</td>
</tr>
<tr>
<td>3000</td>
<td>0.3</td>
</tr>
<tr>
<td>2000</td>
<td>1.0</td>
</tr>
<tr>
<td>800</td>
<td>6.3</td>
</tr>
<tr>
<td>600</td>
<td>15.0</td>
</tr>
</tbody>
</table>

It can be seen that the use of the pitot tube in practical applications is limited at velocities lower than 600-800 fpm.”
An analysis of the accuracy of the pitot tube at 800 fpm follows:

**Velocity Pressure:**
\[ VP = \left( \frac{800}{4005} \right)^2 \]
\[ = 0.0399 \text{ in. wc} \]

**Accuracy:**

- **High**
  
  0.0399 in. wc
  
  +0.0050 in. wc
  
  0.0449 in. wc
  
  or 848.6 fpm

- **Low**
  
  0.0399 in. wc
  
  -0.0050 in. wc
  
  0.0349 in. wc
  
  or 748.2 fpm

**Range**

848.6 fpm

-748.2 fpm

100.4 fpm

or 100.4 ÷ 2 = ±50.2 fpm

**Percent Error:**

\[ (\pm 50.2 \div 800) \times 100 = \pm 6.3\% \]

In practical situations, the velocity of the air stream is not uniform across the cross-section of a duct. Friction slows the air moving close to the walls so the velocity is greater away from the wall.

To determine the average velocity, a series of velocity pressure readings at points of equal area is found. It is recommended to use a formal pattern of sensing points across the duct cross-section. These readings across the duct cross-section are known as traverse readings. Figure 17 shows recommended pitot tube locations for traversing round and rectangular ducts. In round ducts, velocity pressure readings at the centers of the areas of equal concentric areas are taken. Readings are taken along at least two diameters perpendicular to each other. In rectangular ducts, readings at the centers of equal rectangular areas are taken. The velocities are then mathematically totaled and averaged.

**TOTAL AND STATIC PRESSURE SENSORS**

Other arrangements are available to instantaneously average sensed pressures and manifold these values to the exterior of the duct. The Tchebycheff (Tcheb) tube method (Fig. 18) is one such arrangement. This method separately manifolds the total and static pressure sensors. Inside each manifold is a tube with a single slot which receives an average pressure signal from the manifold. The averaged signals from the total and static pressure tubes may be used for indication and control functions. This method assures an accurate reading of flow conditions and a steady signal.

**Fig. 17. Pitot Tube Locations for Traversing Round and Rectangular Ducts.**

The principle behind the manual pitot tube transverse is simple and straightforward. However, care must be taken to obtain accurate readings without inadvertent violation of the formal traverse pattern. With the manual pitot tube traverse, individual velocity readings must be calculated mathematically, totaled, and divided by the number of readings to obtain average velocity.

**Fig. 18. Tcheb Tube Sensors and Manifold.**
AIRFLOW MEASURING DEVICES

Various devices for measuring airflow are available. Figure 19 shows a flow measuring station that consists of:

— An air straightener section to eliminate swirl type airflow and help normalize the velocity profile
— Total pressure sensors aligned into the air stream and spaced on an equal area basis
— Static pressure sensors with ports perpendicular to the airflow direction and positioned on an equal area basis
— Separate manifolds for static and total pressure averaging

The flow measuring station in Figure 19 uses a tube as an air straightener (Fig. 20) with total and static pressure sensors at the downstream end. These tubes are arranged on an equal area basis.

Electronic flow stations use the tube type construction with thermal velocity sensors instead of static and total pressure sensors. Other flow sensing arrangements use holes in round tubing (Fig. 21A and 21B) or airfoil designs (Fig. 21C). It is important to consult the manufacturer of each type of flow station for specifications, calibration data, and application information (location limitations within ducts).

Fig. 19. Averaging Flow Measuring Station.

Fig. 20. Air Straighteners/Sensors and Manifolds.

Fig. 21. Miscellaneous Flow Sensing Arrangements.
Figure 22 illustrates an airflow pickup station typically used in the primary air inlet to a VAV air terminal unit. The pickup station consists of two tubes that measure differential pressure. This measurement can be used in an airflow calculation.

**AIRFLOW CONTROL APPLICATIONS**

**CENTRAL FAN SYSTEM CONTROL**

Figure 23 shows the net airflow balance for a building space and where the return air, outdoor air, and supply fan inlet meet. Assume that the return air damper is open, the relief air damper is closed, and the outdoor air damper is open enough to allow minimum design outdoor air to pass. When the outdoor and relief dampers open further and the return damper closes further, outdoor air increases above minimum. These conditions are used for free cooling (economizer cycle control) and when minimum air must be greater than the difference between supply and return fan airflow rates.

Typically, the supply duct is complex with multiple runs or branches. This duct layout requires a compromise in duct static pressure sensor location that is usually about 75 percent of the distance between the fan and the furthest air terminal unit (Fig. 24B). In complex duct runs where multiple branches split close to the fan, sensors should be located in each end of the branch (Fig. 24C). Each sensed point should have its own setpoint in the control loop. This avoids the assumption that branches and multisensor locations have identical requirements. The sensed point having the lowest duct pressure relative to its own setpoint should control the supply fan.

When a long straight duct (10 diameters) is available, a single point static sensor or pitot tube can be used. When long duct sections are not available, use static or airflow measuring stations which are multipoint and have flow straighteners to provide the most accurate sensing. A reference pressure pickup should be located outside the duct (in or near the controlled space) and adjacent to the duct sensor to measure space static in areas served by the duct. The static pressure sensor should not be located at the control panel in the equipment room if duct static is measured elsewhere. Equipment room static pressure varies as outdoor winds change, outdoor air and relief damper positions change, or exhaust fan operation changes.
A wide proportional band setting (10 times the maximum duct static pressure at the fan discharge) on the fan control is a good starting point and ensures stable fan operation. Integral action is necessary to eliminate offset error caused by the wide proportional band. Integral times should be short for quick response. The use of inverse derivative, which essentially slows system response, does not produce the combination of stability and fast response attainable with wide proportional band and integral control modes. (See the Control Fundamentals section for more information on proportional band and integral action.)

Inlet vane dampers, variable pitch blades (vane axial fans), or variable speed drives are used to modulate airflow (both supply and return). Actuators may require positive positioning to deal with nonlinear forces. Variable speed drives, especially variable frequency, provide excellent fan modulation and control as well as maximum efficiency.

**Duct Static High-Limit Control**

High-limit control of the supply fan duct static should be used to prevent damage to ducts, dampers, and air terminal units (Fig. 25). Damage can occur when fire or smoke dampers in the supply duct close or ducts are blocked, especially during initial system start-up. Fan shut-down and controlling high-limit are two techniques used to limit duct static. Both techniques sense duct static at the supply fan discharge.

Fan shut-down simply shuts down the fan system (supply and return fans) when its setpoint is exceeded. High-limit control requires a manual restart of the fans and should be a discrete component separate from the supply fan primary control loop. The fan shut-down technique is lowest in cost but should not be used with smoke control systems where continued fan operation is required.

A controlling high-limit application is used when the fan system must continue to run if duct blockage occurs, but its operation is limited to a maximum duct static. For example, a fire or smoke damper in the supply duct closes causing the primary duct static pressure sensor to detect no pressure. This would result in maximum output of the supply fan and dangerously high static pressure if the controlling high pressure limit is not present. A controlling high-limit control will modulate the fan to limit its output to the preset maximum duct static (Fig. 26).
RETURN FAN CONTROL FOR VAV SYSTEMS

Return fan operation influences building (space) pressurization and minimum outdoor air. There are four techniques to control the return fan: open loop, direct building static, airflow tracking, and duct static.

Open Loop Control

Open loop control (Fig. 27) modulates the return fan without any feedback. This type of control presumes a fixed relationship between the supply and return fans, controls the return fan in tandem with the supply fan, and changes the output of the return fan without measuring the result. Open loop control requires similar supply and return fan operating characteristics. Therefore, a careful analysis of the supply and return fan operating curves should be done before selecting this technique. Also, accurate balancing is essential to ensure proper adjustment at maximum and minimum operating points. Mechanical linkage adjustments or other means are used to adjust the differential between the two fans for desired flows and to minimize tracking errors at other operating points. With digital control, software can be used to align the fan loading relationships, to vary exhaust effects, and to offset dirty filter effects to minimize flow mismatches.

Fig. 27. Open Loop Control.

Open loop control is often acceptable on small systems more tolerant of minimum outdoor air and building pressurization variations. Since open loop control does not sense or control return airflow, changes caused by the return side dampering and exhaust fan exfiltration change the minimum airflow and building pressurization. Systems with low airflow turndowns also are more suitable for open loop control. As a rule, turndowns should not exceed 50 percent.

In a control sequence where the outdoor air damper is closed (e.g., night cycle or morning warm-up), open loop control should not be used. Excessive negative pressurization will occur in the duct between the return and supply fans.

Direct Building Static Control

In direct building static control, the return fan responds directly to the building space static pressure referenced to the static pressure outside of the building (Fig. 28). The location of the building space static pressure sensor should be away from doors opening to the outside, elevator lobbies, and some confined areas. Usually a hallway on an upper floor is suitable. The outdoor static pressure sensor must avoid wind effects and be at least 15 feet above the building to avoid static pressure effects caused by wind. Due to stack effect, this technique should not be used in tall buildings with open spaces or atriums, unless the building has been partitioned vertically. This technique should use proportional plus integral control with a wide throttling range for stable, accurate, and responsive operation.

Fig. 28. Direct Building Static Control.

Since building static is controlled directly, the pressure remains constant even when exhaust fan airflow changes. Minimum outdoor airflow varies with changes in exhaust fan airflow and building infiltration/exfiltration. In a control sequence where the outdoor air damper is closed, the building static must be reset to zero and all exhaust fans should be turned off.

Airflow Tracking Control

In airflow tracking (Fig. 29) control, the return fan airflow is reset based on the relationship between supply and exhaust fan airflows. That relationship is usually a fixed difference between the supply total airflow and return plus exhaust total airflow, but it can also be a percentage of supply total airflow. When duct layout prevents measuring of total airflow from one flow station, measurements in multiple ducts are totaled. This technique is usually higher cost, especially when multiple flow stations are required. Airflow tracking control and direct building static control are preferable to open loop control. Proportional plus integral control is necessary for accurate operation.
Minimum outdoor airflow should be maintained at a constant level, independent of exhaust fan airflow and changes in building infiltration/exfiltration. Building space pressurization varies only when building infiltration/exfiltration changes. Exhaust fan airflow must reset the return fan airflow for building pressurization to be independent of exhaust fans. In a control sequence where the outdoor air damper is closed, the differential between supply airflow and return airflow must be reset to zero and all exhaust fans should be turned off.

Refer to the Air Handling System Control Applications section for an example of a VAV AHU WITH RETURN FAN AND FLOW TRACKING CONTROL.

**Duct Static Control**

Duct static control is similar to supply fan duct static high-limit control, except return duct static pressure is negative. If individual space returns are damper controlled, return fan control must use this technique (Fig. 30). Duct static control is relatively simple, but individual space return controls make the entire system complex.

Minimum building outdoor air is the difference between supply total airflow and return total plus the exhaust fan total airflow. If controlling the space returns from airflow tracking (using dampers), the exhaust fan volume must be included in the tracking control system for constant building and space pressurization. If controlling the space returns by space pressurization, building and space pressurization remain constant regardless of exhaust fan operation.

**RELIEF FAN CONTROL FOR VAV SYSTEMS**

Relief fans are exhaust fans for the central air handling system. They relieve excessive building pressurization and provide return air removal for economizer cycles.

In Figure 31, a relief damper is located after the relief fan and is controlled to open fully whenever the relief fan operates. Direct building static pressure or airflow tracking controls the relief fan. In direct building static pressure control (Fig. 32), the same guidelines apply as for return fan control. During minimum ventilation cycles and when the outdoor air damper is closed, the relief fan should be turned off and the relief damper closed. In airflow tracking (Fig. 33), flow measuring stations should be located in relief and outdoor air ducts, not in supply and return ducts as with return fans.
RETURN DAMPER CONTROL FOR VAV SYSTEMS

In systems having a return fan, when the mixed air control cycle is not operating, the outdoor air (or maximum outdoor air) and relief dampers are closed and the return damper remains fully open. When the mixed air control cycle is operating, the return damper modulates closed and outdoor air (or maximum outdoor air) and relief dampers modulate open. The return damper should be sized for twice the pressure drop of the outdoor air (or maximum outdoor air) and relief dampers. This prevents the possibility of drawing outdoor air through the relief damper (Fig. 34).

If a mixed air control cycle is required, a relief fan may be required. In airflow tracking (Fig. 37), the mixed air controller opens the outdoor air damper above minimum and the relief fan tracks the outdoor air. In direct building static pressure control (Fig. 38) the mixed air controller opens the outdoor air and relief dampers and closes the return damper, causing the relief fan to eliminate excessive pressure.

Fig. 34. Return Air Damper Mixed Air Control Cycle.

In systems not having a return fan, the return damper controls minimum outdoor airflow or building pressurization. Airflow tracking or direct building static pressure control can be used to control the return damper, depending on which parameter is most important. See Figure 35 for airflow tracking and Figure 36 for direct building static pressure control.

Fig. 35. Return Damper Control Using Airflow Tracking.

Fig. 36. Return Damper Control Using Direct Building Static Pressure.

Fig. 37. Mixed Air Control Cycle with Relief Fan Control Using Airflow Tracking.

Fig. 38. Mixed Air Control Cycle with Relief Fan Control Using Direct Building Static Pressure.
SEQUENCING FAN CONTROL

VAV systems with multiple fans can use fan sequencing. This allows the fans to operate with greater turndown. For example, if a single fan modulates from 100 percent to 50 percent of wide open capacity (50 percent turndown), then two fans with exactly half the capacity of the larger fan can run with 75 percent turndown. Also, sequencing is more efficient. Most of the year the system is not run at full capacity. Under light load conditions, some fans run while others are at standby. It should be taken into account that horsepower varies with the cube of the fan airflow when determining fan staging strategies.

For single supply fan systems, fan output volume is controlled by duct static pressure. However, the decision to turn fans on or off is based on total supply airflow. For centrifugal fans, if Supply Fan 1 in Figure 39 is operating near its maximum velocity capacity, Supply Fan 2 is energized. This opens Damper 2 and Fan 2 is slowly modulated upward. As the duct static is satisfied, Fan 1 will modulate downward until Fan 1 and Fan 2 are operating together, controlled by duct static pressure. If the outputs of Fans 1 and 2 approach maximum capability, Supply Fan 3 is energized. When zone load decreases and terminal units decrease airflow, duct static increases, modulating fans downward. When total supply flow decreases enough, Fan 3 is turned off and Fans 1 and 2 increase in output as required to maintain duct static. Similarly, Fan 2 may be turned off. Time delays protect fan motors from short cycling and fan operation may be alternated to spread wear.

Vanexial fan sequencing is also decided by total supply flow, but the operating fan(s) is modulated to minimum output when the next fan is turned on. This sequence is used to avoid a stall of the starting fan. When all requested fans are running, they are modulated upward to satisfy duct static setpoint.

OTHER CONTROL MODES

Warm-Up Control

If warm-up control is used, it is not necessary to provide outdoor air. The following control actions should be accomplished when using warm-up control:

— Exhaust and relief fans should be off.
— Building pressurization control (if used) should be reset to zero static differential.
— Airflow tracking control (if used) should be reset to zero differential.
— If a return fan is used, the supply fan maximum airflow is limited to no greater than the return fan capacity. With digital control VAV systems, this is accomplished by commanding the VAV boxes to some percent of their maximum airflow setpoints during this mode.
— Space thermostats should change the warm-up mode to normal operation to prevent over or under heating.

Smoke Control

If smoke control is used, the return damper closes and the return fan operates as a relief or exhaust fan. Controls must prevent over pressurization of ducts and spaces.

Night Purge Control

Night purge can be used to cool a building in preparation for occupancy and to cleanse the building of odors, smoke, or other contaminants. Outdoor and relief air dampers must be open and the return damper closed. If airflow tracking is used, supply fan must be limited to the return fan volume. Some control systems allow space thermostats to be set at lower setpoints during this cycle to maximize free cooling. If digital control is used, significant energy savings can be accomplished by commanding all VAV box airflow setpoints to approximately 50 percent of their maximum values.

ZONE AIRFLOW CONTROL

AIRFLOW TRACKING/SPACE STATIC PRESSURE

Zone airflow control provides pressurization control for a portion of a facility. Figure 40 shows a zone airflow control example for a building. Airflow tracking or direct space static pressure control of the return damper on each floor determines the pressurization of each floor.
Airflow tracking is preferred for zone control on the first floor. Direct space static pressure control is difficult to stabilize because of sudden static pressure changes that occur whenever doors open. Also, the leakage around doors would require pressure control at very low setpoints, which are difficult to measure. On upper floors where building permeability is tight, airflow tracking control is not as viable. Direct space static pressure control is preferred for zone control on these floors. Relatively small differences in airflow tracking differentials of tightly sealed zones result in large pressure differentials. Also, all exhaust airflows must be included when dealing with airflow tracking, which makes the control more complex.

Both airflow tracking and direct space static pressure control require accurate sensing. In airflow tracking, the airflow sensors are located in supply and return ducts to sense total airflow. Minimum velocities and location of the airflow sensor relative to any variations from a straight duct are critical considerations. In direct space static pressure control, the indoor static pressure sensor should be in the largest open area and away from doors that open to stairways and elevators. The outdoor static pressure sensor should be at least 15 feet above the building (depending on surrounding conditions) and be specifically designed to accommodate multidirectional winds.

For zone control using airflow tracking or direct space static pressure, return fan control should hold duct pressure constant at a point about two-thirds of the duct length upstream of the return fan (Fig. 41). This control is the same as that used to control the supply fan, except that the duct pressure is negative relative to the ambient surrounding the duct.

To ensure minimum outdoor airflow, an airflow sensor enables control and provides information on the quantity of outdoor air. The airflow sensor is located in the duct having minimum outdoor airflow (Fig. 42). The control modulates the outdoor air, return air, and exhaust air dampers to provide outdoor air as needed. Normally, the difference between total supply and return airflows, as determined by zone controls, provides the minimum outdoor air. Since each zone is set to provide proper pressurization and buildings which are sealed tightly require less outdoor air for pressurization, this control scheme ensures minimum outdoor air. If minimum outdoor air is increased, it does not affect building pressurization.
Essentially, the increase of outdoor air above that required to maintain building pressurization is done the same way as mixed air control except outdoor air is controlled by flow rather than mixed air temperature (Fig. 42). In colder climates, overrides must be included to avoid freezing coils.

**MULTIPLE FAN SYSTEMS**

Multiple fan systems are a form of zone airflow control systems. The same concepts for zone pressurization using airflow tracking or direct space static pressure control apply to multiple fan systems. A return fan is modulated instead of the zone return damper to control zone pressurization.

**EXHAUST SYSTEM CONTROL TYPES**

Local exhausts are individual exhaust fan systems used in toilets, kitchens, and other spaces for spot removal of air contaminants. These fans are generally off/on types. They should be controlled or at least monitored from a central location as the exhaust airflow can significantly affect energy efficiency.

General exhausts route contaminants into common ducts which connect to a common exhaust fan. If the airflow is manually balanced, the exhaust fan runs at a fixed level. However, if the airflow is controlled at each entry to vary the airflow in response to the local need, duct pressurization control of the exhaust fan is required. It may also be necessary to introduce outdoor air prior to the general exhaust fan in order to maintain a minimum discharge velocity.

**FUME HOODS**

Fume hoods are the primary containment devices in most chemical-based research venues. The lab envelope itself becomes the secondary containment barrier. In all cases, the basic use of the fume hood is for the safety of the worker/researcher. Because no air is recirculated to the lab, the fume hood is also the primary user of energy in most labs. The continuing control challenge is to provide the safest possible environment while minimizing operating costs.

There are three types of general purpose fume hoods (Fig. 43): bypass, auxiliary, and standard. Bypass and auxiliary air hoods approximate a constant exhaust airflow rate as the fume hood sash opens and closes. Operation of the standard hood causes the face velocity to increase or decrease with the up and down movement of the sash as a fixed volume of air is exhausted (constant volume).

---

**Fig. 43. General Purpose Fume Hoods.**
The bypass hood limits face velocity to about twice the full sash open face velocity which may be acceptable. However, conditioned air is always exhausted making energy savings improbable.

The auxiliary air hood is a bypass type with a supply air diffuser located in front of and above the sash. If the make-up air through the diffuser is not conditioned as well as room air, some minimal energy savings result by employing this type of equipment. However, the performance of this hood is controversial regarding containment of materials in the hood, operator discomfort, and thermal loading of the laboratory. Its use is usually discouraged.

The standard hood can be controlled either by adding a face velocity sensor at the sash opening or by installing devices to measure the sash opening to calculate face velocity. See Figure 44. This information is then used to modulate a motorized damper, air valve, or variable speed motor to vary exhaust airflow and to maintain a near constant face velocity regardless of sash position. Since the hood removes non-recirculated, conditioned air from the space, significant energy savings can be realized by adding these controls to vary air volume and minimize the rate of exhaust. The other common method used to moderate energy usage is to provide two-position controls which control all hoods at one constant volume rate during occupied periods and a reduced constant volume when the lab is unoccupied. New technologies now available allow the air flow in individual fumehoods to be reduced when sensors determine no one is present at the face of the hood.

The subject of the "correct" face velocity is still debated. However, most research now indicates that 80-100 feet per minute (fpm) at the sash opening provides a zone of maximum containment and operating efficiency provided that the supply air delivery system is designed to minimize cross drafts. Velocities lower than this challenge the containment properties of the hood, and without specialized lab design and training in lab safety protocols, can create unsafe working conditions. Velocities higher than 120 fpm can cause excessive turbulence within the hood which not only compromise its containment properties but contributes to excessive energy usage.

Figure 45 illustrates a face velocity chart showing the comparative face velocities which may be experienced with different types of hoods using either constant volume or variable volume control strategies. The variable volume hood with face velocity controls in this example shows increased air velocities at the low aspect of sash closure because, in most cases, a minimum air volume is required to be continuously exhausted from the fume hood.

![Fig. 44. Variable Exhaust, Constant Face Velocity Fume Hood.](image1.png)

![Fig. 45. Comparative Fume Hood Face Velocities.](image2.png)
LABORATORY PRESSURIZATION

Constant supply airflow often is not capable of constant space pressurization in research laboratories because of the use of constant face velocity fume hoods and the use of other variable exhausts. To accomplish containment and prevent excessive pressurization requires some form of volumetric airflow control (airflow tracking) or control of differential pressure within the lab space (direct pressure control).

Airflow tracking (Fig. 46) measures all exhaust and supply airflows and maintains a relationship between the total exhaust and total supply. For space pressurization to be negative relative to adjacent spaces, the total exhaust must exceed the total supply. The difference between exhaust and supply airflows (offset) should be a fixed quantity for a particular space to keep the pressurization constant. A constant percentage offset value is sometimes used.

If future flexibility and changing lab configurations are important considerations, then flow sensor location, duct size, supply airflow rate, and control system design should all include capability to be modified in the future.

A characteristic of airflow tracking is stability of the system in the face of breaches to the lab envelope. This is most often lab door openings. In a laboratory maintained at a negative pressure, the space static pressure increases and the air velocity through all openings drops significantly when a door opens. Figure 47 shows a laboratory example with a single fume hood, a single door 36 in. wide x 80 in. high (20 ft²), and a crack area estimated at 0.5 ft². If the fixed airflow tracking differential is 200 cfm, the average velocity through the cracks would be 400 fpm which is more than adequate for containment. However, when the door opens, the average velocity in this example decreases to 9.8 fpm which is marginal to inadequate for containment.

Airflow sensors located in all supply and exhaust ducts provide flow signals which can be compared by a controller. Sensor locations must meet the manufacturers minimum installation guidelines, such as velocity range and length of straight duct before and after the sensor, to ensure accuracy. Materials and finishes for sensors in exhaust ducts exposed to corrosive fumes must be carefully selected.

However, the ability of the tracking system to quickly (usually within several seconds) react and compensate for door openings and other breaches is a positive characteristic of this control method.

Supply duct pressure and building pressurization control are simpler and more stable with airflow tracking because they are less affected by this type of unexpected upset. The supply duct pressure control remains stable due to fewer disruptions. Building pressurization, defined as the difference between total air leaving the building and the total air entering, remains the same.

Direct pressure control (Fig. 48) provides the same control function as airflow tracking but its characteristics are quite different. Direct space pressurization control senses the differential pressure between the space being controlled and a reference space which is usually an adjacent space or hallway.
Figure 49 shows a similar example of negative space pressurization utilizing direct pressure control. If the airflow through the hood is 1000 cfm and the pressure control reduces the supply airflow when the door is opened, the average velocity through openings drops from 400 fpm to 48.8 fpm.

When a door is opened, the space pressure control responds by reducing the supply airflow to zero and/or increasing general exhaust flow. Replacement air for the space that is being exhausted migrates from adjacent areas through the doorway and cracks. The supply system for the adjacent area must replace this air in order to maintain a positive building pressurization.

The significant issues are 1) how fast can the room pressurization system respond to upset (a door opening or several hoods being closed at once) and 2) what is the impact on adjacent areas and the rest of the building. Because of the inherent lag of direct pressure control systems (the time it takes the differential pressure sensor to know that several hoods have been closed) the lab can go into a positive pressure mode for a short period of time. Further, with extended door openings and other breaches it is possible for a direct pressure based system to call for amounts of exhaust air which may be drawn excessively from the adjacent spaces. This has the potential for cascading air flow and pressure effects throughout the building.

For reasons of speed and stability, volumetric tracking control is becoming the more accepted method of pressurization control in lab spaces.

Direct pressure control remains a viable alternative, especially in lab spaces that are sealed tightly, where there is sufficient building supply air and good lab operation protocols.
REFERENCES

The following references were useful in preparing this section on Building Airflow System Control Applications. Selected material was included from:

Design of Smoke Control Systems for Buildings
ISBN 0-910110-03-4

American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc.
1791 Tullie Circle, N. E.
Atlanta, GA 30329

Trane Air Conditioning Manual
The Trane Company
Educational Department
3600 Pammel Creek Road
LaCrosse, WI 54601

Engineering Fundamentals of Fans and Roof Ventilators
Plant Engineering Library
Technical Publishing*
1301 S. Grove Avenue
P. O. Box 1030
Barrington, IL 60010

Industrial Ventilation Manual
Committee on Industrial Ventilation
P. O. Box 16153
Lansing, MI 48901

Modern Development In Fluid Dynamics
Dover Publications Inc.
180 Variek Street
New York, NY 10014
