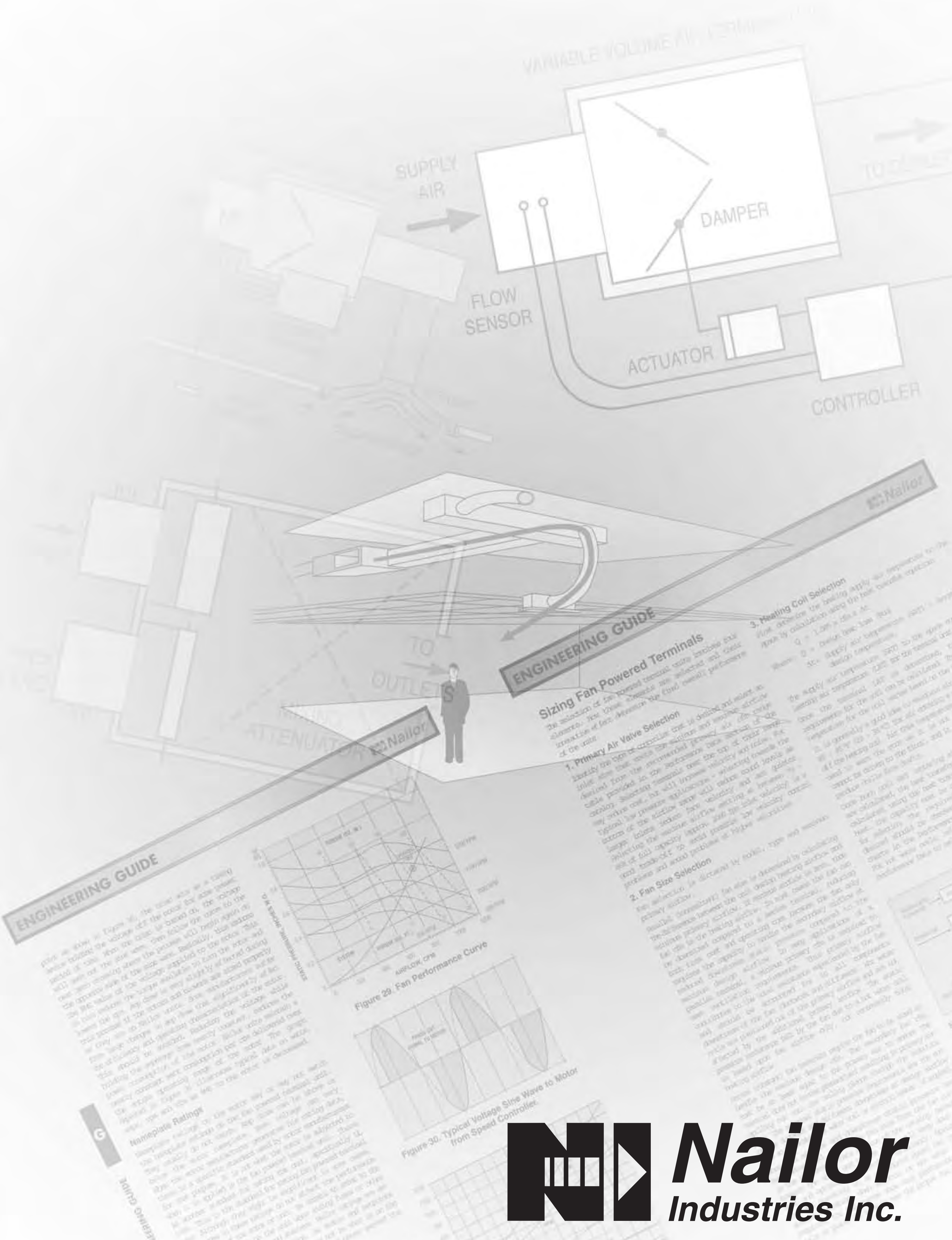
ENGINEERING GUIDE & INDEX



Nailor
Industries Inc.

Contents

System Selection

Designers have various systems to choose from when designing a building. Choosing which one to use is not always easy. The owners' needs must be met for installation, application and cost of operation. The designer must consider performance, capacity, reliability and spatial requirements and restrictions. The following guidelines describe different types of equipment and their general uses, restrictions and limitations.

Building Use

The designer must consider the intended building use as he begins to consider the type of equipment he will use. Office buildings with daily operational schedules frequently use fan powered terminal units. Usually fan powered terminals with auxiliary heaters (supplementary heat) would be used in the perimeter zones. These terminal units allow the greatest flexibility for individual zones while also allowing the central system to be turned off during unoccupied periods. During the unoccupied periods, the fan powered terminal units maintain the minimum or set back temperature levels without the help of the central air conditioning equipment.

Building Size

In large buildings, the central air handlers deliver large quantities of air to many zones with different needs. This is a perfect application for fan powered terminal units. Interior zones may not require heat at all; therefore they can be served either by single duct units or fan powered units with no supplemental heat. Unless the building is located in a tropical climate, the perimeter zones will require some type of heat, either electric or hot water. These are usually included with the terminal units, but sometimes baseboard heat is used. Buildings where the owner desires low operating costs usually employ series type fan powered terminal units, and the static pressure in the ducts is lowered to 0.5" w.g. (125 Pa) or less at the highest points. Interior zones in these buildings would require fan powered terminal units. Buildings with parallel type fan powered terminal units usually employ single duct units in the interior zones.

In shopping malls and other low rise buildings where each tenant area is small and in very small buildings, it is common to use small package air conditioners. If terminal units are employed on these systems, usually bypass units are selected. A variation of this system uses single duct units with a main bypass damper in the supply duct. The bypass damper is regulated by the static pressure in the supply duct. A nearly constant pressure can be maintained allowing the package units to operate at constant volume and the individual zones to be pressure dependent VAV.

Acoustical Constraints

Broadcast studios, theaters, and libraries require very low noise levels. Equipment selection and location is important here. If fan powered terminal units are to be used, careful examination of the equipment sound performance is imperative. RFI and EMI should also be considered when designing television studios.

Environmental Factors

Environmental factors include the climate and air conditions inside as well as outside the building. They also include legislative requirements such as outside air ventilation rates and local building codes. If high ventilation rates are required in interior zones, reheat will be required. In laboratories where high ventilation rates exist when multiple hoods are open, reheat is required. In zones where the load changes significantly during the day such as exterior zones in high rise office buildings that are affected by the season, solar loads and occupancy, fan powered terminal units are ideal. Single duct terminal units are usually employed where the load is usually stable.

Contamination Considerations

Hospitals, clean rooms and laboratories pose special problems. Operating rooms, bone marrow transplant, AIDS patient areas and clean rooms require positive pressurized environments. In addition to the pressure requirements, reheat coils and exposed fiberglass are usually avoided to eliminate the possibility of microbial growth in hospitals. Hospital rooms and clean rooms frequently also require constant and high ventilation rates which tend to favor dual duct terminal units. Patient housing for highly contagious diseases, such as tuberculosis, require negative pressure within the rooms to avoid allowing the germs to escape. Laboratories handling hazardous materials also require negative pressure areas. Single duct and dual duct constant volume terminal units have been usually selected for this type of building. New pressure independent ECM motor technology has lead to the development of fan powered pressurization units also for these applications.

Maintenance and Accessibility

Certain types of buildings such as clean rooms require high levels of reliability from terminal units due to the difficulty and cost associated with servicing or maintaining the equipment. In a clean room, for example, if the ceiling must be opened, the space may require disinfection before it can be used again. Associated costs would include lost production time as well as the cost for disinfecting the room. In cases like these, the equipment should be located outside of the clean room space or highly reliable, low maintenance equipment should be used.

Cost Factors

Costs must be considered before the final system selection is made. Installation, operation and maintenance all contribute to total cost. Sometimes one of these costs is more important than others. For example, if the owner/builder sells the building before construction begins, then his main concern will be construction costs, and operating costs will be unimportant. If the tenants pay their own utilities, operating costs are not a concern to the developer/builder. Electric heaters usually have a lower installed cost than hot water coils, but they may have a higher operating cost. Local rates will have to be researched to arrive at the correct decision before making the final selection.

System Selection (con't).

The following table presents a summary of the different types of terminal units currently available and their suitability for particular commercial building applications.

	Facility Type												
	Office Space, Educational & Institutional Buildings				Hospitals, Clean Rooms & Laboratories*			Noise Sensitive Applications #			Other Facilities		
	Large Building		Small Building										
Terminal Type	Interior Zone	Exterior Zone	Interior Zone	Exterior Zone	Patient Areas	Operating Areas	Laboratory Space	Broadcast Studios	Theaters	Libraries	Public Use	Shopping Centers	Hotels, Multi-Residential
Single Duct													
VAV Without Reheat	👉	👉	👉	👉	👉	👍	👉	👉	👉	👉	👉	👉	👉
VAV With Reheat	👉	👉	👉	👉	👉	👎	👍	👉	👉	👉	👍	👉	👉
Dual Duct													
VAV No Mixing	👉	👉	👎	👎	👎	👎	👎	👉	👉	👉	👉	👎	👉
VAV With Mixing	👉	👉	👎	👎	👉	👉	👎	👉	👉	👉	👉	👎	👉
Constant Volume	👉	👉	👎	👎	👍	👍	👉	👍	👉	👉	👉	👎	👉
Fan Powered													
Parallel With Heat	👎	👉	👎	👉	👎	👎	👎	👎	👉	👉	👍	👉	👉
Series Without Heat	👍	👉	👉	👉	👎	👎	👎	👍	👍	👍	👍	👍	👍
Series With Heat	👉	👍	👉	👍	👉	👎	👉	👍	👍	👍	👍	👍	👍
Low Temperature	👉	👍	👎	👎	👎	👎	👎	👉	👉	👉	👍	👉	👎
Bypass	👎	👎	👉	👉	👎	👎	👎	👎	👎	👎	👎	👎	👎

Table 1.

👍 = Preferred for this application.
👉 = Sometimes used for this application.
👎 = Not recommended for this application.
* = Sealed lining is recommended to minimize entrainment of airborne fibers into the occupied spaces.
# = Special consideration should be given to selecting very quiet operating equipment and use of attenuators.

Types of Terminal Units

All of the terminal units described below share several common components; corrosion resistant zinc coated steel casing, sound absorbing internal insulation with coated edges and an erosion resistant facing and a throttling damper to control conditioned air. Associated controls may be pneumatic, analog electronic or digital.

Single Duct

Description

Basic unit consists of a damper, actuator, flow sensor and selected controls. Accessory discharge attenuators and multiple outlet attenuators are also frequently used.

Operation

The terminal resets the volume (variable air volume) of conditioned air delivery to the space in response to the room thermostat. The terminal can handle hot or cold air. Occasionally, the terminal is used to control both hot and cold air, where a dual function thermostat and inlet temperature sensing with change-over controls are utilized.

Common Applications

Interior zones of a building which have a permanent cooling load and therefore no heating requirement.

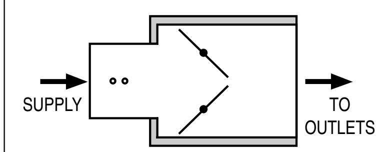


Figure 1. Single Duct
(Elevation)

Single Duct with Reheat

Description

Basic unit consists of a damper, actuator, flow sensor and selected controls as above with the addition of a heating coil (hot water or electric). Accessory discharge attenuators and controlled outlet attenuators are also frequently used.

Operation

The terminal resets the volume of conditioned cold air delivery to the space in response to the room thermostat. Upon a call for heat in the space the heating coil is energized and reheats the conditioned air. Electric coils are activated in stages upon thermostat demand and water coils are modulated using a proportional or two position on/off hot water valve.

Common Applications

1. Exterior zones (adjacent to outside walls or the upper floor in the case of multi-story buildings) where convective and radiated heat losses create an intermittent need for moderate heating as the terminal usually reheats at the minimum setting. An auxiliary higher minimum setting is available as an option with additional controls.

2. Interior zones where ventilation requirements preclude full shut-off of the terminal or minimum airflow requires some added heat.

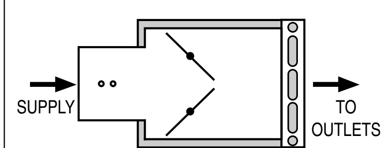


Figure 2. Single Duct with Heating Coil
(Elevation)

Dual Duct (Non-mixing)

Description

Essentially two single duct boxes side-by-side. Basic unit incorporates separate cold and hot air inlets and volume control assemblies consisting of a damper, actuator, flow sensor and selected controls.

Operation

The terminal unit resets the volume flow of either hot or cold air (without mixing) to the space in response to the room thermostat. Air is supplied from a dual duct central air handling unit. There is no provision for mixing and therefore hot and cold air should not be supplied simultaneously as stratification in the discharge duct will occur, causing uneven temperature discharge from outlets.

Common Applications

Exterior zones in buildings (such as hospitals) where overhead heating and cooling is desired but use of auxiliary hot water coils is not feasible and zero to low minimum flow is acceptable during changeover.

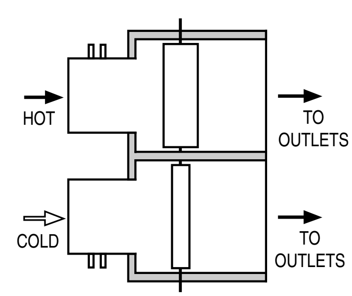


Figure 3. Dual Duct, Non-mixing
(Plan View)

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Dual Duct (Mixing)

Description

Basic unit incorporates separate cold and hot air inlets and volume control assemblies consisting of a damper, actuator, flow sensor and selected controls, and a common mixing/attenuator section which minimizes stratification of the discharge airstream.

Operation

The terminal unit resets the volume flow of the hot and cold air supply ducts in response to the room thermostat. Airflow delivery to the space may be variable volume (with a minimum flow established through a mixing of the two airstreams) or constant volume.

Common Applications

Interior and exterior zones in buildings (such as hospitals) where overhead heating and cooling is desired but use of an auxiliary heat coil is not feasible.

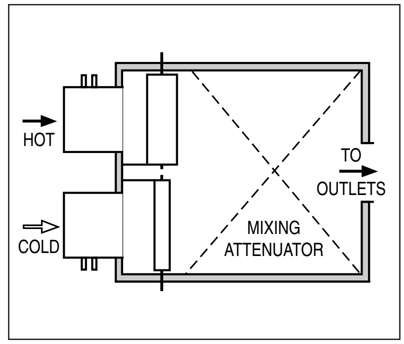


Figure 4. Dual Duct, Mixing
(Plan View)

Fan Powered Series Flow (Constant Volume)

Description

Basic unit consists of a primary air damper, actuator, flow sensor, blower/motor (with flow adjustment), and selected controls. Accessory heating coils either hot water or electric are also generally required.

Operation

The primary air damper throttles conditioned cold air in response to the room thermostat and delivers this air stream to the mixing chamber upstream of the blower/motor located in series with the primary airflow. The blower/motor then delivers a constant volume of air to the space. Upon demand for maximum cooling, the airflow is derived entirely from the conditioned air supply. As the cooling demand diminishes, the primary damper reduces the conditioned air supply and the blower/motor compensates for this reduction by inducing make-up quantities of plenum air from the ceiling plenum thereby reclaiming otherwise wasted heat and mixing it with the conditioned air to maintain a constant volume variable temperature delivery of air to the space. Upon further reductions in space temperature, the supplemental heating coil is energized. The result is a constant volume of air diffusion to the space while the central system encounters a variable volume distribution system.

Common Applications

1. Exterior zones where heating and cooling loads may vary considerably and occupancy variances allow the central system to be shut-down or set-back during unoccupied hours.

2. Situations where central system economy is desired as central fans can be reduced in size because they only need to provide sufficient static to deliver air to the terminal.

3. Where occupant comfort is very important since the constant volume air variable temperature delivery produces optimal air distribution and optimum ventilation.

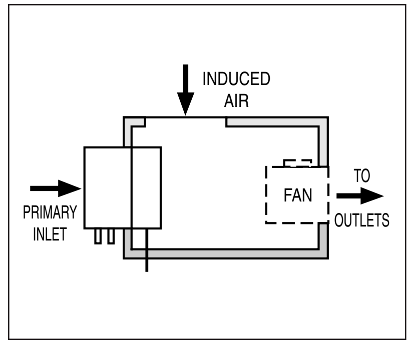


Figure 5a. Fan Powered, Series (Constant Volume)
(Plan View)

Fan Powered "STEALTH" High Performance Extra Quiet Series Flow (Constant Volume)

Description

A terminal similar to above, but incorporating special design and construction features that provide unusually quiet operation.

Operation

As described on page above.

Common Applications

As described above, but premium performance and high quality construction are ideally suited to high profile design projects and applications requiring minimum noise. Especially suitable for larger zones than standard series flow fan powered terminal units, as reduced radiated sound levels can lower first cost.

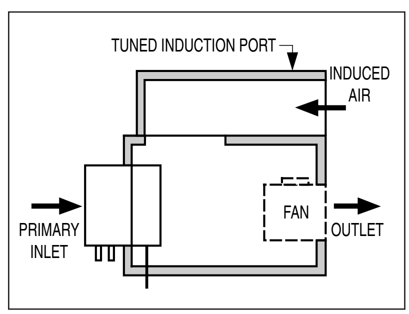


Figure 5b. Fan Powered, "STEALTH" Series (Constant Volume)
(Plan View)

Low Profile Fan Powered Series Flow (Constant Volume)

Description

Similar in construction to the standard series flow terminal described earlier, but only 11" (279) in height for all sizes, to minimize the depth of ceiling space required. Unlike standard fan powered terminals, the fan/motor assembly is installed flat on its side as shown in the diagram. Sound power levels are somewhat higher than standard units. **A "Stealth" model is also available from Nailor.**

Operation

As described on page G6.

Common Applications

As described on page G6. Where zoning requirements limit building height and the architect wishes to maximize the number of floors, these units will fit in a shallow ceiling plenum as small as 12" (305) deep.

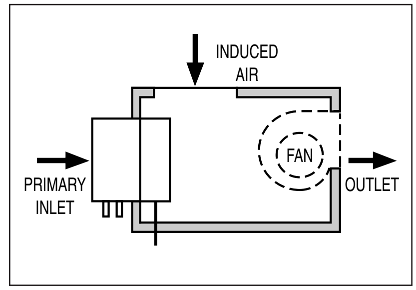


Figure 6. Low Profile Fan Powered, Series Flow
(Plan View)

Outside Air Inlet Fan Powered Series Flow (Constant Volume)

Description

Similar in construction to the standard series flow terminal described earlier, but with the addition of a secondary air inlet that provides a direct connection for outside air. **A "Stealth" model is also available from Nailor.**

Operation

As described on page G6. The second air inlet, which is usually smaller than the primary air inlet, provides a constant volume of outside air to the zone, ensuring the minimum ventilation air requirements are met.

Common Applications

General building applications described on page G6 where maintenance and assurance of high Indoor Air Quality (IAQ) standards are a prime concern e.g. schools.

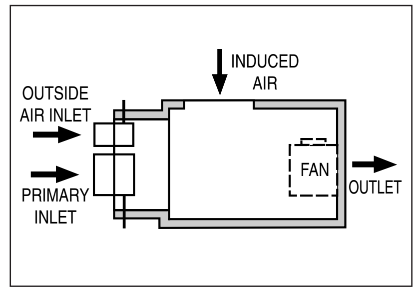


Figure 7. Outside Air Inlet Fan Powered, Series
(Plan View)

Low Temperature Fan Powered Series Flow (Constant Volume)

Description

Same as Fan Powered Series (Constant Volume) with the addition of a special vapor barrier lining and a thermally isolated inlet collar to prevent condensation for use with "cold air" systems.

Operation

Same as Fan Powered Series (Constant Volume) description above.

The maximum cold air volume is established lower than the fan delivery volume in order to maintain the minimal mixing required to raise and temper the unit discharge air temperature to a level acceptable for introduction to the occupied space, usually 55°F (13°C), with standard air outlets and to maintain ceiling coanda effect.

Common Applications

This unit is used with chilled water/ice storage systems that are designed to provide low temperature [40 – 48°F (4 – 9°C)] central system air distribution to the zone terminals.

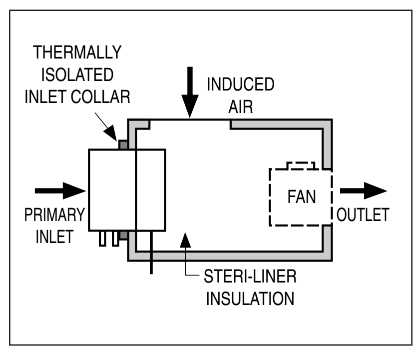


Figure 8. Fan Powered, Low Temperature
(Plan View)

Underfloor Fan Powered Series Flow (Constant Volume)

This low profile terminal is designed to fit between the pedestal support grid of raised, or access floor system HVAC designs, without any modifications to the floor. Available in two unit sizes, only 8" (203) or 11" (279) deep and 20" (508) wide.

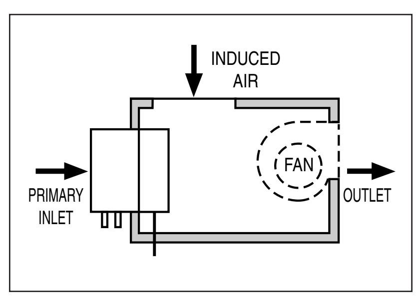


Figure 9. Underfloor Fan Powered, Series
(Plan View)

Fan Powered Parallel Flow (Variable Volume)

Description

Basic unit consists of a primary air damper, actuator, flow sensor, blower/motor assembly (with flow adjustment) and selected controls. An accessory heating coil, either hot water or electric, is usually required to satisfy space load conditions.

Operation

The primary air damper throttles the conditioned cold airflow in response to the room thermostat. As the room temperature decreases, the primary damper throttles toward its minimum flow setting and the unit blower, situated in parallel outside the primary airstream, is energized to provide warm ceiling plenum air to the space. A further drop in space temperature energizes the supplementary heating coil. The resultant control provides variable volume air diffusion to the space as well as a variable air volume distribution system to the central equipment.

Common Applications

This terminal is used primarily in exterior zones of buildings where varying occupancy allows the central system to be shut down during unoccupied periods. The unit blower and accessory heater provide heating as required to maintain minimal space temperatures during the shut down periods.
As the fan handles only a reduced heated air volume, the fan can be sized smaller than a series flow terminal. A backdraft damper prevents reverse flow through the fan during the cooling cycle.

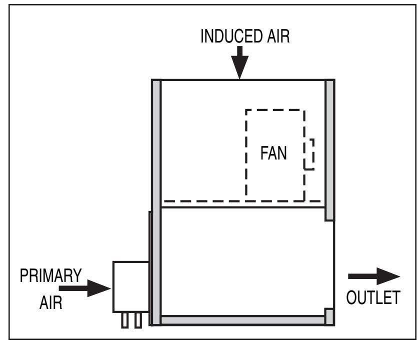


Figure 10. Fan Powered, Parallel (Variable Volume)
(Plan View)

Bypass Terminals

Description

Basic unit consists of a diverter type damper, actuator, bypass port and selected pressure dependent controls. A balancing damper is required ahead of the inlet. Accessory reheat coils are a common requirement.

Operation

The terminal delivers conditioned air to the space during periods of maximum cooling requirements (as determined and signalled by the room thermostat). As cooling demands diminish, the unit damper is modulated to bypass increasing amounts of conditioned air to the ceiling plenum. The result is a variable volume air supply to the space while a relatively constant volume of air is maintained across the central system air handling unit.

Common Applications

Bypass terminals are used primarily with packaged roof-top air conditioning equipment with a direct expansion coil where zoning is desired, but relatively constant airflows across the system components (i.e. coils, fans) are required. This allows the coil to operate at 100% airflow at all times in order to avoid freeze-up. The system offers an economical VAV supply design with low first cost. It does not provide the energy saving advantages of variable fan volume, but avoids the expense of a more sophisticated system.

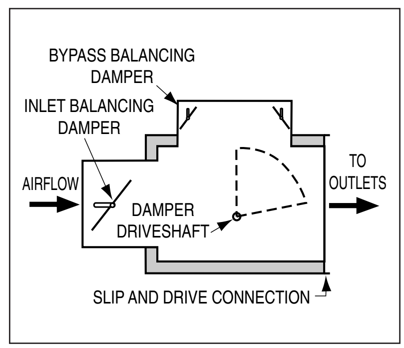


Figure 11. Bypass
(Elevation)

Introduction to VAV Terminal Controls

The control of air temperature in a space requires that the variable heating and/or cooling loads in the space are offset by some means. Space loads vary within a building and are influenced by many factors. These may include climate, season, time of day and zone position within the building, i.e. interior or exterior zone and geographic orientation. Other variable loads include people, mechanical equipment, lighting, computers, etc.

In an air conditioning system compensating for the loads is achieved by introducing air into the space at a given temperature and quantity. Since space loads are always fluctuating the compensation to offset the loads must also change in a corresponding manner. Varying the air temperature or varying the air volume or a combination of both in a controlled manner in response to changing load conditions will offset the space load as required.

The variable air volume terminal unit or VAV box allows us to vary the air volume into a room and depending on type selected, also lets us vary the air temperature into a room.

The VAV terminal unit may be pressure dependent or pressure independent. This is a function of the control package.

VAV terminals are the most energy efficient means of providing control as the central system supply may be sized based on the simultaneous peak demand of the total zones. The diversity factor allows a reduction in capacity as the central unit does not have to be sized for the sum of the peak demands of the entire building.

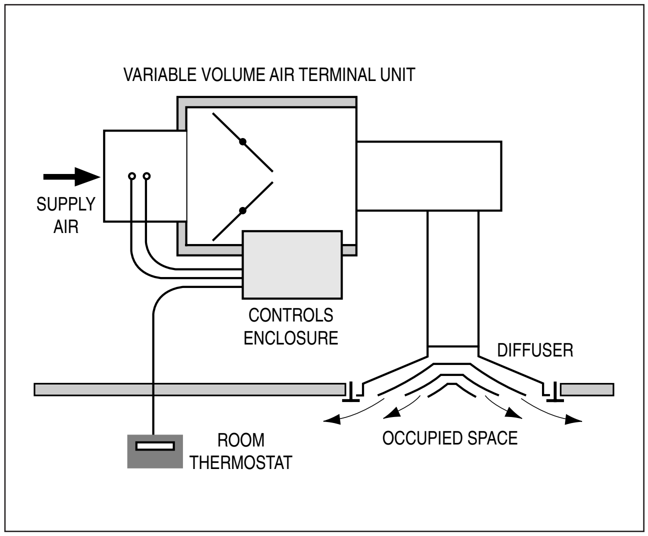


Figure 12. Typical Pressure Independent terminal unit controls and installation.

Pressure Dependent

A device is said to be pressure dependent when the flow rate passing through it varies as the system inlet pressure fluctuates. The flow rate is dependent on both the inlet pressure and the damper position of the terminal unit.

The pressure dependent terminal unit consists of a damper and a damper actuator controlled directly by a room thermostat. The actuator is modulated in response to room temperature only and acts as a damper positioner. (There is no flow sensor or reset controller).

Since the air volume varies with inlet pressure, the room may experience temperature swings until the thermostat repositions the damper. Excessive airflow may also lead to unacceptable noise levels in the space.

The logarithmic graph shown in figure 13a illustrates a pressure dependent terminals' reaction to duct pressure changes for several given damper positions. The line 1a – 1b represents one damper setting or position. As duct pressure increases, so does the airflow over the damper, with the flow rate varying in proportion to the square root of the static pressure drop across the terminal. This characteristic is typical of any fixed orifice or in this case, a throttling damper. Lines 2a – 2b and 3a – 3b represent additional random positions as the damper moves toward

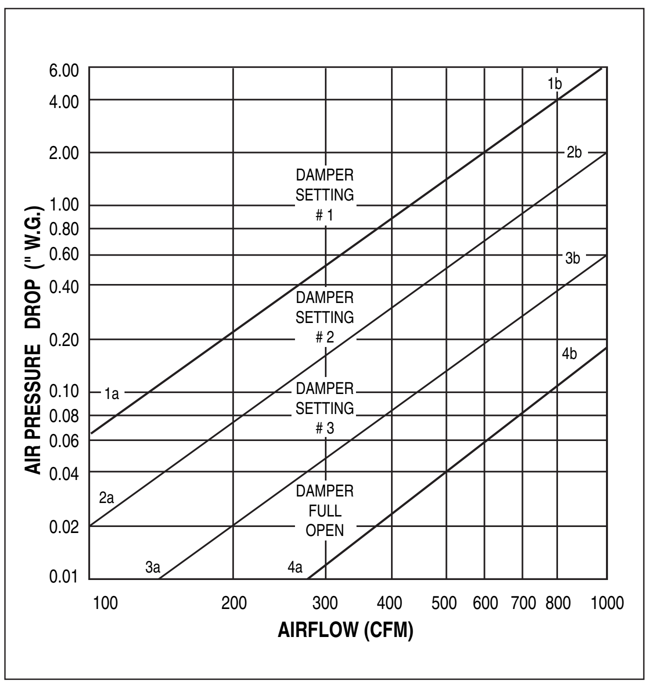


Figure 13a. Pressure Dependent terminal damper characteristics.

the full open position, line 4a – 4b. Pressure dependent terminals are therefore more prone to constant hunting when static pressures fluctuate at the terminal inlet, as the thermostat is responding to variations in flow that it didn't call for. Control accuracy is therefore poorer, when compared to a pressure independent terminal.

The pressure dependent terminal is for applications where neither pressure independence nor airflow limit regulation is required at the terminal. An example is a constant volume central air supply where the downstream static pressure is held constant by other controls. Another example utilizes a constant volume central fan and zone bypass dampers that respond to static pressure variations and short circuit excess air directly back to the air handler.

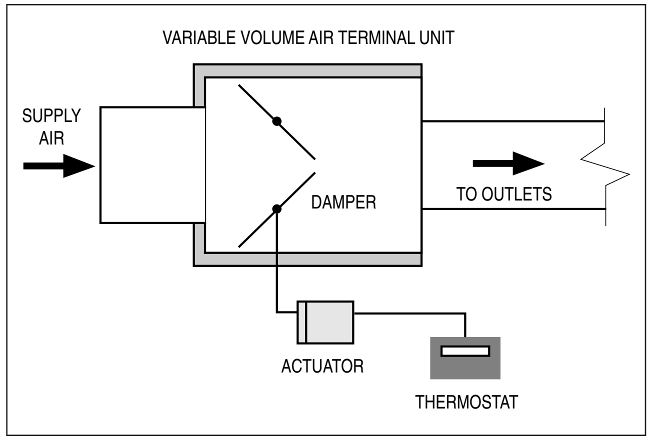


Figure 13b. Pressure Dependent terminal controls.

Pressure Independent

A device is said to be pressure independent when the flow rate passing through it is maintained constant regardless of variations in system inlet pressure.

The pressure independent control is achieved with the addition of a flow sensor and flow controller to the VAV box. The controller maintains a preset volume by measuring the flow through the inlet and modulating the damper in response to the flow signal. The preset volume can be varied between calibrated minimum or the maximum limits by the thermostat output.

The logarithmic graph shown in figure 14a illustrates pressure independent terminals' typical airflow settings and characteristics. The vertical lines 1a – 1b and 3a – 3b represent the calibrated minimum and maximum airflow settings respectively, that are adjusted at the flow controller. Line 2a – 2b represents any intermediate airflow setting maintained by the flow controller in response to thermostat demand. The damper will modulate (open and close) as required to hold the airflow setting constant up and down this vertical line regardless of upstream static pressure variations. Airflow will only change when the thermostat signal (demand) changes. The vertical lines are cut off by the diagonal line 1a – 3a, which represents the minimum operating static pressure requirement of the terminal unit for the given airflow – the pressure drop across the terminal with the damper in the fully open position.

Pressure independence assures the proper distribution of air to the conditioned space as required and allows the engineer to know that the design limits specified will be maintained. Maximum and minimum airflow limits are important for maintaining proper air distribution.

- Maximum airflow limits prevent over-cooling and excess noise in the occupied space.
- Minimum airflow limits assure that proper ventilation is maintained.

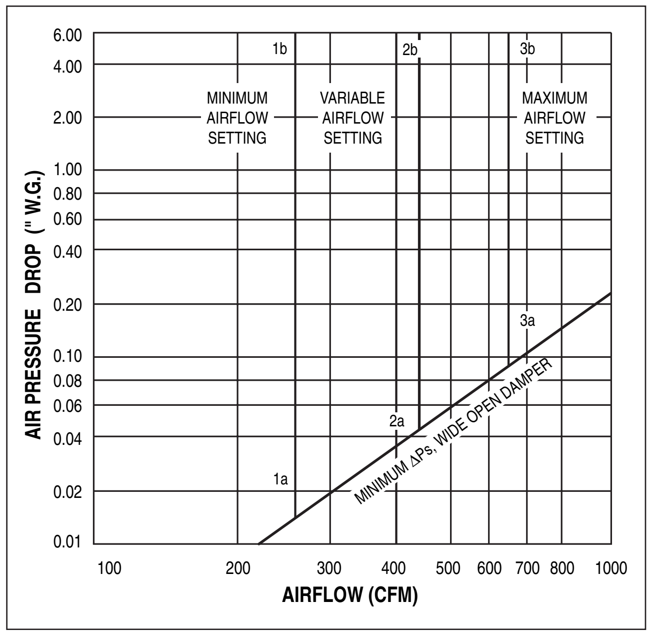


Figure 14a. Pressure Independent terminal damper characteristics.

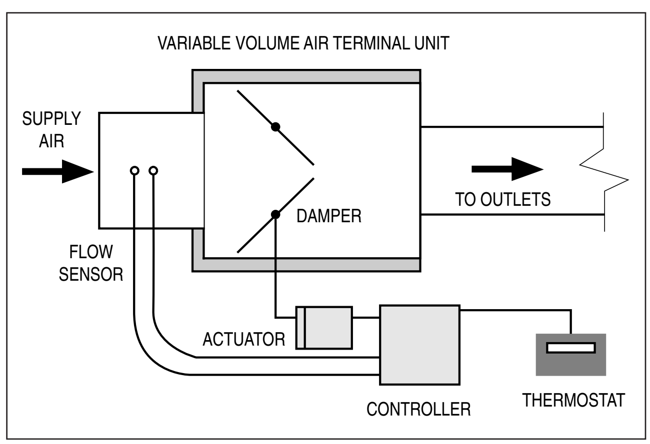


Figure 14b. Pressure Independent terminal controls.

Types of Controls

The various VAV controls available may include some or all of the following common components:

a) Flow Sensor/Pick-up

This device monitors the primary air inlet, measures air velocity and provides a feedback signal to the controller which directs the operation of the damper actuator. This control loop is the essence of the pressure independent operation.

b) Room Thermostat or Temperature Sensor

A room thermostat senses the room temperature, allows set point adjustment and also signals the controller to direct the damper actuator accordingly. Digital controls utilize a temperature sensor. Setpoint changes are managed by the digital controller.

c) Flow Controller

This device is 'the brain' and receives the signals from the Flow Sensor and the Room Thermostat or Temperature Sensor and processes the data to regulate the damper actuator.

d) Damper Actuator

This device receives the commands from the controller and opens or closes the damper to change or maintain the required airflow setting.

Electric Systems (Pressure Dependent)

Electric controls operate at 24 VAC powered by a transformer usually mounted within the control box of the terminal. These systems have no velocity sensor and no controller. There is no compensation for duct pressure fluctuations.

Traditional:
The room thermostat has single-pole-double-throw (SPDT) contacts. A rise in temperature drives a slow cycling damper actuator open in the cooling mode and a fall in temperature reverses the actuator in the heating mode.Thermostat response time to room temperature change is typically less than the actuator response time. Control is sluggish and large temperature swings may result.

State of the Art: Micro-processor based P + I thermostat provides superior control.

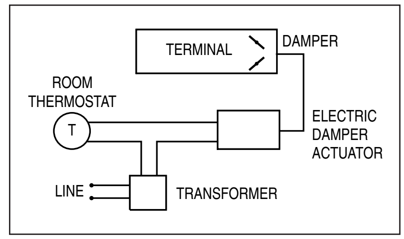


Figure 15. Electric Control Schematic.

Pneumatic Systems (Pressure Independent)

The pneumatic control system components are powered by compressed air at 20 – 25 psi from a central system.

The room thermostat receives main air at full pressure directly from the air supply. In response to room temperature changes, the air pressure is modulated to the controller, which regulates the damper actuator and provides proportional control. A rise in temperature drives the actuator open in the cooling mode and a fall in temperature reverses the actuator in the heating mode.

The sensor and controller compensate for changes in duct pressure, so that operation is pressure independent.

The controller allows the thermostat to modulate the airflow as the room temperature dictates from a preset minimum to a preset maximum.

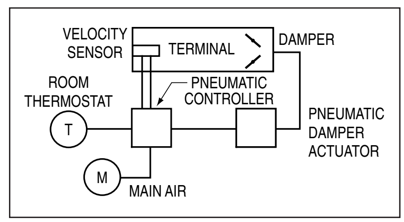


Figure 16. Pneumatic (PI) Control Schematic.

Analog Electronic Systems (Pressure Independent)

Analog electronic controls operate at 24 VAC powered by a transformer usually mounted within the control box of the terminal.

The electronic controls feature a velocity sensor (either the hot wire thermistor or pneumatic multi-point type with an electronic transducer) and an electronic velocity controller. They provide a proportional control function.

The electronic thermostat is selected from one of four types; cooling, heating, cooling with reheat or cooling-heating. A three-stage reheat (two stages for fan powered terminals) or automatic heat/cool changeover relay can be furnished in the control box.

Analog electronic controls compensate for changes in duct pressure.

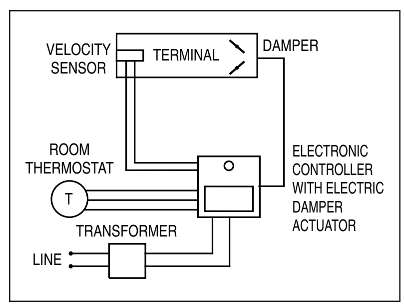


Figure 17. Analog Electronic Control Schematic.

Direct Digital Control (DDC) Systems (Pressure Independent)

These micro-processor based electronic controls also operate at 24 VAC powered by a transformer usually mounted within the control box of the terminal.

The flow signal from a pneumatic or electronic velocity sensor and signals from the room temperature sensor are converted to digital impulses in the specialized micro-computer controller. The program usually includes a proportional, integral and derivative (PID) control algorithm for excellent and highly accurate operation.

The controller not only performs the reset and volume control functions, but also can be programmed and adjusted either locally or remotely. It can link to other controllers and interface with fans, lighting and other equipment. Control can be centralized in one computer.

DDC Controls compensate for changes in duct pressure.

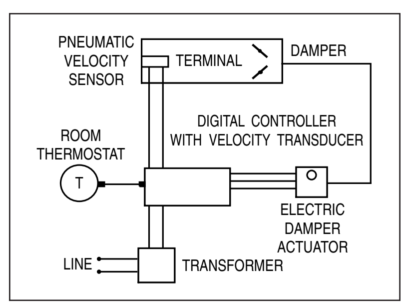


Figure 18. Digital Control Schematic.

Types of Controls (continued)

Digital Control Overview

A direct digital controller uses a digital computer to implement control algorithms on one or multiple control loops. Interface hardware allows the digital computer to process signals from various input devices. The control software calculates the required state of the output devices, such as valve and damper actuators and fan starters. The output devices are then positioned to the calculated state via interface hardware.

The basic principles of temperature control for heating, ventilation and air conditioning systems are well established. These control strategies have been implemented using pneumatic, electric, and analog electronic control devices. In this computer age, the micro-processor technology is now available in applications specifically designed for HVAC control. Micro-processor based controllers bring cost effective, state of the art computing power to the control of VAV terminal units, air handling units, packaged heating and cooling units, and entire building HVAC systems.

Micro-processor based controllers use direct digital control to replace conventional pneumatic or analog electronic controls. A direct digital controller takes input signals from sensors to generate numbers, processes this information digitally as directed by the programmed sequence of operation, and generates control action through binary on/off outputs or analog output voltages.

Controls Terminology

Thermostat Action (Figure 19).

Directing Acting means that a room temperature increase causes a corresponding increase in thermostat output (pressure or voltage).

Reverse Acting means that a room temperature increase causes a corresponding decrease in the thermostat output.

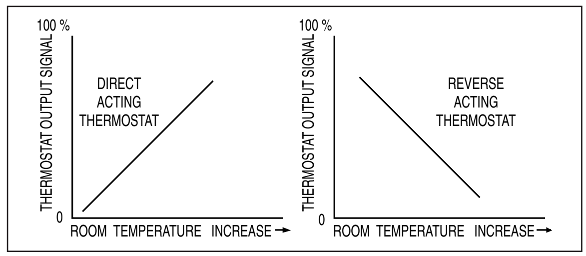


Figure 19. Thermostat Action

Damper Failure State

Normally Open indicates the fail safe position of the damper in a typical pneumatic system. When the control air pressure is removed or fails, the damper is opened by the actuator spring. Control air pressure is required to oppose the spring and close the damper or valve. (Figure 20).

Normally Closed indicates the fail safe position of the damper in a typical pneumatic system. When the control air pressure is removed or fails, the damper is closed by the actuator spring. Control air pressure is required to oppose the spring and open the damper or valve. (Figure 21).

Electric actuators as used with analog electronic and digital controls are typically of the non-spring return type and therefore the above usually does not apply.

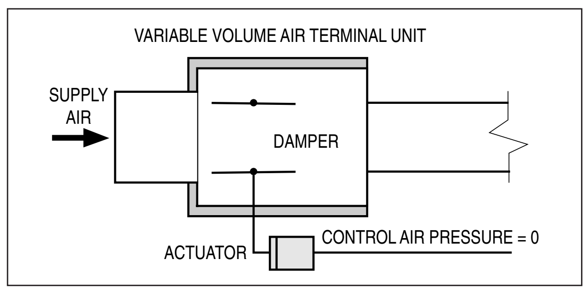


Figure 20. Normally Open Damper

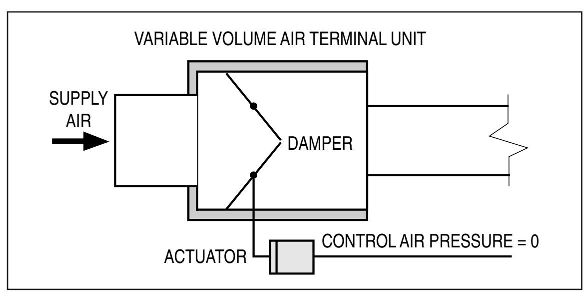


Figure 21. Normally Closed Damper

Direct Reset/Reverse Reset Pneumatic Velocity Controller Action (Figure 22).

In the direct reset controller, an increase in the thermostat output pressure causes a corresponding increase in controller airflow setting.

In the reverse reset controller, an increase in the thermostat output pressure causes a corresponding decrease in controller airflow setting.

The damper will open and close to maintain the setting when duct pressures change.

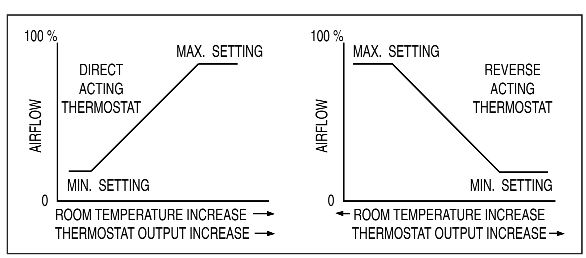


Figure 22. Velocity Controller Action

Pneumatic Thermostat/Controller Combinations (Figure 23).

For systems supplying cold air when a direct acting pneumatic thermostat signals a direct acting controller, an increase in room temperature produces an increase in airflow setting. A reverse acting thermostat with a reverse reset controller produces the same result. A direct acting thermostat with a reverse reset controller or a reverse acting thermostat with a direct reset controller will produce a decrease in airflow as the room temperature increases. With hot supply air, the logic is reversed.

Pneumatic Controller/Actuator Combinations

Controllers and actuators work in concert to control space temperatures. With pneumatic controls the most common combinations are DANO (Direct Acting Normally Open) and RANC (Reverse Acting Normally Closed). With most pneumatic controls, dedicated controllers are used for direct and reverse acting and any combinations other than DANO or RANC require extra components and increase air consumption. The Nailor 3000 Universal controller requires no extra components as the unit is switchable.

When electric heaters are used, the most common is RANO. Normally open is the most popular configuration for warm climates because the damper fails to a cooling position. RA fails the heater off.

Typical Operation of a Pneumatic Velocity Controller

The **thermostat set point** is the desired value (room temperature) of the controlled variable. When thermostat output equals this value, the control system is in equilibrium. Most pneumatic thermostats are factory calibrated @ 9 psi thermostat output. This calibration setting may be field adjusted. (Figure 24).

The controller **control point** is the airflow setting that the thermostat is signalling at any given moment and represents the actual equilibrium value of the controlled variable. **Offset** is the difference between the set point and the actual control point at any given moment in time. The damper opening may vary widely to compensate for any duct pressure changes reported by the inlet sensor, and to hold the airflow constant.

The range of values of the controlled variable over which the output of the controller goes from maximum to minimum airflow setting at the controlled device is called the **reset span** or throttling range. This band is adjustable on the controller.

The set point (9 psi in the example) is offset by the action of thermostat anywhere between the **maximum** and **minimum airflow** settings of the controller as room load changes. The corresponding thermostat output pressures are called the **start** and **stop points**. The start point is adjustable on the Nailor 3000 controller.

The thermostat may also control an auxiliary unit, such as a proportioning valve on a hot water coil, modulating over a range of 3 to 8 psi, in sequence with the reset span of the controller. The overall range over which the thermostat controls these devices is its proportional band (3 to 13 psi in the example).

Thermostat Sensitivity

The change in output pressure caused by a change in room temperature (Figure 25). Usually this is 1°F = 2.5 psi for pneumatic systems. In this case therefore, the proportional band, 3 – 13 psi represents a temperature range of 4°F.

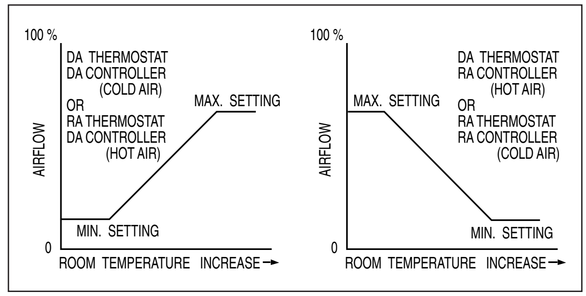


Figure 23. Thermostat/Controller Interaction

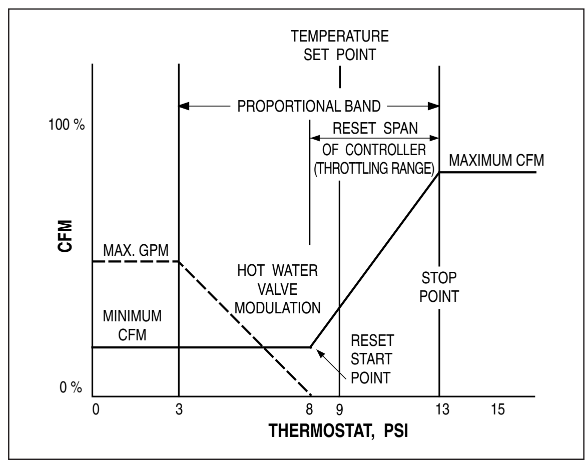


Figure 24.

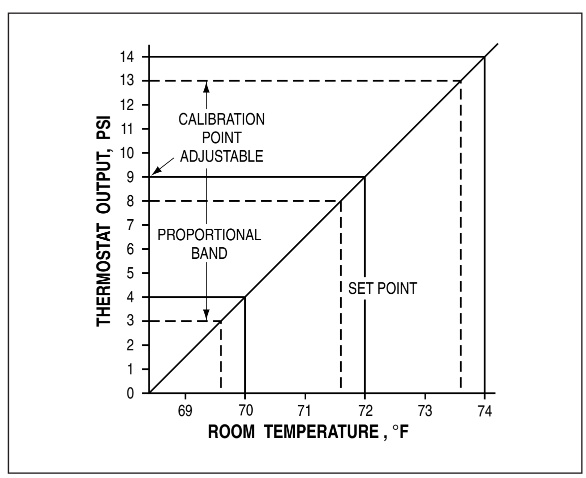


Figure 25.

G

ENGINEERING GUIDE

Features of Series and Parallel Flow Fan Powered Terminal Units

General

Fan powered variable air volume terminal units are the most economical, and consequently the most popular, way to heat and cool many types of buildings today. Typically used for exterior zones, they have advantages for interior zones as well.

Applications

Series

Series units, sometimes called Constant Volume Units because the fan runs constantly, are typically installed in the ceiling plenum. Induction air is either from the ceiling plenum or occasionally ducted from the conditioned space.

Parallel

Parallel units, sometimes called Variable Volume Units because the fan is intermittent and the cooling is variable volume, are typically installed in the ceiling plenum. Induction air is either from the ceiling plenum or occasionally ducted from the conditioned space.

Configuration

Series

The fan and VAV damper are aligned so that all the conditioned air that enters the mixing section as well as all the induced air that enters the mixing section must go through the fan to exit the unit and enter the occupied space. The mixing section is between the VAV damper and the fan. See figure 26 below.

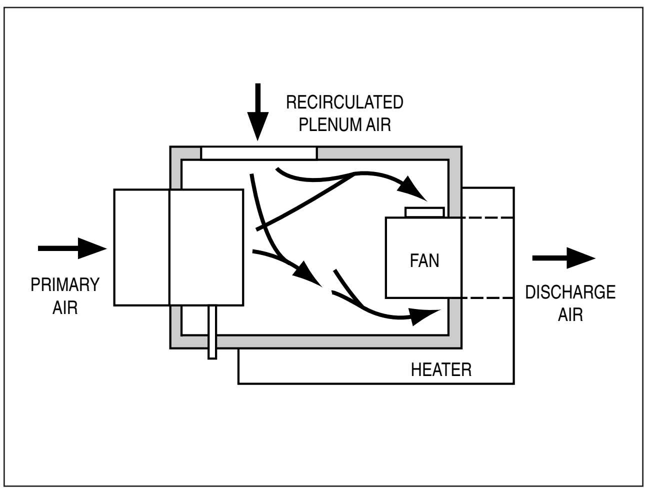


Figure 26. Series Flow Terminal Configuration

Parallel

The fan and VAV damper are aligned so that all the induced air enters the fan, but the conditioned air bypasses the fan and mixes with the induced plenum air on the discharge side of the fan. See figure 27 below.

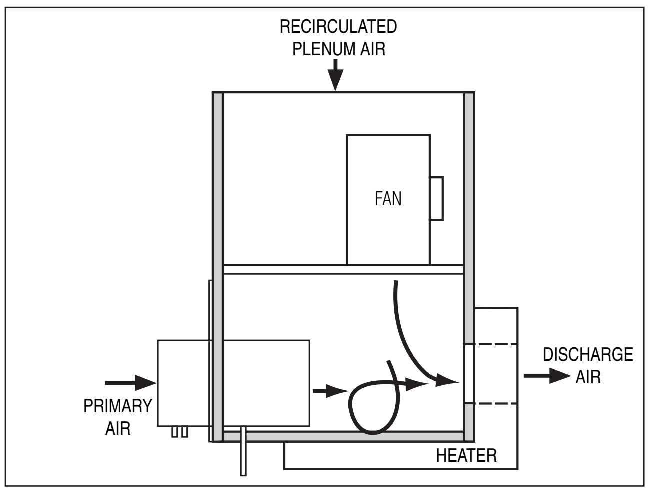


Figure 27. Parallel Flow Terminal Configuration

Fan Design

Series

Typically the fan runs continuously supplying a constant volume to the space. Some DDC controls manufacturers provide an optional analog output on their controller which may be used for controlling fan speed via the Building Management System. This allows dynamic fan speed control which may be either modulating or multiple speed operation from a single speed motor. Usually this would require Nailor EPIC™/ECM fan volume control technology. The fan must be sized to match the maximum airflow to be supplied to the zone. These units usually have larger fans than similar zones with parallel units. Fan energy consumption is constant during occupied periods.

Parallel

Typically the fan runs intermittently supplying a constant volume to the space while it runs during a call for heating. The fan must be sized to match the heating airflow to be supplied to the zone. These units usually have smaller fans than similar zones with series units. Fan energy consumption is intermittent during occupied periods when heating is required.

VAV Cooling and Inlet Static Pressure Requirements

Series

All the savings of VAV operation at the air handler and at the chiller are retained by using the series unit. Additional savings compared to single or dual duct VAV are realized due to the low inlet static pressure requirement of the Nailor 35S. Since the air handler is only required to push the conditioned air through the ducts to the unit and across the VAV damper into the mixing section, the pressure at the air handler can be greatly reduced. Nailor 35S units require only 0.05" w.g. (12 Pa) static pressure at the inlet to operate properly. This is much less than the competition. Using the 35S allows the duct designer to reduce the minimum static pressure in the upstream ductwork to (typically) 0.1 to 0.2" w.g. (25 – 50 Pa) or whatever is required to allow 0.05" w.g. (12 Pa) at the terminal while allowing a further reduction in horsepower and static pressure requirement from the air handler.

Parallel

All the savings of VAV operation at the air handler and at the chiller are incorporated in the parallel unit. Like the single or dual duct VAV, the air handler must push the conditioned air through the ducts to the unit, across the VAV damper, into the mixing section, through the discharge duct from the unit and across the diffuser(s) into the room. Compared to other manufacturers, Nailor 35N units require very low static pressure at the inlet to operate properly. This means that the duct system should be designed for minimum static pressure, typically 0.5 to 1.25" w.g. (125 – 310 Pa). Then the air handler should be adjusted as low as possible to keep the minimum requirement at the farthest VAV terminal unit.

Control Sequence

Series

The fan runs constantly during occupied periods.

During full cooling, the controls open the VAV damper to its maximum set point, delivering primary air to the mixing chamber. If the fan is set at the same airflow as the primary air VAV damper, then no air is induced from the plenum. If the fan is set at a higher airflow than the VAV damper, as it would be in a low temperature application, then air is induced from the plenum to meet the set point of the fan. The primary air and the induced air are blended before they enter the fan. Constant volume, constant temperature air is then discharged into the downstream duct and into the conditioned space.

As cooling demand decreases, the VAV damper modulates to lower set points until it reaches its minimum set point. Reducing the primary air into the plenum increases the volume of warmer induced air into the mixing chamber. The unit delivers blended, constant volume, variable temperature air to the zone. The increased plenum air causes the discharge temperature to rise to nearly meet the plenum temperature taking advantage of the recaptured heat from lights, people and machinery.

Upon a further decrease in zone temperature, the controls will automatically energize the supplemental heat (optional equipment), either electric or hot water coils. The discharge temperature will increase as heat is applied.

As the temperature increases in the zone, the sequence will reverse.

Parallel

During full cooling, the controls open the VAV damper to its maximum set point while the fan does not run. Constant volume, constant temperature air is then discharged into the downstream ducts and into the conditioned space.

As cooling demand decreases, the VAV damper modulates to lower setpoints. The unit delivers variable volume, constant temperature air to the zone.

Upon a further decrease in zone temperature, the controls will automatically energize the fan. Fan air and primary air are blended in the mixing chamber on the discharge side of the fan. The increased plenum air causes the discharge temperature to rise to nearly meet the plenum temperature as the zone temperature continues to fall. This takes advantage of the recaptured heat from lights, people and machinery. Blended, variable volume, variable temperature air is delivered to the zone.

At this same point, the VAV damper will reach its minimum set point. At some preset zone temperature, supplemental heat (optional equipment), either electric or hot water coils will be energized. The discharge temperature will increase as heat is applied but volume will be constant beyond this point.

As the temperature increases in the zone, the sequence will reverse.

Fan Interlocks

Series

Sometimes series units are designed to run continuously. Usually, they are energized only during occupied periods or when needed for emergency heating during unoccupied periods. It is important to interlock the unit fan with the air handlers in the building to insure that they start during occupied periods. Series unit fans should be started ahead of the air handler to prevent back flow into the plenum and backwards rotation of the fan. Nailor 35S series units have a built-in, anti-backward rotation device; however, if the fan is allowed to rotate backwards at unusually high rpms before the motor is energized, the device can be overwhelmed by the backward momentum causing the motor to run backwards. Interlocking the unit fan with the air handler eliminates this problem. Interlocks can be airflow switches or relays to match the building management system.

Parallel

The fans in Nailor 35N units are designed to be energized as needed throughout the day. The primary air enters the mixing chamber at the fan discharge. When the fan is not energized, there is a positive pressure at the discharge of the fan. Typically, this would cause the blower and motor to rotate backwards. However, all Nailor 35N units are equipped with a backdraft damper at the fan discharge inside the unit. This damper prevents backward airflow through the fan and into the plenum. Nailor backdraft dampers are gasketed for low leakage and quiet operation.

Acoustics

Series

Series fans are sized to match the maximum airflow required in the zone. The fan runs constantly during occupied periods. There are two sound sources in the unit, the fan and the VAV damper. While both contribute to the overall discharge and radiated sound emitted from the unit, the fan is primarily responsible for discharge noise while both the damper and the fan are responsible for radiated noise. Usually the radiated noise into the room is the larger and therefore more critical of the two components.

Comparing the sound level between a series and a parallel unit in similar zones, the series unit might generate slightly more noise. The fan and damper would be at their peak when the unit operates at full cooling capacity, the worst position in the sequence of operation for noise generation. As the primary air decreases, the noise generated would eventually be only from the fan.

Damper noise must be considered, however, as the noise decreases with decreasing inlet static pressure. It would be possible to select a very quiet series unit if very low inlet static pressure were utilized along with a very quiet fan since both components would decrease the radiated noise significantly.

Fan noise is constantly emitted into the zone. If the building is designed well and the terminal units are selected correctly, the fan will be the major noise component.

Parallel

Parallel fans are sized to match the heating cfm required in the zone. The fan runs intermittently when heating is required. There are two sound sources in the unit, the fan and the VAV damper. Both the damper and the fan are responsible for radiated and discharge noise. Usually the radiated noise into the room is the larger and therefore more critical of the two components.

Comparing the sound level between a parallel and a series unit in similar zones, the parallel unit might generate slightly less noise. The fan and damper would never peak simultaneously. When the unit operates at full cooling capacity, the damper would be at its peak noise generation. During heating requirements, the fan would peak while the damper was at minimum noise generation.

Damper noise must be considered, however, as the noise increases with increasing inlet static pressure. Parallel units require much higher inlet static pressures at the unit.

Fan noise is constant into the zone when the fan is running; however, it is intermittent during much of the day. This fan cycling can be very annoying in the occupied space. Even if the overall sound level is lower than that of a similar series unit, the variation in sound levels in the space during the day can be much more noticeable than a higher constant sound level.

Energy Consumption

Fan powered VAV terminal units were originally designed and introduced to our industry for their ability to save energy. That is what makes them so necessary and popular. They take advantage of typical VAV savings at the air handler and the chiller during the cooling periods, but the real savings kick in when heating is required. Fan powered terminals induce warm plenum air from the ceiling and blend it with the primary air at minimum ventilation requirements during the heating sequence. This recaptures all the heat created in the zone and plenum by lights, occupants, solar loading, and machinery or equipment such as computers, coffee machines, copiers, etc. Then the unit returns this heat as free heating rather than wasting it back at the air handler. If additional heating is required, then supplemental heat is added to the sequence, but the unit still saves energy by warming blended air at 75°F (24°C) rather than reheating primary cooled air at 55°F (13°C), saving the cost of 20°F (11°C) at the heating airflow. Costs of operating the units pale in comparison to the savings over other systems.

Series

Series fans run constantly during occupied periods, and the fan is sized for the full airflow to the zone. This causes the energy consumption from the fan to be higher than that of a parallel fan in a similar zone.

On the other hand, series units are designed for very low inlet static pressures. This saves energy at the air handler compared to a parallel unit for a similar zone.

Parallel

Parallel fans run only when required during the heating sequence and deadband. The fan is sized for the heating airflow, which may be much less than the total airflow requirement for the zone.

As the mixing of minimum primary air and induced air takes place downstream of the terminal fan, the terminal inlet static pressure requirement is greater than a series terminal. This usually adds cost at the air handler. Some studies have shown this added cost to be in excess of the added operating cost of running the motors constantly in the series terminal configuration.

Fan Selection

Many times specifications call for fan selections to be made using medium speed on the blower motor. This can cause some concern with Nailor fan powered terminal units because we use only single speed motors, and consequently there is no medium speed for selecting the equipment sizes. There is a solution.

Examine a Nailor fan curve like the one shown below (figure 28). Then examine the comparative fan curve (also illustrated) for a competitor's unit using a 3 speed motor.

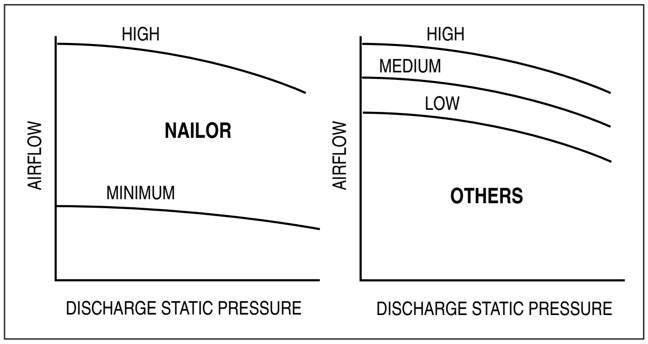


Figure 28. Nailor Fan Curve vs. 3 Speed Fan Curve.

It is important to note that the Nailor unit has a much *larger turn down ratio* than the unit with the 3 speed motor. Medium speed is not halfway between the high and low curves on the Nailor unit, but rather nearly 80% above the minimum speed curve and paralleling the high speed curve. This is typical of units which employ 3 speed motors.

When selecting a unit for a particular set of conditions, care should be taken to select the unit such that the air delivery is designed to meet the room sound and static requirements. Specific sound data can be found on the sound data sheets for various airflow deliveries for each unit. This should be the guiding factor in selecting unit sizes.

A simple rule of thumb is that when considering a unit selection for a typical office space, the fan should be selected for performance down from the high end performance by 20% to 25% of the distance to the low end curve at the specified external static requirement. This allows for very low room sound levels while maintaining some flexibility for future changes in the zone. If you are selecting equipment for large open areas where sound is not critical, select closer to the maximum cfm curve. If you are selecting equipment for a meeting room or an executive office, maybe you should select equipment slightly below center. If you are selecting equipment for an auditorium or some similarly sensitive area, select operation very near the minimum curve.

Avoid selecting equipment right on the maximum or minimum curves. This leaves no flexibility in the equipment for future changes.

Fan Airflow Control on Fan Powered Terminals

Introduction

When designing air systems and using fan powered VAV terminal units, it is as important to match the fan air to the space requirements as it is to match the primary air. This is even more true on series units than on parallel units. To facilitate this process, Nailor Industries designed their units to work over a wide range of adjustability. Some competitive products are not as friendly to adjust. The two commonly used methods are electronic fan speed control and mechanical trimming.

Fan Shift in Series Fan Powered Terminal Units

Before adjusting the fan, the possibility of fan shift must be considered. Some VAV terminal units suffer from a condition known as fan shift. This occurs when the blower is subjected to variations in pressure on the inlet side of the fan. As the primary damper changes from full cooling to minimum cooling, the pressure drop caused by the induction mixing chamber and associated inlet attenuators may cause the fan to shift its performance as it rides the fan curve. Consequences from the phenomenon vary from building to building and zone to zone, but if diffusers add background masking noise at design flow, then the noise levels will change as the volume changes and this can be very annoying. Design ventilation rates can also vary. These are serious problems and that is why Nailor series fan powered terminal units are designed to eliminate fan shift.

Mechanical Trimming (PSC Motors)

Mechanical trimming involves the use of a damper, usually manually adjustable, and usually employs a 3 speed motor. The damper can be located either in the induction port opening or on the discharge of the fan. Usually, it is on the fan. Mechanical trimming is used to balance the fan airflow. After "ball parking" the fan by selecting the fan speed, high, medium or low, then the damper is adjusted to fine tune the desired airflow. While this is a lower first cost option than the voltage adjustment speed controller, it causes higher operating costs and higher noise levels. When adjusted, the damper will regulate the airflow by raising the static pressure at the fan. The fan must then overcome the higher static levels. This increases rpm, thereby increasing tip speed, air velocity and vibration. Noise goes up. The fan will ride the fan curve similar to the one shown in figure 29. Airflow drops and power consumption drops, too. However, the power consumption does not drop as fast as the cfm drops. Overall efficiency diminishes as the damper throttles the fan.

Electronic Fan Speed Control (PSC Motors)

Nailor fan powered terminals equipped with PSC motors (standard) feature SCR solid state speed controllers.

Electronic fan speed controls utilize a triac to adjust the fans electrical AC voltage. This is called phase proportioning or wave chopping. When the sine wave crosses the zero point as shown in figure 30, the triac acts as a timing device holding the voltage off the motor for some preset period of time. When the triac is turned on, the voltage will seek out the sine wave, then follow the curve to the next zero crossing where the process will begin again on the opposite side of the sine wave. Basically, this reduces the RMS value of the voltage supplied to the motor. This in turn reduces the torque available to turn the rotor and lowers the rpm. Amp draw is very slightly affected during this process if the motors and blowers are sized properly as they are on Nailor units. Some manufacturers suffer from large changes in amp draw that significantly affect the efficiency and operating characteristics of the motor. This should be avoided. Reducing the voltage while holding the amperage draw nearly constant, reduces the power consumption of the motor. Nailor units maintain a nearly constant watt consumption per cfm delivered over the entire operating range of the motor. The graph depicted in figure 31 illustrates typical data on watts, amps, rpm and cfm as RMS to the motor is decreased.

Nameplate Ratings

Nameplate ratings on the motor may or may not match the nameplate ratings on the fan powered terminal unit. They usually do not match. Amp draw can be above or below the motor nameplate. Even voltage can vary. When the motor manufacturer generates his rating data, there is a specific standard used by motor manufacturers for that purpose. It is not what the motor is subjected to when it is applied in the fan powered terminal unit. There is another standard for rating the unit, specifically UL 1995. This is the standard for rating fan powered terminal units. Although they might be significant in some cases, differences in these ratings do not affect the performance or lifetime of the motor or unit. Be careful to refer to the nameplate ratings on the unit when sizing fuses or other overcurrent protection and starters. Nailor ratings are set at the worst possible condition. As static and setpoints vary on each unit, performance may not be what is on the unit nameplate, but amp draws should never exceed the unit nameplate.

Caution on Meters

Many digital multi-meters are not designed for true RMS readings. Using these meters when measuring amps or voltage on the motor in the fan powered terminal unit can result in erroneous readings. To measure the correct current and voltage, a true RMS DMM designed for this type of sine wave is required. These meters can be relatively expensive.

Nailor EPIC™/ECM Motor Technology

The recent introduction and availability of brushless DC motors as an option for series fan powered terminal units is rapidly replacing the PSC induction motor. These motors provide significant energy savings and superior controllability. See page D9 in the catalog for a complete explanation.

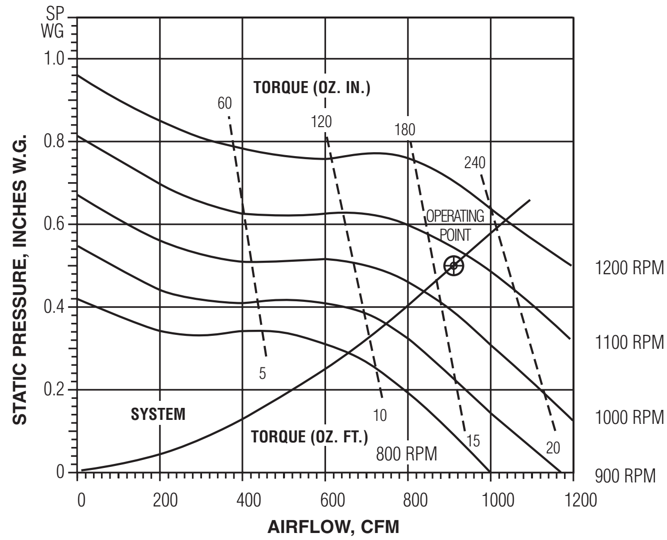


Figure 29. Fan Performance Curve

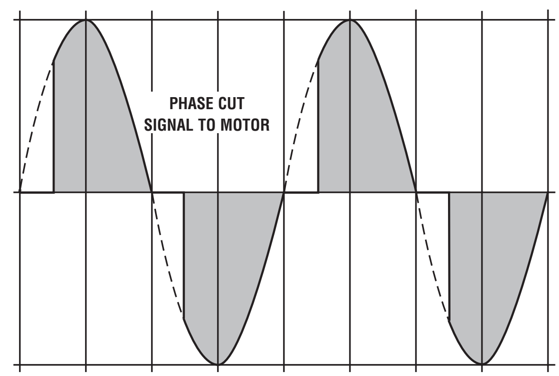


Figure 30. Typical Voltage Sine Wave to Motor from Speed Controller.

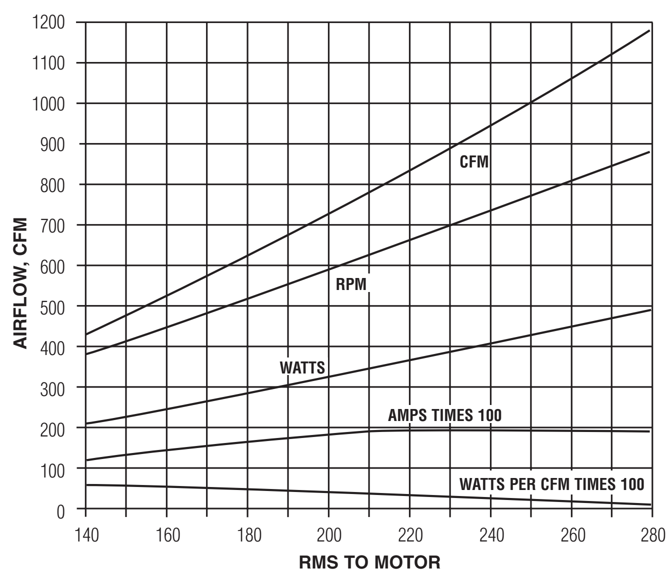


Figure 31. Typical Motor Data

Sizing Fan Powered Terminals

The selection of fan powered terminal units involves four elements. How these elements are selected and their interactive effect determine the final overall performance of the units.

1. Primary Air Valve Selection

Identify the type of controller that is desired and select an inlet size that meets the minimum and maximum airflow desired from the recommended primary air cfm range table provided in the Performance Data section of the catalog. Selecting terminals near the top of their range may reduce cost, but will increase velocity and noise. For typical low pressure applications – selecting towards the bottom of the airflow range will reduce sound levels as larger inlets reduce face velocity and are quieter. Selecting the maximum airflow setting at between 70 – 85% of full capacity (approx. 2000 fpm inlet velocity) is a good trade-off to avoid possible low velocity control problems and sound problems at higher velocities.

2. Fan Size Selection

Fan selection is dictated by model, type and maximum primary airflow.

Parallel (intermittent) fan size is determined by calculating the difference between the unit design heating airflow and minimum primary airflow. If minimum airflow is zero, then fan cfm is the heating airflow. In most cases the fan can be downsized compared to a series terminal, reducing both first cost and operating cost because the fan only requires the capacity to handle the secondary airflow at reduced downstream static pressure compared to the maximum design airflow. In many applications of a parallel terminal, a minimum primary cfm is required to meet ventilation requirements. This primary airflow contributes to the total resistance experienced by the fan and should be accounted for in all components downstream of the fan (ductwork and diffusers). Hot water coils are positioned out of the primary airflow and are not affected by the additional primary airflow. The static pressure resistance felt by the fan due to a hot water coil is based upon fan airflow only, not necessarily total heating airflow.

Series (constant) fan terminals require the fan to be sized to handle the maximum design cfm. The secondary fan cfm must be at least equal to the primary air to ensure the terminal does not become pressurized resulting in primary air spilling out into the ceiling plenum through the induction ports. The external static pressure requirements are the sum of the ductwork and diffusers downstream at design airflow plus an applicable hot water coil or electric heater, if required.

When fan airflow and downstream static pressure have been determined, select the fan size from the fan curves in the Performance Data section of the catalog. Selecting towards the upper end of the range will keep down first cost and optimize fan operating efficiency. Upsizing the fan and operating it at a reduced speed can result in quieter operation. When electric or hot water coils are required, the fan curves show unit performance with the coils in place. Be sure to use the proper fan curve.

3. Heating Coil Selection

First determine the heating supply air temperature to the space by calculation using the heat transfer equation:

$$Q = 1.085 \times cfm \times \Delta t$$

Where: Q = Design heat loss (Btu)

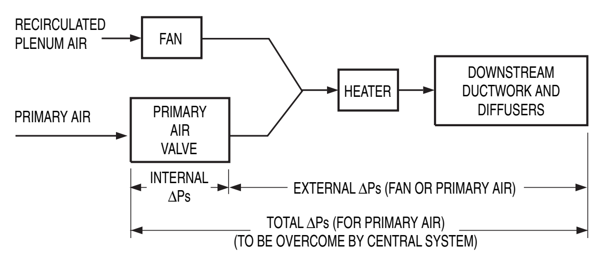
Δt = Supply air temperature (SAT) – Room design temperature.

The supply air temperature (SAT) to the space equals the leaving air temperature (LAT) for the terminal unit.

Once the terminal LAT is determined, the heating requirements for the coil can be calculated. The leaving air temperature for the coil varies based on the type of model.

It is generally a good idea to maintain air temperatures of 85 – 95°F (29 – 35°C) for air entering a room. This is LAT off the heating coil. Air this temperature can be effectively used to warm the room as it is not so buoyant that it cannot be driven to the floor, and it is warm enough to not produce chills from drafts.

Once both coil EAT (entering air temperature) and LAT are calculated, the heat transfer (Q) for the coil must be calculated, using the heat transfer equation. For electric heat, the capacity must be converted from Btu/h to kWh for selection. The required kW and number of steps desired should be checked with availability from the charts in the Performance Data section of the catalog. For hot water coils, reference the capacity charts in the Performance Data to select the appropriate coil.



SCHEMATIC DIAGRAM OF PRESSURE LOSSES AND AIRFLOW FOR A PARALLEL TYPE VARIABLE AIR VOLUME TERMINAL UNIT WITH HOT WATER COIL.

Figure 32. Parallel Terminal with Hot Water Coil: Heating coils are located on the leaving air side of the fan after mixing with the primary air. In this case, coil EAT must be calculated using a mixing equation if minimum primary air is other than zero as it will be blended with plenum air.

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In figures 32, 33 and 34, heating coils are located on the unit discharge so LAT for the coil equals the LAT for the terminal unit. Heating coil EAT equals the temperature of blended primary air and plenum air.

$$\therefore \text{EAT (of coil)} = \frac{T_1Q_1 + T_2Q_2}{Q_T}$$

Where:

T_1 = Plenum air temperature

T_2 = Primary air temperature

Q_1 = Plenum air quantity (cfm)

Q_2 = Primary air quantity (cfm)

Q_T = Total air moved by terminal fan (cfm)

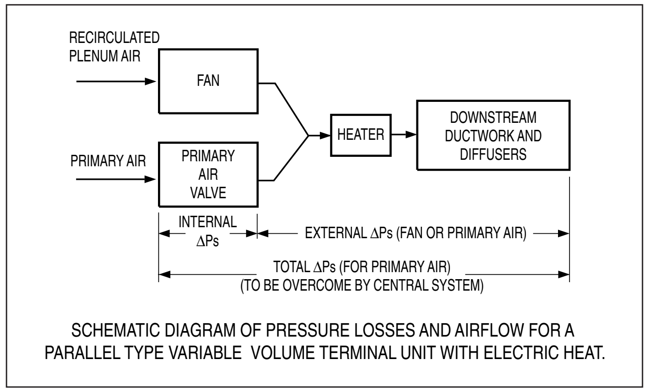

Figure 33. Parallel Terminal with Electric Heat.

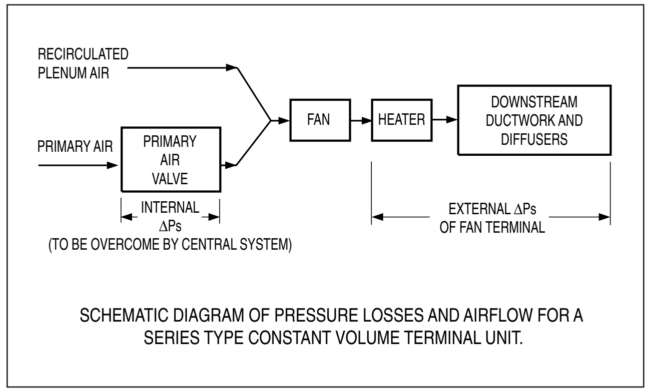

Figure 34. Series Terminals with Hot Water or Electric Heat.

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4. Acoustics

Resulting sound levels are due to air valve generated noise and fan generated noise. The maximum noise generated by a given air valve size is determined by the difference between design inlet static pressure (the valve's most pressurized condition) and external static pressure at design cooling airflow. This represents the most extreme operating condition.

To determine fan noise levels, fan airflow (adjusted within its range by the speed controller) and external static pressure conditions are required.

The acoustical performance data is presented in two formats for the parallel and series type as their sequence of operation differs. With a parallel unit, air valve and fan operation are evaluated separately as their operations are not simultaneous under most conditions. With a series unit, air valve and fan are evaluated together for cooling, as they operate simultaneously and fan only for heating, in the occupied mode (in the unoccupied mode, a night setback fan cycling option is available).

From the performance data, determine the sound power levels and NC predictions for both discharge and radiated path under the appropriate operating conditions. If the terminal is properly located some distance from the supply air space, discharge air noise is generally a secondary concern. Radiated noise from the unit casing typically dictates the noise level when the terminal is installed above the occupied space.

Care should be taken as published NC levels are based upon certain path attenuation assumptions which may not be indicative of a specific design. The size of the appropriate portions of the terminal may be increased to reduce noise, but it is also preferable to plot NC reductions on an NC curve chart to ensure the necessary attenuation reductions are achieved and finished levels do not exceed the NC design goal in the occupied space. To do this properly, the engineer must specify all reductions in the building specifications that will apply.

Example: Parallel Terminal with Hot Water Heat

Select a **Model 35NW** for a maximum/minimum primary airflow at 1000/250 cfm with 1" w.g. inlet static pressure. The heating airflow required is 600 cfm. Downstream resistance at 1000 cfm is 0.4" w.g.. Zone design heat loss is 20,000 Btu, design room temperature 72°F, plenum air temperature 75°F, primary air temperature 55°F.

Air Valve Selection: Choose a size 10 inlet with a minimum wide open static pressure drop of 0.05" w.g..

Fan Selection:

Fan heating airflow = Heating airflow (600 cfm) – primary airflow (250 cfm) = 350 cfm.

The downstream static pressure the fan must overcome is the fan airflow plus primary airflow (600 cfm) and since this is less than maximum design airflow (1000 cfm); fan downstream S.P. = $(600/1000)^2$ x 0.4 = 0.144" w.g..

From the fan curves a size 2 unit will handle 350 cfm at 0.144" S.P. with correct setting of the speed controller and allows for the selection of a one or two row hot water coil.

Heating Coil Selection:

For heating, the temperature difference (Δt) is the zone supply air temperature (SAT) minus the design set point temperature.

20,000 Btu = 1.085 x 600 x (SAT – 72)

(using the heat transfer equation)

∴ SAT = 103°F

As the heating coil is on the unit discharge, the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of plenum and minimum primary air.

Design Heating Flow x Coil EAT = (Primary Airflow x Primary Air Temp.) + [(Design Heating Airflow – Primary Airflow) x Plenum Temperature]

600 x Coil EAT = 250 x 55 + (600 - 250) x 75

∴ Coil EAT = 67°F

For the heating coil, the temperature difference is the coil LAT minus the coil EAT.

Coil heat pick up (Q) = 1.085 x Design cfm x (Coil LAT - Coil EAT)

Coil Q = 1.085 x 600 x (103 – 67) = 21,600 Btu = 21.6 Mbh

From the hot water coil data, unit size 2, selection of a 2 row coil at 600 cfm will provide 21.6 Mbh at about 0.8 GPM (based upon a Δt of 110°F between entering air and entering water).

Note: While there is air side pressure drop data in the catalog, it is not necessary to calculate it. The coil pressure drop is included in the fan curves marked as maximum with water coil.

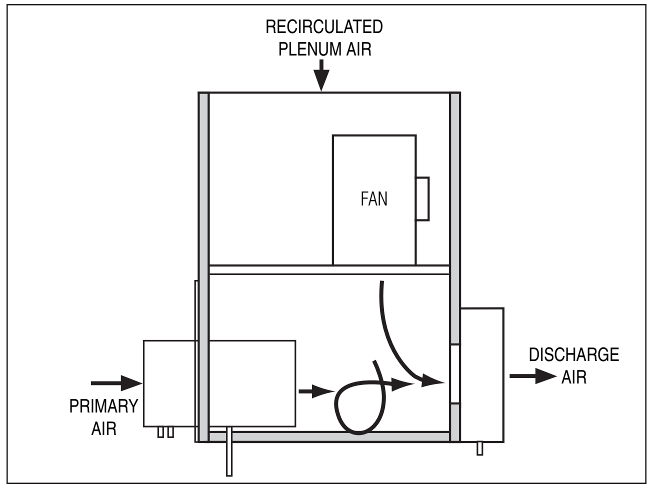

Figure 35. Parallel Fan Terminal With Hot Water Heat in Heating Mode. Model 35NW.

Acoustics: The selection is a 35NW-2-10 with a 2 row hot water coil. At 1" w.g. design inlet static pressure, the closest tabulated sound data @ 1100 cfm cooling and 600 cfm (400 cfm from fan) heating is:

Octave Band		2	3	4	5	6	7
Design max. Cooling (1100 cfm)	Disch.	63	64	61	53	54	55
	Rad.	61	56	52	40	40	40
Design Heating (400 cfm)	Disch.	54	44	47	42	38	33
	Rad.	50	49	49	42	34	30

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Example: Series Terminal with Electric Heat

Select a **Model 35SE** to supply a constant 1500 cfm with 0.5" w.g. inlet static pressure. Minimum primary airflow is 375 cfm and downstream resistance due to ductwork and diffusers is 0.4" w.g.. Zone design heat loss is 45,000 Btu, design room temperature 72°F, plenum air temperature is 75°F, primary air temperature 55°F.

Air Valve Section: Choose a size 12 inlet with a minimum wide open pressure drop of 0.05" w.g.. The damper will throttle to maintain desired airflow.

Fan Selection:

Fan airflow equals design airflow with a series unit. Fan external static pressure equals downstream static pressure (ductwork and diffusers). The resistance of electric and hot water heating coils and their associated additional pressure drop is taken into account on the fan curves. From the fan curves, a size 5 unit will handle 1500 cfm at 0.4" and falls nicely in the middle of the fan range which can be adjusted with the speed controller.

Heating Coil Selection:

For heating, the temperature difference (Δt) is the zone supply air temperature (SAT) minus the design set point temperature.

45,000 Btu = 1.085 x 1500 x (SAT – 72)

∴ SAT = 100°F

As the heating coil is on the unit discharge, the unit supply temperature equals the coil LAT. Coil entering air temperature (EAT) is a mixture of plenum and minimum primary air.

Design Heating Flow x Coil EAT = (Primary Airflow x Primary Air Temp.) + [(Design Heating Airflow – Primary Airflow) x Plenum Temperature]

1500 x Coil EAT = 375 x 55 + (1500 - 375) x 75

∴ Coil EAT = 70°F

For the heating coil, the temperature difference is the coil LAT minus the coil EAT.

Coil heat pick up (Q) = 1.085 x Design cfm x (Coil LAT - Coil EAT)

Coil Q = 1.085 x 1500 x (100 – 70) = 48,825 Btu = 48.8 Mbh

Convert to kWh: 48.8 ÷ 3.413 = 14.3 kWh

From the electric heat selection data in the Performance Data section of the catalog, a size 5 unit requires a 208, 240 or 480 volt/3-phase electric heat coil and would be available with up to 3 stages with pneumatic or digital control or 2 stages with analog electronic control.

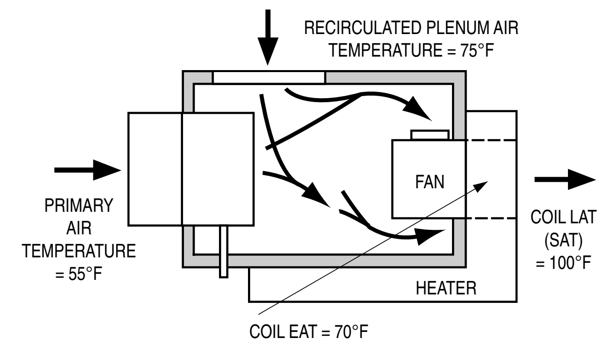

Figure 36. Series Fan Terminal With Electric Heat in Heating Mode. Model 35SE.

Acoustics: The selection is a Model 35SE-5-12. At 0.5" w.g. design inlet static pressure, the closest tabulated sound data is @ 1600 cfm is:

Octave Band	2	3	4	5	6	7
Discharge	69	69	69	68	65	65
Radiated	74	67	63	56	58	57

Terminal Installation and Application Precautions – Avoiding Common Errors and Problems

Sizing Terminals

Select terminals based upon recommended air volume ranges. The pressure independent terminals main feature is its ability to accept factory calibrated minimum and maximum airflow limits that correspond to the designers space load and ventilation requirements for a given zone.

A common misconception is that oversizing a terminal will make the unit operation quieter. In reality, the terminal damper will have to operate in a pinched-down condition most of the time which may actually increase noise levels to the space. Control accuracy may suffer as the terminal is only using a fraction of its total damper travel or stroke. In addition, the low inlet velocities may be insufficient to produce a readable signal for the sensor and reset controller. This means minimum settings may not hold with a resultant loss of control accuracy and undesirable hunting.

The recommended selection for maximizing performance is to size the terminals maximum airflow limit for 70 – 85% of its rated capacity (approx. 2000 fpm) in accordance with the catalog recommendations. For accurate control the minimum setting guideline is not lower than 20% of the units rated total capacity.

Another problem associated with oversizing terminals with electric heat is again insufficient velocity causing occasional tripping of the airflow safety switch.

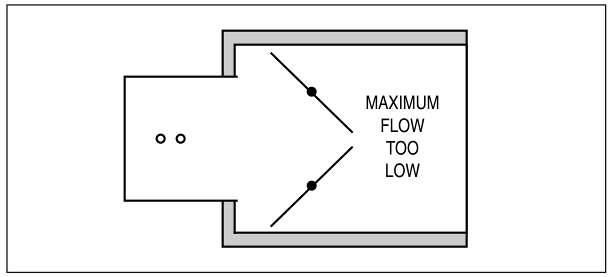

Figure 37. Severe Throttling: Oversized terminals will operate in a near closed position even at maximum airflow. Control accuracy may also suffer.

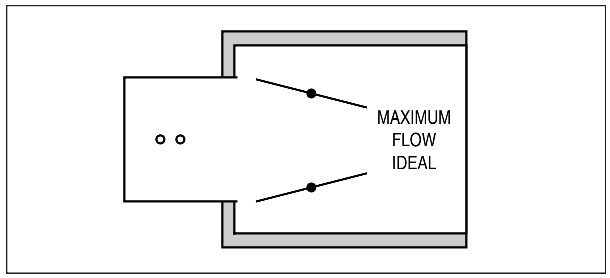

Figure 38. Ideal Throttling: Correctly sized terminal will utilize the majority of its damper travel and improve performance.

Observe Space Restrictions

During the design phase try and ensure terminals are located for ease of installation, optimum performance and maintenance accessibility. Figure 39 shows all of the worst conditions: a convoluted inlet, controls and heating coil connections are restricted as the terminal is against a wall and the outlet restricted condition reduces performance.

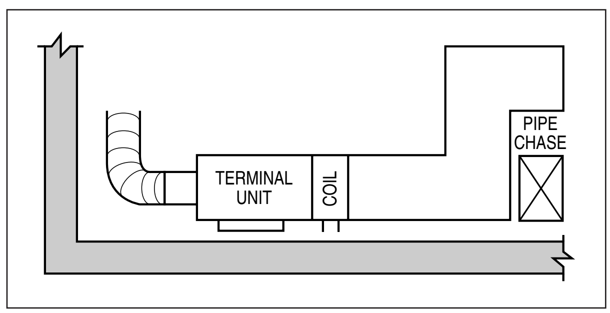

Figure 39. Restricted Installation, Poor Location.

Optimize Inlet Conditions

The type of duct and its approach may have a large and adverse impact on both pressure drop and control accuracy. Figure 40 shows several typical poor conditions that generate unwanted turbulence. Although multi-point sensors can compensate to a large degree, good design practice should always prevail. Nailor recommends wherever possible, a straight duct inlet connection with a minimum length of two duct diameters, the same size as the inlet.

Terminal collars are undersized to suit nominal ductwork dimensions. The inlet duct slips over the terminal inlet collar and is fastened and sealed in accordance with job specifications. Never insert a duct inside the inlet collar, control calibration will be adversely affected.

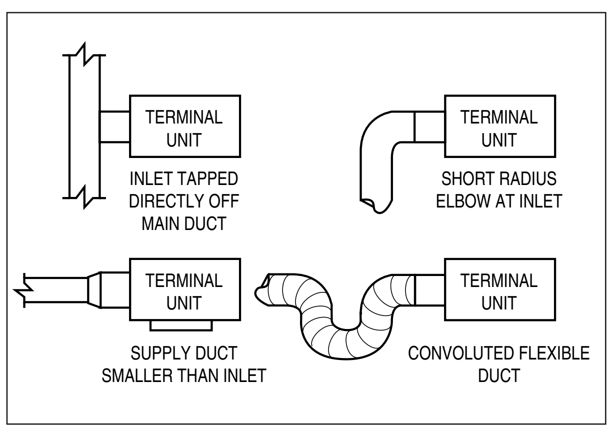

Figure 40. Poor Inlet Conditions.

Sometimes it is not possible due to space restrictions to provide an ideal inlet condition. In this case field adjustment of the airflow settings on the velocity controller may be required to compensate. The use of flow straightening devices (equalizing grids) are recommended after short radius elbows that are immediately ahead of the terminal and where terminals are unavoidably tapped directly off the main duct.

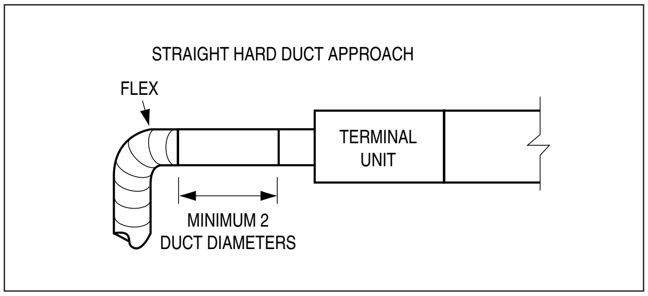

Figure 41. Ideal Inlet Conditions.

Observe Zoning Requirements

Correctly sizing terminals with regard to the physical conditions of the occupied space is vital to ensure acceptable performance. One large terminal serving a space with divided work areas may result in the single thermostat only providing acceptable temperature control where it is located. The other space(s) served may be too cold or too hot if it has differing space load requirements.

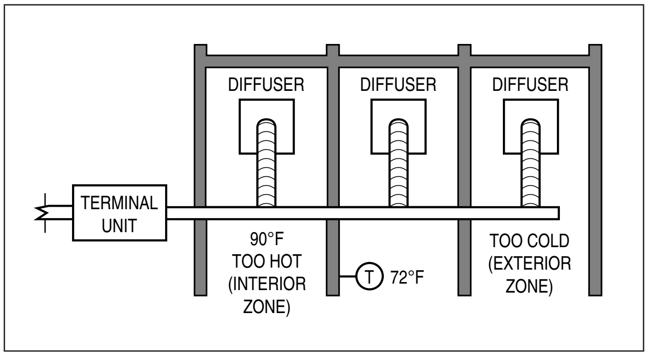

Figure 42. Poor Zone Application.

Optimize Discharge Conditions

Poor discharge duct connections may have an adverse affect on pressure drop. Try and avoid installing tees, transitions and elbows close to the inlet discharge. Avoid long runs of flex and keep short flex runs as straight as possible. Make curves as shallow as possible and ensure entrance condition to diffuser outlet is straight.

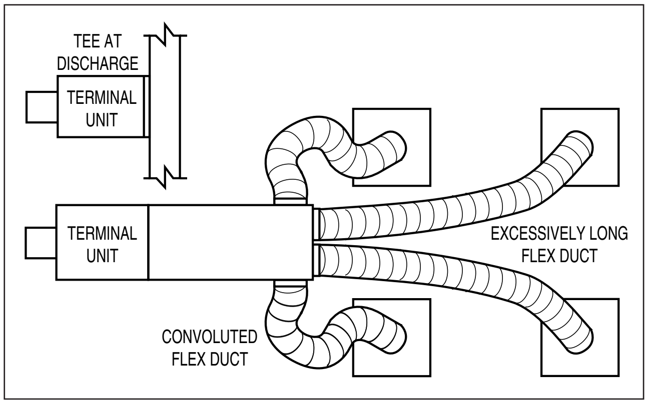

Figure 43. Poor Discharge Conditions.

Non-Compliance with Local Electrical Codes

Some local jurisdictions have more exacting codes than the minimum requirements of national codes such as NEC, UL and CSA. One example is the primary fusing required of the power circuit in some areas.

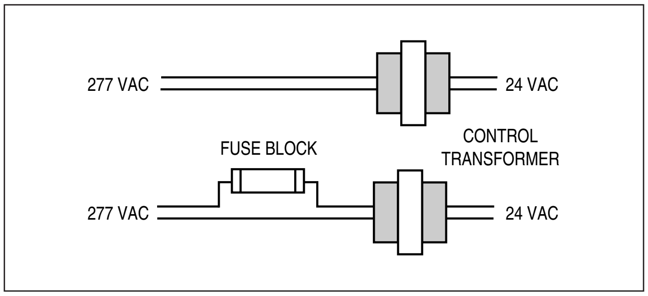

Figure 44. Power circuit fusing is required by some local code authorities.

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Power Source Compatibility

Terminals with an electrical power supply such as fan powered terminals and single duct terminals with electric heat should be checked for compatibility with source. Voltage, phase and frequency must match. Where motor voltage differs, the single phase voltage requirement must be tappable from a three phase (4 wire wye) power source.

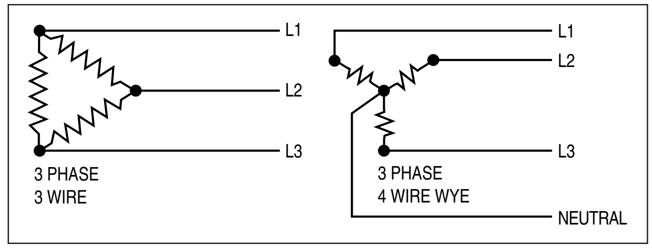


Figure 45. The requirement for three phase electricity must be specified as 3 wire or 4 wire wye.

Avoid Excessive Air Temperature Rise

Terminals with electric or hot water reheat coils should be designed to satisfy load conditions but attention should be paid to the temperature differential (Δt) between the entering room air and ambient temperature. The ASHRAE handbook recommends a maximum Δt of 15°F (8°C) to avoid possible stratification due to the excessive buoyancy of the warm air and ensure good room mixing and temperature equalization. Absolute maximum air entry temperature is 120°F (49°C) for comfort heating.

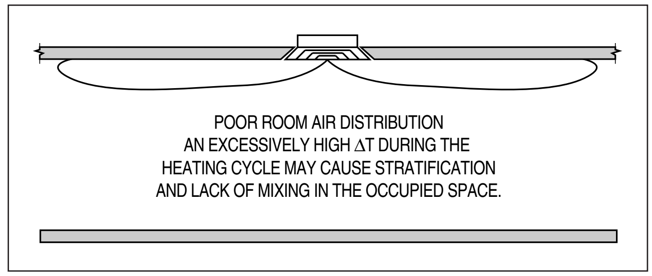


Figure 46. Avoid excessive temperature differentials

Correctly Supported Terminal

Although the basic single duct terminal is light enough that it can be supported by the ductwork in which it is installed, we recommend that the units be independently supported. When accessory modules such as heating coils, attenuators or multiple-outlet plenums are included, the assembly should also be supported directly.

Larger terminals such as fan powered terminal units should always be independently supported and secured to building structure.

Be careful not to block access panels with straps, all-thread rods or trapeze supports.

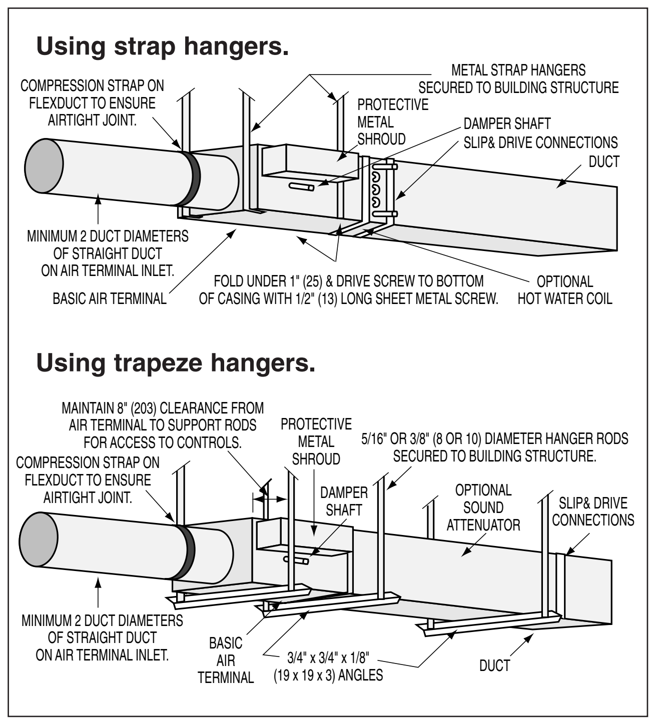


Figure 47. Recommended Terminal Suspension.

Minimize Duct Leakage

To prevent excess air leakage and minimize energy waste, all joints should be sealed with an UL approved duct sealer. Most leakage can be avoided by practicing good fabrication and installation techniques, particularly upstream of the terminal which may invariably be required to hold significantly higher pressures than downstream of the terminal.

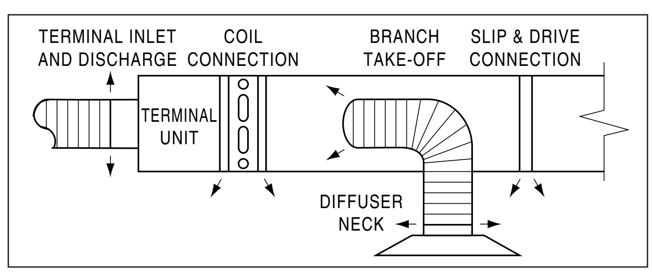


Figure 48. Possible Air Leakage Points.

Estimating Sound Levels

1. Noise Criteria - NC

NC level or Noise Criteria is a widely accepted and popular way for many engineers to estimate and predict room noise levels. NC levels are also used as a rating scale for equipment that is expected to operate within certain levels.

Sound Power levels for VAV terminal units are expressed over six octave bands: 2, 3, 4, 5, 6 and 7. Each octave band is defined by the center frequency within that particular band. Frequency is measured in Hz. Each of these respective frequencies and octave bands are shown in Table 1.

Octave Band	2	3	4	5	6	7
Center Frequency	125	250	500	1000	2000	4000

Table 1: Octave band designation

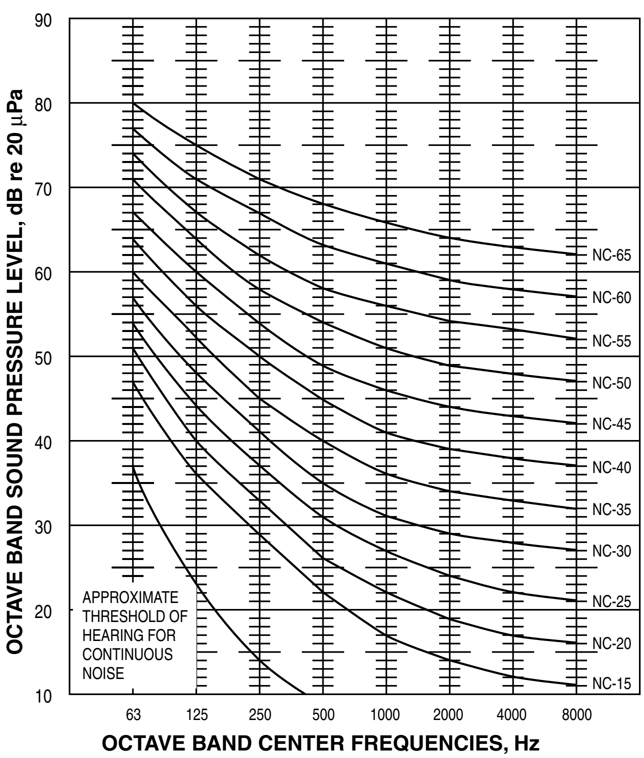


Figure 1: NC (Noise Criteria) Curves for Specifying the Design Level in Terms of the Maximum Permissible Sound Pressure Level for Each Frequency Band

When deriving NC levels for terminal units, the pressure levels are plotted on a standard NC Chart (Figure 1). The highest pressure level when measured against the NC curves, regardless of the octave band, determines the NC of the unit. Table 2 shows the ASHRAE recommended RC and NC levels for several different types of space. RC (Room Criteria) Levels are a newer method of evaluating sound performance of an HVAC system as a whole in order to achieve a well balanced, bland-sounding spectrum. While RC ratings may be an excellent tool for evaluating all sound in a space, they are not practical as a means of rating air terminals. For a full explanation, consult the ASHRAE Applications Handbook Chapter on Sound and Vibration Control and ARI 885.

Room Type	RC (N) or NC Level[a,b]
Residences, Apartments, Condominiums	25 – 35
Hotels/Motels	
Individual rooms or suites	25 – 35
Meeting/banquet rooms	25 – 35
Corridors, lobbies	35 – 45
Service/support areas	35 – 45
Offices Buildings	
Executive and private offices	25 – 35
Conference rooms	25 – 35
Teleconference rooms	25 (max)
Open-plan offices	30 – 40
Corridors and lobbies	30 – 45
Hospitals and Clinics	
Private rooms	25 – 35
Wards	30 – 40
Operating rooms	25 – 35
Corridors and public areas	30 – 40
Performing Arts Spaces	
Drama theaters	25 (max)
Concert and recital halls[c]	
Music teaching studios	25 (max)
Music practice rooms	35 (max)
Laboratories (with fume hoods)	
Testing/research, minimal speech communication	45 – 55
Research, extensive telephone use	40 – 50
speech communication	
Group teaching	35 – 45
Church, Mosque, Synagogue	
General assembly	25 – 35
With critical music programs[c]	
School[d]	25 – 35
Classrooms up to 750 ft^2	40 (max)
Classrooms over 750 ft^2	35 (max)
Large lecture rooms w/out speech amplification	35 (max)
Libraries	30 – 40
Courtrooms	
Unamplified speech	25 – 35
Amplified speech	30 – 40
Indoor Stadiums, Gymnasiums	
Gymnasiums and natatoriums[e]	40 – 50
Large seating-capacity spaces	45 – 55
with speech amplification[e]	

Table 2: Design Guidelines for HVAC-Related Background Sound in Rooms

[a]The values and ranges are based on judgment and experience, not on quantitative evaluations of human reactions. They represent general limits of acceptability for typical building occupancies. Higher or lower values may be appropriate and should be based on a careful analysis of economics, space use and user needs.

[b]When quality of sound in the space is important, specify criteria in terms of RC(N). If the quality of the sound in the space is of secondary concern, the criteria may be specified in terms of NC or NCB levels of similar magnitude.

[c]An experienced acoustical consultant should be retained for guidance on acoustically critical spaces (below RC 30) and for all performing arts spaces.

[d]HVAC-related sound criteria for schools, such as those listed in this table, may be too high and impede learning by children in primary grades whose vocabulary is limited. Some educators and others believe that the HVAC-related background sound should not exceed RC 25 (N).

[e]RC or NC criteria for these spaces need only be selected for the desired speech and hearing conditions.

Most manufacturers list raw sound power levels for a wide range of operating conditions for their equipment. To predict sound pressure levels within a space, the Air Conditioning and Refrigeration Institute (ARI) has developed ARI Standard 885-98. "Procedure for Estimating Occupied Space Sound Levels in the Application of Air Terminals and Air Outlets." It has been widely accepted. The standard displays several paths that sound could take into the space. Each of these paths must be evaluated. The attenuation in each octave band is calculated and subtracted from the manufacturers sound power levels.

The ARI 885 Standard forms the basis for the sound estimation guidelines and examples presented on the following pages. For a more detailed analysis, refer to ARI Standard 885-98, the 1997 ASHRAE Fundamentals Handbook, chapter 7 and the 1999 ASHRAE HVAC Applications Handbook, chapter 46.

2. Environmental Adjustment Factor

According to ARI Standard 885-1998, an environmental adjustment factor must be applied to manufacturer's data if the sound power data has been obtained in accordance with ARI Standard 880.

Sound power levels obtained in accordance with Standard 880 are based on a free field calibration of the reference sound source in accordance with ANSI S12.31-90. "Precision Methods for the Determination of Sound Power Levels of Broad-Band Noise Sources in Reverberation Rooms." Real rooms at low frequencies behave acoustically more like reverberant rooms than open spaces (free field). Therefore it is necessary to adjust power levels obtained in accordance with ARI Standard 880 by the Environmental Adjustment Factor listed in Table 3. These factors are subtracted from the manufacturer's sound power level data. Nailor tests all terminal units in accordance with ANSI/ASHRAE Std. 130 and ARI Standard 880; therefore these corrections should be applied when estimating the sound power in occupied spaces.

Octave Band	2	3	4	5	6	7
dB Reduction	2	1	0	0	0	0

Table 3: Environmental Adjustment Factor
Ref: ARI Std 885-98, Appendix C, Table C1, page 35.

3. Sound Paths

In order to estimate the sound power level in the occupied space, one must first identify the sound source and then determine by which paths the sound enters the occupied space. The example in Figure 2 illustrates a fan powered terminal as the sound source and identifies all the sound paths. These are:

1. **Upstream Duct Breakout Radiated**
   This is the sound generated by the terminal unit which is transmitted through the upstream ductwork.
2. **Casing and Induction Inlet Radiated**
   This is the sound transmitted through the terminal unit casing and the induction air inlet on a fan powered terminal.
3. **Discharge Duct Breakout Radiated**
   This is the sound generated by the terminal unit which is transmitted through the downstream ductwork walls. This occurs at several locations.
4. **Outlet Discharge**
   This is the sound generated by the terminal which travels down the ductwork and escapes at the air outlet.
5. **Outlet Generated**
   This is the sound generated by the air outlet (grille, diffuser) itself.

While a discerning engineer should evaluate each of these paths when designing a building, the discussion and examples that follow will only consider the radiated sound from the terminal and the discharge sound emitted into the room at the diffuser. These are a function of terminal performance and usually the most significant and therefore critical sound paths requiring analysis.

Upstream and discharge duct breakout radiated noise paths are not usually a contributing factor to the occupied sound level so long as care is taken in the design and installation of the ductwork. However, a detailed analysis of these paths is covered in ARI-885.

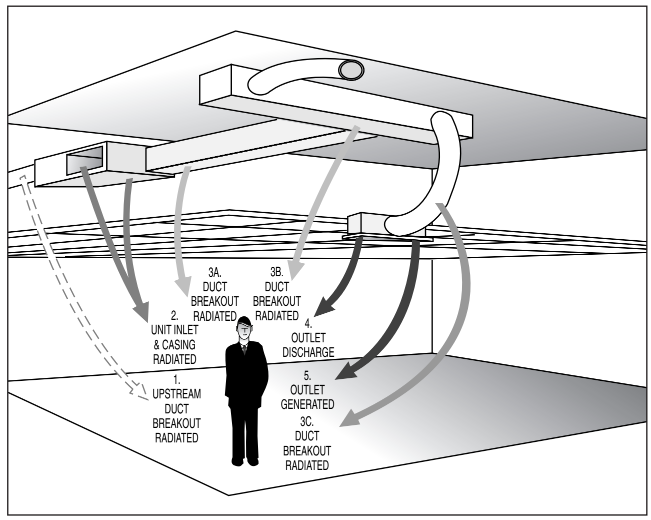


Figure 2: Typical Sound Paths for a Terminal Unit (Fan Powered Type illustrated)

4. Radiated Sound

Fig. 3 illustrates the sound path for casing and inlet radiated sound. The attenuation factors that apply to this sound path are Ceiling/Space Effect and Environmental Adjustment Factor (which was presented earlier).

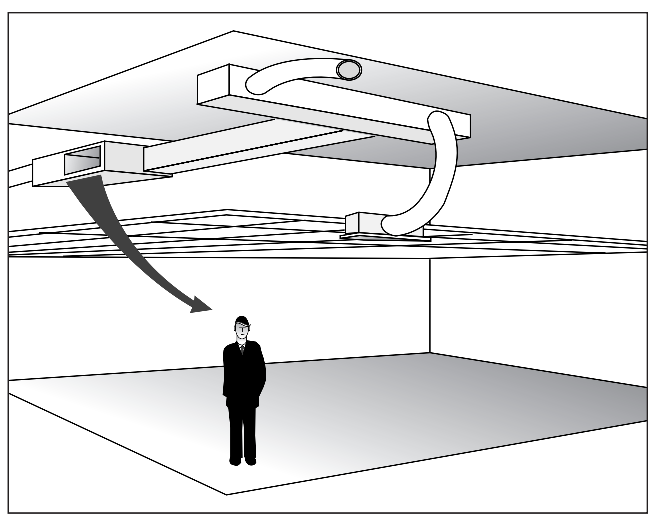


Figure 3: Radiated Sound Path

Ceiling/Space Effect

To calculate the sound level in a space resulting from a sound source located in the ceiling cavity, a transfer function is provided which is used to calculate the sound pressure in the space. This transfer function includes the combined effect of the absorption of the ceiling tile, plenum absorption and room absorption. This procedure is based on research conducted under ASHRAE research project RP-755, approved June 1997. The procedure assumes the following conditions:

a. The plenum is at least 3 feet (0.9 meters) deep.

b. The Plenum space is either wide [over 30 feet (9 meters)] or lined with insulation.

c. The ceiling has no significant penetrations directly under the units.

Table 4 provides typical values for ceiling space effect of several ceiling types. For conditions other than these, sound transfer functions may be less. For instance, in a shallow plenum, 2 ft. (0.6 m) or less, tests have shown that sound in the space can be expected to be 5 – 7 dB louder below 500 Hz (4th Octave Band).

Octave Band	2	3	4	5	6	7
Mineral Fiber Tile 5/8", 20lb/ft^3	16	18	20	26	31	36
Glass Fiber Tile-2", 4 lb/ft^3	17	18	21	25	29	35
Solid Gypsum Board – 5/8", 43 lb/ft^3	23	27	27	29	29	30

Table 4: Ceiling/Space Effect Attenuation Values
Ref: ARI Std 885-98, Appendix D, Table 14, Page 48.

5. Discharge Sound

Figure 4 illustrates the sound path for outlet discharge sound. The attenuation factors which apply to this sound path are:

- Environmental Adjustment Factor
- Lined Duct Insertion Loss
- Elbow and Tee Loss
- Branch Power Division
- Lined Flexible Duct Insertion Loss
- End Reflecton Factor
- Space Effect

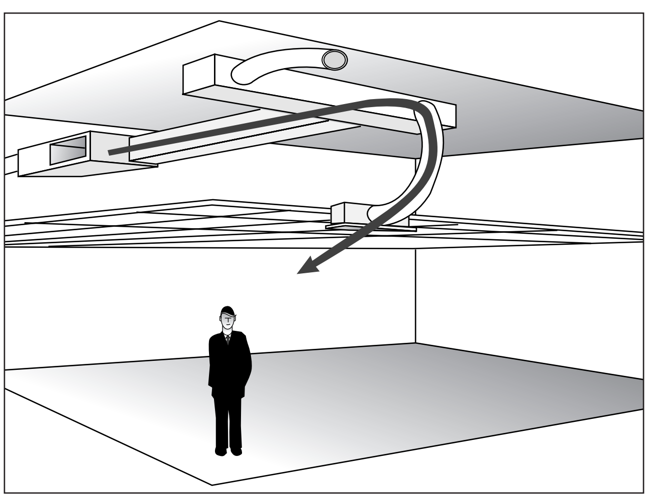


Figure 4: Discharge Sound

Lined Duct Insertion Loss

As sound travels down a duct, some of its energy is absorbed by the duct and its lining. Some of the energy is also radiated or transmitted through the duct walls. Consequently, the sound pressure level at the discharge end of the duct will be lower than at the inlet of the duct.

Duct Insertion Loss is affected by the size of the duct at the discharge of the terminal unit. Table 5 shows several different sizes of discharge ducts that are commonly used for Nailor VAV terminal units and their associated attenuation factors for each octave band in dB/linear foot when internally lined with 1" (25) thick, 1 1/2 lb/ft^3 density insulation.

Nominal Duct Dimension		Octave Band					
inches	mm	2	3	4	5	6	7
10 x 10	254 x 254	0.4	1.0	2.1	4.4	4.7	3.1
12 x 12 1/2	305 x 318	0.4	0.8	1.9	4.0	4.1	2.8
14 x 12 1/2	356 x 318	0.3	0.8	1.8	3.9	3.7	2.6
18 x 12 1/2	458 x 318	0.3	0.7	1.7	3.7	3.5	2.5
24 x 12 1/2	610 x 318	0.3	0.6	1.6	3.3	2.9	2.2
28 x 12 1/2	711 x 318	0.3	0.6	1.7	3.5	3.0	2.3
38 x 12 1/2	966 x 445	0.2	0.4	1.3	2.7	2.0	1.7
14 x 14	356 x 356	0.3	0.8	1.8	3.7	3.7	2.6
15 x 15	381 x 381	0.3	0.7	1.7	3.6	3.3	2.4
16 x 16	406 x 406	0.3	0.7	1.7	3.5	3.3	2.4
28 x 18	711 x 457	0.2	0.5	1.4	3.0	2.4	1.9
18 x 16	457 x 406	0.3	0.6	1.7	3.5	3.2	2.3
22 x 16	558 x 406	0.3	0.6	1.6	3.3	2.9	2.2
16 x 12	406 x 305	0.3	0.8	1.8	3.8	3.7	2.6
24 x 15	610 x 318	0.3	0.6	1.6	3.3	2.9	2.2
50 x 15	1270 x 381	0.2	0.5	1.4	2.9	2.4	1.9
40 x 12	1016 x 305	0.3	0.6	1.6	3.3	2.9	2.2
12 x 10	305 x 254	0.4	0.9	2.0	4.2	4.4	2.9
14 x 10	356 x 254	0.4	0.8	1.9	4.1	4.2	2.8
28 x 10	711 x 254	0.3	0.7	1.7	3.6	3.3	2.4
40 x 9	1016 x 229	0.3	0.6	1.6	3.3	2.9	2.2
24 x 8	610 x 203	0.4	0.8	1.9	4.0	4.1	2.8

Table 5: Sound Insertion Loss/Attenuation in Straight Lined Metal Ducts, dB/ft.
Ref: ARI Standard 885-98, Appendix D, Table 8, page 42. 1999 ASHRAE handbrook, HVAC Applications , Ch. 46, Table 8.

Elbow and Rectangular Tee Loss

Lined and unlined rectangular elbows provide attenuation as per Tables 6a & 6b. Tee fittings can be considered as if they are two elbows side by side, where mean duct width for estimation purposes from Tables 6a & 6b is taken as being 1/2 of the actual duct width. See Figure 5.

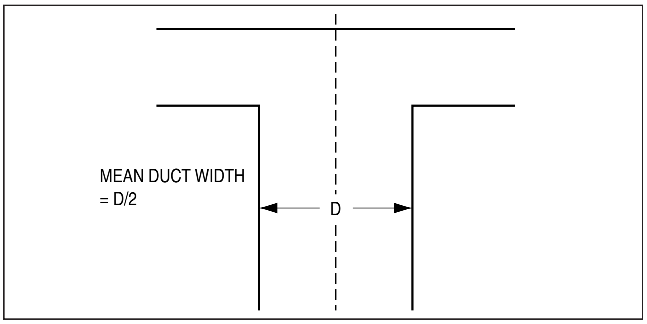


Figure 5: Tee Fitting Loss.

	Duct Width		Octave Band					
	Inches	mm	2	3	4	5	6	7
Unlined Duct	5-10	100-125	0	0	1	5	8	4
	11-20	260-700	1	5	5	8	4	3
	21-40	710-1000	5	5	8	4	3	3
	41-80	1010-2000	5	8	4	3	3	3
Lined Duct	5-10	100-125	0	0	1	6	11	10
	11-20	260-700	1	6	6	11	10	10
	21-40	710-1000	6	6	11	10	10	10
	41-80	1010-2000	6	11	10	10	10	10

Table 6a: Insertion Loss of Unlined and Lined Elbows without Turning Vanes, db

	Duct Width		Octave Band					
	Inches	mm	2	3	4	5	6	7
Unlined Duct	5-10	100-125	0	0	1	4	6	4
	11-20	260-700	1	4	6	4	4	4
	21-40	710-1000	4	6	6	4	4	4
	41-80	1010-2000	4	6	6	4	4	4
Lined Duct	5-10	100-125	0	0	1	4	7	7
	11-20	260-700	1	4	7	7	7	7
	21-40	710-1000	4	7	7	7	7	7
	41-80	1010-2000	4	7	7	7	7	7

Table 6b: Insertion Loss of Unlined and Lined Elbows with Turning Vanes, db

Ref: ARI Std 885-98. Appendix D, Table D12, page 46. ASHRAE Applications Handbook, 1999, Chapter 46, Table 13 & 15.

Branch Power (Flow) Division

This calculation should be performed for each junction, where a division of airflow exists, such as tees and branch takeoffs. At Branch takeoffs, acoustic energy is distributed between the branches and/or the main duct in accordance with the ratio of the branch cross sectional area to the total cross sectional area of all ducts leaving the takeoff. Acoustic energy is divided in proportion to the flow division. Table 7 lists the attenuation at various percentages of total flow carried by the branch ductwork.

% of Total Flow	60	50	40	30	20	15	10	5
dB Attenuation	1	3	4	5	7	8	10	13

Table 7: Power Level Division at Branch Takeoffs

Ref: ARI Standard 885-98, Appendix D, Table D2, page 37.

Lined Flexible Duct Insertion Loss

Insertion loss values for lined flexible duct are listed in Table 8. Unlined flexible duct should be conservatively modeled as unlined hard duct, which ARI-885 regards as negligible.

Non-metallic insulated flexible ducts can significantly reduce airborne noise. Recommended duct lengths are normally from 3 to 6 ft. (0.9 to 1.8 m). Care should be taken to keep flexible ducts straight; bends should have as long a radius as possible. While an abrupt bend may provide some additional insertion loss, the airflow generated noise associated with the airflow in the bend may be unacceptably high. Because of potentially high duct breakout sound levels associated with flexible ducts, care should be exercised when using flexible ducts, above sound sensitive spaces, The data in Table 8 is based on solid core (non-perforated or woven), 1" (25) thick insulation and plastic jacket.

Duct Dia. inches (mm)	Duct Length ft. (m)	Octave Band 2	3	4	5	6	7
4 (100)	3 (3.0)	5	4	12	20	23	15
	5 (1.5)	7	5	16	24	27	18
	10 (3.0)	10	9	27	33	38	24
5 (125)	3 (0.9)	5	5	13	19	21	13
	5 (1.5)	6	7	17	23	25	16
	10 (3.0)	10	12	28	33	37	23
6 (150)	3 (0.9)	5	6	13	17	19	11
	5 (1.5)	6	9	18	22	24	15
	10 (3.0)	10	15	28	33	35	22
7 (175)	3 (0.9)	5	7	14	16	17	10
	5 (1.5)	6	10	18	21	22	13
	10 (3.0)	10	16	29	33	34	21
8 (200)	3 (0.9)	4	7	14	15	16	8
	5 (1.5)	6	10	18	20	21	12
	10 (0.9)	10	18	29	32	32	20
10 (250)	3 (0.9)	4	7	14	12	13	6
	5 (1.5)	5	11	18	18	18	9
	10 (3.0)	9	19	28	31	29	18
12 (300)	3 (0.9)	3	6	12	10	11	4
	5 (1.5)	4	9	16	16	15	7
	10 (3.0)	8	17	26	29	26	15
14 (350)	3 (0.9)	2	4	10	9	9	4
	5 (1.5)	3	7	14	14	13	6
	10 (3.0)	6	13	23	26	23	12
16 (400)	3 (0.9)	1	0	8	8	8	4
	5 (1.5)	2	2	11	12	11	5
	10 (3.0)	4	7	19	24	20	8

Table 8: Lined Flexible Duct Insertion Loss, dB

Ref: ARI Std 885-98, Appendix D, Table D9, page 43.

End Reflection Factor

When plane wave sound passes from a small space such as a duct into a large space the size of a room, a certain amount of sound is reflected back into the duct, significantly reducing low frequency sound. See Table 9. The values of Table 9 apply to straight runs of duct entering a room, therefore caution should be exercised when a condition differs drastically from the test conditions used to derive the table.

Duct Dia. inches (mm)	Octave Band 2	3	4	5	6	7
6 (152)	12	6	3	1	0	0
8 (203)	9	5	2	0	0	0
10 (254)	8	3	1	0	0	0
12 (305)	6	3	1	0	0	0
16 (406)	5	2	0	0	0	0

Table 9: End Reflection Loss, dB. ISO Std 5135.
Ref: ARI Std 885-98. Appendix D. Table D13, page 47

Space Effect

Space effect is the attenuation of sound power entering a space as a result of the absorption properties of the space and the distance from the sound source to the receiver location (recipient). A sound source terminating in the occupied space is assumed to be a point source. The calculation of the sound pressure level L_P in rooms for the entering sound power L_W can be accomplished using the Schultz equation:

$$L_P = L_W - 10 \log r - 5 \log V - 3 \log f + 25$$

Where:

L_P = sound pressure level in dB, re 20 μPa

L_W = sound power level in dB, re 10^{-12} watts

r = shortest distance in ft. from noise source to receiver

V = room volume in ft.^{3}

f = octave band center frequency in Hz.

Since Space Effect = $L_W - L_P$, then,

$$\text{Space Effect} = 10 \log r + 5 \log V + 3 \log f\ (\text{Hz}) - 25$$

Table 10 provides space effect values for several typical conditions that may be used for easy reference. Attenuation values for Space Effect should be used for both the discharge sound traveling from an air terminal through the supply ductwork and entering the room through the diffuser and separately for the air outlet (diffuser) itself.

In order to compare the noise levels of different systems at the design stage where exact room dimensions are not known, the following default room values are suggested.

1. Small Room, Single Outlet 1,500 ft^3 (42 m^3)
2. Large Room, Four Outlets 8,000 ft^3 (220 m^3)

It is also recommended that noise level predictions be made at heights 5 ft. [1.5 m] above the floor when no specific height is specified.

Room Volume	Distance from Source	Octave Band 2	3	4	5	6	7
2000 ft.^{3} (56 m^3)	5 ft. (1.5 m)	5	6	7	7	8	9
	10 ft. (3.0 m)	8	9	10	11	11	12
	15 ft. (4.6 m)	10	10	11	12	13	14
2500 ft.^{3} (69 m^3)	5 ft. (1.5 m)	5	6	7	8	9	10
	10 ft. (3.0 m)	8	9	10	11	12	13
	15 ft. (4.6 m)	10	11	12	13	14	14
3000 ft.^{3} (83 m^3)	5 ft. (1.5 m)	6	7	7	8	9	10
	10 ft. (3.0 m)	9	10	10	11	12	13
	15 ft. (4.6 m)	10	11	12	13	14	15
5000 ft.^{3} (140 m^3)	5 ft. (1.5 m)	7	8	9	9	10	11
	10 ft. (3.0 m)	10	11	12	12	13	14
	15 ft. (4.6 m)	12	12	13	14	15	16

Table 10: Space Effect, Point Source, dB.
Ref: ARI Std 885-98, Appendix D, Table D16, page 49.

6. Outlet Generated Sound

This is the sound generated by the air outlet (diffuser) itself and is considered a point source. The attenuation factor which applies to this sound path is space effect (from the Schultz equation described on page G26). The attenuation allowances in Table 10 may be used for a single sound source in the room.

Due to the large number and diverse range of model sizes and airflow rating points that must be presented, in order to simplify selection and reduce the amount of documented performance data, manufacturers of grilles, registers, diffusers and other air outlet devices publish a single NC sound rating, rather than presenting the individual sound power levels in each octave band. Published NC ratings commonly subtract 10 dB from measured sound power levels in each octave band to account for an average room attenuation (absorption). As discussed earlier, under environmental adjustment factor and space effect (Tables 3 and 10), this will be a valid assumption for a number of combinations of room volume and distance from the source.

A conservative estimate of outlet generated sound power levels can be obtained by assuming the individual octave band sound pressure levels associated with the published NC rating (presented in Table 11), and then adding to these values in each octave band, the manufacturer's assumed (10 dB) room absorption.

For a closer approximation of diffuser sound power when only NC is known, one can assume that the sound power for the diffuser in the 5th octave band (1,000 Hz) is equal to the reported NC plus 10 dB, the 4th band (500 Hz) is 3 greater than this and the 6th band (2000 Hz) is 5 less. The 2nd, 3rd and 7th octave bands do not significantly contribute to the space sound level and can be ignored.

NC	Octave Band 2	3	4	5	6	7
15	36	29	22	17	14	12
20	40	33	26	22	19	17
25	44	37	31	27	24	22
30	48	41	35	31	29	28
35	52	45	40	36	34	33
40	56	50	45	41	39	38
45	60	54	49	46	44	43
50	64	58	54	51	49	48

Table 11: Tabular Representation of NC Curves, dB
Ref: ARI Std 885-98, Table 13, page 27

Multiple Sound Sources

Method A. Logarithmic addition of single sound sources using Schultz Equation for Space Effect.

Manufacturers published NC sound data is for a single source. Allowances must therefore be made for multiple outlets in a single space, since the overall noise level may be higher. Table 12 lists the additive effect of multiple outlets when their sound levels are equal.

No. of Outlets	1	2	3	4	8	10	20
dB addition	0	3	5	6	9	10	13

Table 12: Sound allowance for multiple outlets of equal sound level.

When the sound at each outlet is not equal, they must be added in pairs. Sound power and pressure levels expressed in decibels (dB) are logarithmic functions and are therefore not added directly. Figure 6 provides a simple means of estimating the result.

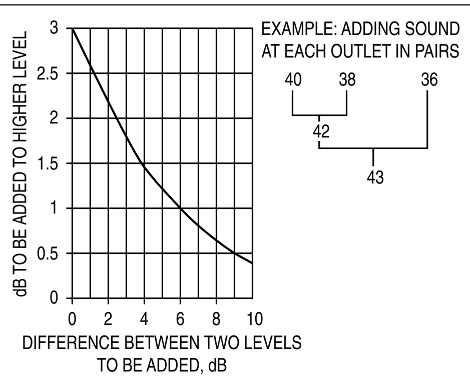


Figure 6: dB Addition for Multiple Outlets.
Ref: ARI Std. 885-98, Figure 4, page 9.

For a large open space with a large number of diffuser outlets, consider an area of 400 to 600 sq. ft. with an aspect ratio no less than 1 to 2, as the maximum area where the number of diffusers present contribute to the overall sound level.

Method B. Distributed Array Space Effect.

The above calculation procedure can be tedious and time consuming for a large number of outlets. The Schultz equation must be used to calculate the sound pressure levels for each individual air outlet at their specific location in the room relative to the receiver location and then logarithmically added.

For the special case of a distributed ceiling array of air outlets where all of the sources have the same L_W, the calculation can be simplified by using the following equation for space effect:

S_a (Distributed Ceiling Array Effect) = $L_W - L_P$

Where:

$S_a = 5 \log x + 28 \log h - 1.13 \log N + 3 \log f - 31$ dB

x = ratio of floor area served by each outlet to the square of the ceiling height, ft.

h = ceiling height in ft.

N = number of evenly spaced outlets in the room, minimum four.

f = octave band center frequency in Hz.

Data based on the above calculation is presented in Table 13 for easy reference, based upon an array of four outlets for four different room heights with three different outlet areas. This table does not apply for a row of linear diffusers.

Area/ Diffuser	Ceiling Height	Octave Band 2	3	4	5	6	7
200 ft² (18.6 m²)	8 ft. (2.4 m)	2	3	4	5	6	7
300 ft² (27.8 m²)		3	4	5	6	7	8
400 ft² (37.2 m²)		4	5	6	7	8	9
200 ft² (18.6 m²)	9 ft. (2.7 m)	3	4	5	6	7	8
300 ft² (27.8 m²)		4	5	6	7	8	9
400 ft² (37.2 m²)		5	6	7	8	8	9
200 ft² (18.6 m²)	10 ft. (3.0 m)	4	5	6	7	8	9
300 ft² (27.8 m²)		5	6	7	8	9	10
400 ft² (37.2 m²)		6	7	7	8	9	10
200 ft² (18.6 m²)	12 ft. (3.6 m)	6	6	7	8	9	10
300 ft² (27.8 m²)		6	7	8	9	10	11
400 ft² (37.2 m²)		7	8	9	10	11	12

Table 13. Room Sound Attenuation for an Outlet Array, 4 outlets, dB.
Ref: ARI Std 885-98. Appendix D. Table D17, page 50.

Example 1:

Determining Sound Pressure Levels at Receiver Location from Radiated and Discharge Paths.

A size 4 – 12 Model 35SST 'STEALTH' fan powered terminal unit is selected to deliver 1100 cfm at 0.5" w.g. inlet static pressure, with 0.25" w.g. downstream resistance. The unit serves four rooms, each with its own supply outlet with the distribution ductwork as illustrated in Figure 7. The terminal unit is located in the ceiling plenum above a mineral fiber tile ceiling in one of the rooms.

The results are tabulated in Table 14 below – radiated sound being applicable only for the room above which the terminal is located and discharge sound applicable to each room.

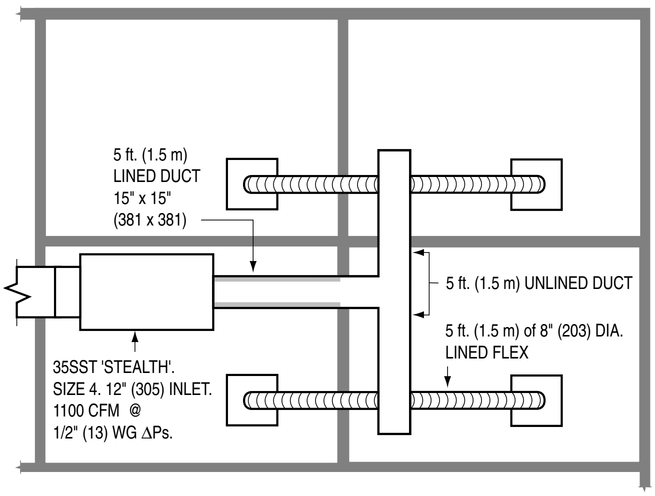


Figure 7: Illustration of working Example 1.
Individual room size: 18 ft. L x 15 ft. W x 9 ft. H (2430 ft.^{3}).

		Octave Band						
Radiated Sound Path		**2**	**3**	**4**	**5**	**6**	**7**	
L_w, Model 35SST 'STEALTH'	(Page D32)	63	57	51	43	39	37	
Environmental / Space Adjustment Factor	(Table 3)	–2	–1	0	0	0	0	
Ceiling / Space Effect	(Table 4)	–16	–18	–20	–26	–31	–36	
L_p, Radiated sound at receiver location		45	38	31	17	8	1	(NC = 26)
Discharge Sound Path		**2**	**3**	**4**	**5**	**6**	**7**	
L_w, Model 35SST 'STEALTH'	(Page D31)	64	64	66	63	59	57	
Environmental Adjustment Factor	(Table 3)	–2	–1	0	0	0	0	
5' Lined Duct, 15" x 15"	(Table 5)	–2	–4	–9	–18	–17	–12	
Rectangular tee ($^D/_2 = 7^1/_2$")	(Table 6a)	0	0	–1	–5	–8	–4	
Branch Power Division (50%)	(Table 7)	–3	–3	–3	–3	–3	–3	
5' Unlined Duct		0	0	0	0	0	0	
5' Lined Flex Duct, 8" dia.	(Table 8)	–6	–10	–18	–20	–21	–12	
Branch Power Division (50%)	(Table 7)	–3	–3	–3	–3	–3	–3	
End Reflection	(Table 9)	–9	–5	–2	0	0	0	
Space Effect V = 2500 ft^3, r = 5 ft	(Table 10)	–5	–6	–7	–8	–9	–10	
L_p, Discharge sound at receiver location		34	32	23	6	*	13	(NC = 19)

Table 14: Resultant Radiated and Discharge Sound Pressure Levels due to terminal unit for Example 1.
Note: * = Less than zero dB.

Example 2:

Determining Sound Pressure Level at Receiver Location from Air Outlet Sound Path.

Let us assume the same terminal selection and operating conditions as in the previous Example 1, but instead of supplying four rooms, the terminal is now supplying two larger rooms, each supplied by two diffusers. (Figure 8).

The diffusers selected are Nailor Model RNS, 24" x 24" (610 x 610) ceiling module, 8" (203) dia. neck, each handling 275 cfm (1100 ÷ 4). From page D82 in 2005 Air Distribution Catalog; NC = 20.

Octave Band	2	3	4	5	6	7
NC 20 Sound Power (Table 11)				22		
10 dB add to 5th band (room absorption)				+10		
4th band=5th+3. 6th band=5th–5			35	32	27	
Env. Adjustment Factor (Table 3)			0	0	0	
Space Effect * (Table 10)			-9	-9	-10	
Two Outlets (Table 12)			+3	+3	+3	
Lp, Outlet Sound at Receiver Location (NC=24)			29	26	20	
* Space Effect: V = 5000 ft.^{3}, r = 5 ft.						

Table 15: Air Outlet Sound Path resultant Sound Pressure Levels for Example 2.

Example 3:

Determining Total Overall Sound Pressure Level at Receiver Location.

Using the same terminal selection, operating conditions, room layout and diffuser selection as in Example 2 above, the contributions of the critical sound paths must be combined to obtain the total Lp at the receiver location. The discharge and radiated sound paths are modeled in a similar fashion to Example 1, but with adjustments for room size and number of outlets.

Octave Band	2	3	4	5	6	7
L_W, Model 35SST 'STEALTH' (P. D32)	63	57	51	43	39	37
Env. Adjustment Factor (Table 3)	-2	-1	0	0	0	0
Ceiling/Space Effect (Table 4)	-16	-18	-20	-26	-31	-36
Lp, Radiated Sound at Receiver Location (NC=26)	45	38	31	17	8	1

Table 16: Radiated Sound Path resultant Sound Pressure Levels due to terminal unit for Example 3.

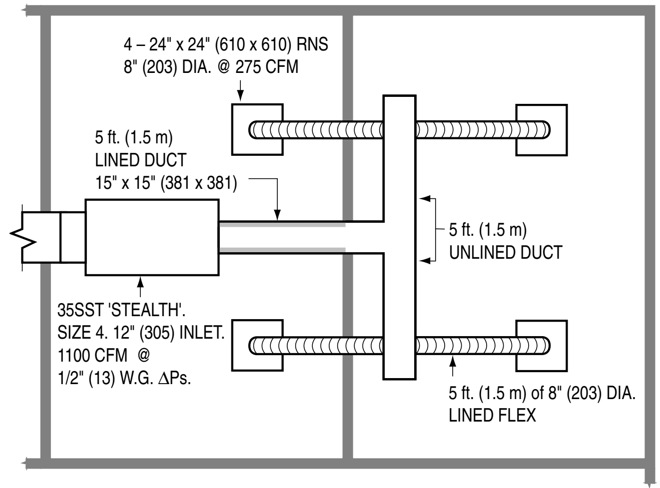


Figure 8: Illustration of working Example 2 and 3.
Individual room size = 30 ft. L x 18 ft. W x 9 ft. H (4860 ft.^{3}).

Octave Band	2	3	4	5	6	7
Lw, Model 35SST 'STEALTH' (P. D31)	64	64	66	63	59	57
Env. Adjustment Factor (Table 3)	-2	-1	0	0	0	0
5' Lined Duct 15" x 15" (Table 5)	-2	-4	-9	-18	-17	-12
Rectangular Tee (Table 6a)	0	0	-1	-5	-8	-4
Branch Power Division (50%) (Table 7)	-3	-3	-3	-3	-3	-3
5' Unlined Duct	0	0	0	0	0	0
5' Lined Flex, 8" Dia. (Table 8)	-6	-10	-18	-20	-21	-12
Branch Power Division (Table 7)	-3	-3	-3	-3	-3	-3
End Reflection (Table 9)	-9	-5	-2	0	0	0
Space Effect * (Table 10)	-7	-8	-9	-9	-10	-11
Two Outlets (Table 12)	+3	+3	+3	+3	+3	+3
Lp, Discharge Sound at Receiver Location (NC=20)	35	33	24	8	0	15
* Space Effect: V = 5000 ft.^{3}, r = 5 ft.						

Table 17: Discharge Sound Path resultant Sound Pressure Levels due to terminal unit for Example 3.

Total Overall Sound

Total Sound = Radiated + Discharge + Air Outlet

The paths are totalled in each octave band using logarithmic addition. Figure 6 may be used as an approximation to save time and simplify this calculation.

Calculation:

$$\text{Total } L_P(\text{octave band}) = 10\log_{10}\left[10^{\left(\frac{Rad.}{10}\right)}+10^{\left(\frac{Disch.}{10}\right)}+10^{\left(\frac{Outlet}{10}\right)}\right]$$

		Rad.		Disch.		Outlet	
Octave Band	2 =	45	+	35	+	–	= 45
	3 =	38	+	33	+	–	= 39
	4 =	31	+	24	+	29	= 34
	5 =	17	+	8	+	26	= 27
	6 =	8	+	0	+	20	= 20
	7 =	1	+	15	+	–	= 15

Plot sound pressure levels on NC curve chart. (Figure 1).

Result: Overall Room NC Level = 29.

Acoustic Design and Installation Considerations

To help ensure an acceptable NC Level in the occupied space, engineers can minimize the sound contribution of air terminals by taking into account several design considerations and by using the following guidelines for good design practice.

1. Design systems to operate at low (minimum) supply static pressure at the primary air inlet. This will reduce the generated sound level, provide more energy efficient operation and allow the central fan to be downsized. Excessive static pressure generates noise.

2. Use metal ducts before the inlet. Flexible duct allows significantly greater breakout noise and should be avoided wherever possible. Flexible duct can also generate sound if bends or sagging are present.

3. Select terminals to operate toward the middle area of their operating range. Larger inlets reduce velocity and hence noise. For fan powered terminals, lower fan speeds produce lower sound levels. Sound emissions will be lower when fan speed controllers are used to reduce fan rpm rather than using mechanical dampers to restrict airflow.

4. Whenever possible, locate terminals above non-critical areas that are less sensitive to noise such as corridors, copy rooms or storage/file rooms. This will isolate critical areas from potential radiated noise.

5. Locate terminals in the largest ceiling plenum space available in order to maximize radiated noise reduction. Install terminals at highest practical point above ceiling in order to optimize radiated sound dissipation.

6. Avoid locating terminals near return air openings or light fixtures. This allows a direct path for radiated sound to enter the space without the benefit of ceiling attenuation.

7. Locate terminals to allow the use of lined discharge ductwork to help attenuate discharge sound.

8. To avoid possible aerodynamic noise, keep airflow velocities below 1000 fpm (5 m/s) in branch ducts and below 800 fpm (4 m/s) in run-outs to air outlet devices.

9. Consider the use of a larger number of smaller air outlets to minimize outlet generated sound. Insulated flexible duct on diffuser run-outs provides excellent attenuation performance.

10. The use of ceilings with a high sound transmission loss classification will help reduce radiated sound.

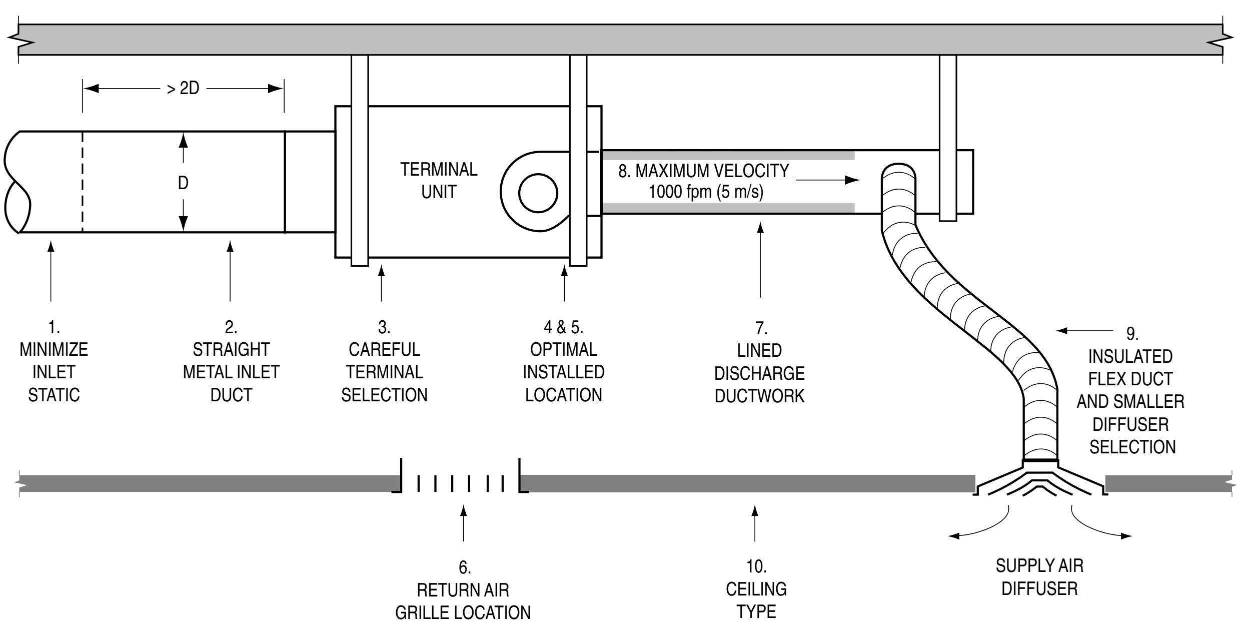


Figure 9: Guidelines for VAV and Fan Powered Terminal Unit Installation for Optimal Acoustic Performance.

Useful Formulas and Definitions

Airflow

Q = V x A
Q = Airflow Rate, cfm (l/s)
V = Velocity, fpm (m/s)
A = Area, ft^2 (m^2)

Pressure

Imperial Units

$$VP\ (\text{" w.g.}) = \left(\frac{V\ (fpm)}{4005}\right)^2$$

Metric Units

$$VP\ (Pa) = \left(\frac{V\ (m/s)}{1.3}\right)^2$$

VP = Velocity Pressure

TP = SP + VP
TP = Total Pressure, " w.g. (Pa)
SP = Static Pressure, " w.g. (Pa)

Heat Transfer

Imperial Units

H = 1.085 x cfm x Δt (°F)
H = Heat Transfer, Btu's/hr.
Btu = British Thermal Unit
Δt = Temp. Differential

Metric Units

H= 1.23 x l/s x Δt (°C)
H = Heat Transfer, watts

Water Coils

Imperial Units

$$\Delta t(^\circ F) = 927 \times \frac{Mbh}{cfm}$$

Metric Units

$$\Delta t\ (^\circ C) = 829 \times \frac{kW}{l/s}$$

Δt = Air Temperature Rise
Mbh = 1000's of Btu's/hr.

$$\Delta t(^\circ F) = 2.04 \times \frac{Mbh}{GPM} \qquad \Delta t\ (^\circ C) = 0.244 \times \frac{kW}{l/s}$$

Δt = Water Temperature Drop
GPM = Water Flow, gallons per minute
l/s = Liters per second

Electric Coils

$$\Delta t(^\circ F) = \frac{kW \times 3160}{cfm}$$

$$kW = \frac{cfm \times \Delta t}{3160}$$

Δt = Air Temperature Rise
kW = Kilowatts

Power DC Circuits

$$hp = \frac{E \times I \times Eff.}{746}$$

W = E x I

$$Eff. = \frac{746 \times bhp}{W}$$

Power AC Circuits (Single Phase)

$$PF = \frac{W}{E \times I}$$

$$I = \frac{746 \times hp}{E \times Eff. \times PF}$$

$$Eff. = \frac{746 \times hp}{E \times I \times PF}$$

$$kW = \frac{E \times I \times PF \times Eff.}{1000}$$

$$hp = \frac{E \times I \times PF \times Eff.}{746}$$

$$kVA = \frac{I \times E}{1000}$$

Power AC Circuits (Three Phase)

$$PF = \frac{W}{E \times I \times 1.732}$$

$$I = \frac{746 \times hp}{1.732 \times E \times PF \times Eff.}$$

$$Eff. = \frac{746 \times hp}{E \times I \times PF \times 1.732}$$

$$kW = \frac{E \times I \times PF \times 1.732}{1000}$$

$$hp = \frac{E \times I \times 1.732 \times PF \times Eff.}{746}$$

$$kVA = \frac{1.732 \times I \times E}{1000}$$

PF = Power Factor
W = Watts
E = Volts
I = Amperes
hp = Horsepower
Eff. = Efficiency

Imperial/Metric Guide Conversion Factors

Quantity	Imperial Unit	Metric Unit	From Imperial To Metric Multiply By:	From Metric To Imperial Multiply By:
Area	square foot square inch	square meter (m^2) square millimeter (mm^2)	0.0929 645.16	10.764 .00155
Density	pounds per cubic foot	kilograms per cubic meter (kg/M^3)	16.018	.0624
Energy	British thermal unit (BTU) kilowatt hour watts per second horsepower hour	joule (J) megajoule (MJ) joule (J) megajoule (MJ)	1055.056 3.6 1.0 2.6845	.000948 .2778 1.0 .3725
Force	ounce force pound force kilogram force	newton (N) newton (N) newton (N)	.278 4.4482 9.8067	3.597 .2248 .102
Heat	BTU per hour BTU per pound	watt (W) joules per kilogram (J/kg)	.2931 2326.0	3.412 .00043
Length	inch foot foot yard	millimeter (mm) millimeter (mm) meter (m) meter (m)	25.4 304.8 .3048 .9144	.0394 .00328 3.2808 1.0936
Mass (weight)	ounce (avoirdupois) pound (avoirdupois)	gram (g) kilogram (kg)	28.350 .4536	.0353 2.2046
Power	horsepower horsepower (boiler) foot pound - force per minute ton of refrigeration	kilowatt (kW) kilowatt (kW) watt (W) kilowatt (kW)	.7457 9.8095 .0226 3.517	1.341 .1019 44.254 .2843
Pressure	inch of water column foot of water column inch of mercury column ounces per square inch pounds per square inch	kilopascal (kPa) kilopascal (kPa) kilopascal (kPa) kilopascal (kPa) kilopascal (kPa)	.2486 2.9837 3.3741 .4309 6.8948	4.0219 .3352 .2964 2.3206 .145
Temperature	Fahrenheit	Celsius (°C)	5/9(°F-32)	(9/5°C)+32
Torque	ounce - force inch pound - force inch pound - force foot	millinewton-meter (mN.m) newton-meter (N.m) newton-meter (N.m)	7.0616 .1130 1.3558	.1416 8.8495 .7376
Velocity	feet per second feet per minute miles per hour	meters per second (m/s) meters per second (m/s) meters per second (m/s)	.3048 .00508 .44704	3.2808 196.85 2.2369
Volume (capacity)	cubic foot cubic inch cubic yard gallon (U.S.) gallon (imperial)	liter (l) cubic centimeter (cm^2) cubic meter (m^3) liter (l) liter (l)	28.3168 16.3871 .7646 3.785 4.546	.03531 .06102 1.308 .2642 .2120
Volume (flow)	cubic feet per minute (cfm) cubic feet per minute (cfm) cubic feet per hour (cfh) gallons per minute (U.S.) gallons per minute (imperial)	liters per second (l/s) cubic meters per second (m^3/s) milliliters per second (ml/s) liters per second (l/s) liters per second (l/s)	.4719 .0004719 7.8658 .06309 0.7577	2.119 2119.0 .127133 15.850 13.198

Pressure Measurement

Concepts of Pressure. Pressure is force per unit area. This may also be defined as energy per unit volume of fluid. There are three categories of pressure — Total Pressure, Static Pressure and Velocity Pressure. They are all associated with air handling. Unit of pressure is expressed in inches of water, designated **in. w.g.**

Static Pressure is the normal force per unit area at a small hole in the wall of a duct or other boundaries. It is a function of air density and degree of compression. It may be thought of as the pressure in a tire or in a balloon which extends in all directions.

Velocity Pressure is the force per unit area capable of causing an equivalent velocity in moving air. Velocity pressure is a function of air density and velocity. At standard air density, the relationship between velocity pressure and velocity is expressed in the following formula:

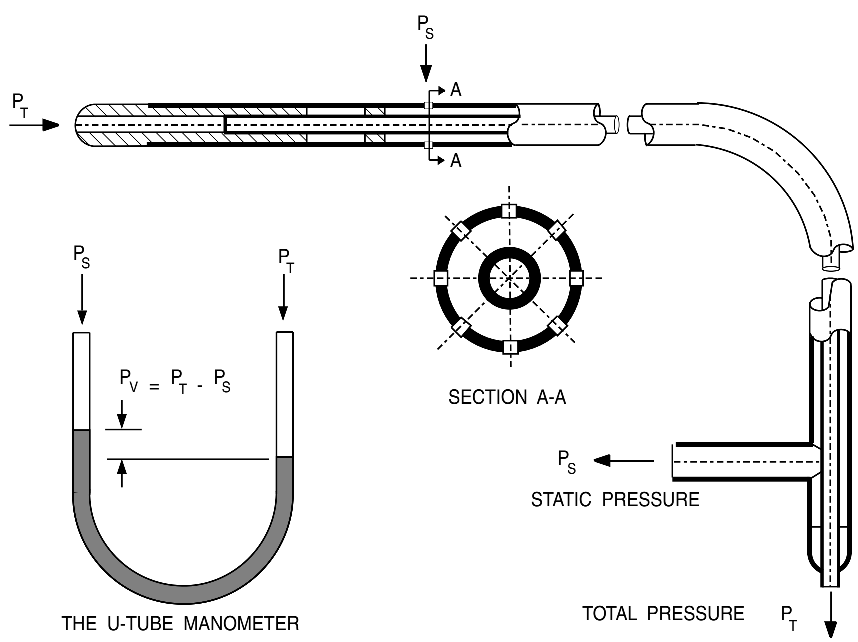
$$Pv = \left(\frac{V}{4005}\right)^2 \text{ or } V = 4005\sqrt{Pv}$$

Where: V = Air Velocity (FPM)
Pv = Velocity Pressure (in. w.g.)

Total Pressure, as its name implies, is the sum of static pressure and velocity pressure.

The Pitot Static Tube is an instrument used to measure pressure and velocities as illustrated below. It is constructed of two tubes. The inner, or impact tube, senses the total pressure as the impact opening faces upstream. The outer tube senses only the static pressure, which communicates with the airstream through small holes in its wall.

The U-Tube Manometer connects both parts of the Pitot Static Tube. The manometer functions as a subtracting device to give a reading of velocity pressure.



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ENGINEERING GUIDE

CONVERSION CHART for converting *VELOCITY PRESSURE* in inches of water to *VELOCITY* in feet per minute

Note: This chart is based upon standard air conditions of 70° Fahrenheit and 29.92 inches of mercury (barometric pressure), and assumes that the airflow is essentially non-compressible (under 10 inches of water pressure); as reflected by the following formula.

$$\text{Velocity (fpm)} = 4005\sqrt{\text{Velocity Pressure in inches of water}}$$

VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V	VP	V
.001"	127	.062"	996	.123"	1404	.184"	1718	.245"	1982	.306"	2215	.367"	2426	.77"	3514	1.38"	4705	1.99"	5651	2.60"	6458
.002"	179	.063"	1004	.124"	1410	.185"	1723	.246"	1987	.307"	2219	.368"	2429	.78"	3537	1.39"	4722	2.00"	5664	2.61"	6470
.003"	219	.064"	1012	.125"	1416	.186"	1727	.247"	1991	.308"	2223	.369"	2433	.79"	3560	1.40"	4739	2.01"	5678	2.62"	6482
.004"	253	.065"	1020	.126"	1422	.187"	1732	.248"	1995	.309"	2226	.370"	2436	.80"	3582	1.41"	4756	2.02"	5692	2.63"	6495
.005"	283	.066"	1029	.127"	1427	.188"	1737	.249"	1999	.310"	2230	.371"	2439	.81"	3604	1.42"	4773	2.03"	5706	2.64"	6507
.006"	310	.067"	1037	.128"	1433	.189"	1741	.250"	2003	.311"	2233	.372"	2443	.82"	3625	1.43"	4790	2.04"	5720	2.65"	6519
.007"	335	.068"	1045	.129"	1439	.190"	1746	.251"	2007	.312"	2236	.373"	2445	.83"	3657	1.44"	4806	2.05"	5734	2.66"	6532
.008"	358	.069"	1052	.130"	1444	.191"	1750	.252"	2011	.313"	2239	.374"	2449	.84"	3669	1.45"	4823	2.06"	5748	2.67"	6544
.009"	380	.070"	1060	.131"	1449	.192"	1755	.253"	2015	.314"	2242	.375"	2453	.85"	3690	1.46"	4840	2.07"	5762	2.68"	6556
.010"	400	.071"	1067	.132"	1455	.193"	1759	.254"	2019	.315"	2245	.376"	2456	.86"	3709	1.47"	4856	2.08"	5776	2.69"	6569
.011"	420	.072"	1075	.133"	1461	.194"	1764	.255"	2023	.316"	2248	.377"	2459	.87"	3729	1.48"	4873	2.09"	5790	2.70"	6581
.012"	439	.073"	1082	.134"	1466	.195"	1768	.256"	2027	.317"	2251	.378"	2462	.88"	3758	1.49"	4889	2.10"	5804	2.71"	6593
.013"	457	.074"	1089	.135"	1471	.196"	1773	.257"	2031	.318"	2254	.379"	2466	.89"	3779	1.50"	4905	2.11"	5817	2.72"	6605
.014"	474	.075"	1097	.136"	1477	.197"	1777	.258"	2035	.319"	2257	.380"	2469	.90"	3800	1.51"	4921	2.12"	5831	2.73"	6617
.015"	491	.076"	1104	.137"	1482	.198"	1782	.259"	2039	.320"	2260	.381"	2472	.91"	3821	1.52"	4938	2.13"	5845	2.74"	6629
.016"	507	.077"	1111	.138"	1488	.199"	1787	.260"	2042	.321"	2264	.382"	2475	.92"	3842	1.53"	4954	2.14"	5859	2.75"	6641
.017"	522	.078"	1119	.139"	1493	.200"	1791	.261"	2046	.322"	2268	.383"	2479	.93"	3863	1.54"	4970	2.15"	5872	2.76"	6654
.018"	537	.079"	1125	.140"	1498	.201"	1795	.262"	2050	.323"	2272	.384"	2481	.94"	3884	1.55"	4986	2.16"	5886	2.77"	6666
.019"	552	.080"	1133	.141"	1504	.202"	1800	.263"	2054	.324"	2276	.385"	2485	.95"	3904	1.56"	5002	2.17"	5899	2.78"	6678
.020"	566	.081"	1140	.142"	1509	.203"	1804	.264"	2058	.325"	2280	.386"	2488	.96"	3924	1.57"	5018	2.18"	5913	2.79"	6690
.021"	580	.082"	1147	.143"	1515	.204"	1809	.265"	2062	.326"	2284	.387"	2491	.97"	3945	1.58"	5034	2.19"	5927	2.80"	6702
.022"	594	.083"	1154	.144"	1520	.205"	1813	.266"	2066	.327"	2289	.388"	2495	.98"	3965	1.59"	5050	2.20"	5940	2.81"	6714
.023"	607	.084"	1161	.145"	1525	.206"	1818	.267"	2070	.328"	2293	.389"	2499	.99"	3985	1.60"	5066	2.21"	5954	2.82"	6725
.024"	620	.085"	1167	.146"	1530	.207"	1822	.268"	2074	.329"	2297	.390"	2501	1.00"	4005	1.61"	5082	2.22"	5967	2.83"	6737
.025"	633	.086"	1175	.147"	1536	.208"	1827	.269"	2078	.330"	2301	.40"	2533	1.01"	4025	1.62"	5098	2.23"	5981	2.84"	6749
.026"	645	.087"	1181	.148"	1541	.209"	1831	.270"	2081	.331"	2304	.41"	2563	1.02"	4045	1.63"	5114	2.24"	5994	2.85"	6761
.027"	658	.088"	1188	.149"	1546	.210"	1835	.271"	2085	.332"	2308	.42"	2595	1.03"	4064	1.64"	5129	2.25"	6008	2.86"	6773
.028"	670	.089"	1193	.150"	1551	.211"	1839	.272"	2089	.333"	2311	.43"	2626	1.04"	4084	1.65"	5144	2.26"	6021	2.87"	6785
.029"	682	.090"	1201	.151"	1556	.212"	1844	.273"	2093	.334"	2315	.44"	2656	1.05"	4103	1.66"	5160	2.27"	6034	2.88"	6797
.030"	694	.091"	1208	.152"	1561	.213"	1848	.274"	2097	.335"	2318	.45"	2687	1.06"	4123	1.67"	5175	2.28"	6047	2.89"	6809
.031"	705	.092"	1215	.153"	1567	.214"	1853	.275"	2101	.336"	2322	.46"	2716	1.07"	4142	1.68"	5191	2.29"	6060	2.90"	6820
.032"	716	.093"	1222	.154"	1572	.215"	1857	.276"	2105	.337"	2325	.47"	2746	1.08"	4162	1.69"	5206	2.30"	6074	2.91"	6832
.033"	727	.094"	1228	.155"	1577	.216"	1862	.277"	2119	.338"	2329	.48"	2775	1.09"	4181	1.70"	5222	2.31"	6087	2.92"	6844
.034"	738	.095"	1234	.156"	1582	.217"	1866	.278"	2113	.339"	2332	.49"	2804	1.10"	4200	1.71"	5237	2.32"	6100	2.93"	6855
.035"	749	.096"	1241	.157"	1587	.218"	1870	.279"	2116	.340"	2335	.50"	2832	1.11"	4219	1.72"	5253	2.33"	6113	2.94"	6867
.036"	759	.097"	1247	.158"	1592	.219"	1875	.280"	2119	.341"	2338	.51"	2860	1.12"	4238	1.73"	5268	2.34"	6126	2.95"	6879
.037"	770	.098"	1254	.159"	1597	.220"	1879	.281"	2123	.342"	2342	.52"	2888	1.13"	4257	1.74"	5283	2.35"	6139	2.96"	6890
.038"	780	.099"	1260	.160"	1602	.221"	1883	.282"	2127	.343"	2345	.53"	2916	1.14"	4276	1.75"	5298	2.36"	6152	2.97"	6902
.039"	791	.100"	1266	.161"	1607	.222"	1887	.283"	2131	.344"	2349	.54"	2943	1.15"	4295	1.76"	5313	2.37"	6165	2.98"	6913
.040"	801	.101"	1273	.162"	1612	.223"	1892	.284"	2135	.345"	2352	.55"	2970	1.16"	4314	1.77"	5328	2.38"	6179	2.99"	6925
.041"	811	.102"	1279	.163"	1617	.224"	1896	.285"	2139	.346"	2356	.56"	2997	1.17"	4332	1.78"	5343	2.39"	6191	3.00"	6937
.042"	821	.103"	1285	.164"	1622	.225"	1900	.286"	2143	.347"	2360	.57"	3024	1.18"	4350	1.79"	5359	2.40"	6204	3.01"	6948
.043"	831	.104"	1292	.165"	1627	.226"	1905	.287"	2147	.348"	2363	.58"	3050	1.19"	4368	1.80"	5374	2.41"	6217	3.02"	6960
.044"	840	.105"	1298	.166"	1632	.227"	1909	.288"	2151	.349"	2366	.59"	3076	1.20"	4386	1.81"	5388	2.42"	6230	3.03"	6971
.045"	849	.106"	1304	.167"	1637	.228"	1913	.289"	2154	.350"	2369	.60"	3102	1.21"	4405	1.82"	5403	2.43"	6243	3.04"	6983
.046"	859	.107"	1310	.168"	1642	.229"	1917	.290"	2157	.351"	2372	.61"	3127	1.22"	4423	1.83"	5418	2.44"	6256	3.05"	6994
.047"	868	.108"	1316	.169"	1646	.230"	1921	.291"	2161	.352"	2376	.62"	3153	1.23"	4442	1.84"	5433	2.45"	6269	3.06"	7006
.048"	877	.109"	1322	.170"	1651	.231"	1925	.292"	2164	.353"	2379	.63"	3179	1.24"	4460	1.85"	5447	2.46"	6281	3.07"	7017
.049"	887	.110"	1328	.171"	1656	.232"	1929	.293"	2168	.354"	2383	.64"	3204	1.25"	4478	1.86"	5462	2.47"	6294	3.08"	7028
.050"	896	.111"	1334	.172"	1661	.233"	1933	.294"	2171	.355"	2386	.65"	3229	1.26"	4495	1.87"	5477	2.48"	6307	3.09"	7040
.051"	904	.112"	1340	.173"	1666	.234"	1937	.295"	2175	.356"	2389	.66"	3254	1.27"	4513	1.88"	5491	2.49"	6319	3.10"	7051
.052"	913	.113"	1346	.174"	1670	.235"	1941	.296"	2179	.357"	2393	.67"	3279	1.28"	4531	1.89"	5506	2.50"	6332	3.11"	7063
.053"	922	.114"	1352	.175"	1675	.236"	1945	.297"	2182	.358"	2396	.68"	3303	1.29"	4549	1.90"	5521	2.51"	6345	3.12"	7074
.054"	931	.115"	1358	.176"	1680	.237"	1950	.298"	2186	.359"	2400	.69"	3327	1.30"	4566	1.91"	5535	2.52"	6358	3.13"	7085
.055"	939	.116"	1364	.177"	1685	.238"	1954	.299"	2189	.360"	2403	.70"	3351	1.31"	4583	1.92"	5550	2.53"	6370	3.14"	7097
.056"	948	.117"	1370	.178"	1690	.239"	1958	.300"	2193	.361"	2406	.71"	3375	1.32"	4601	1.93"	5564	2.54"	6383	3.15"	7108
.057"	956	.118"	1376	.179"	1695	.240"	1962	.301"	2197	.362"	2410	.72"	3398	1.33"	4619	1.94"	5579	2.55"	6395	3.16"	7119
.058"	964	.119"	1382	.180"	1699	.241"	1966	.302"	2200	.363"	2413	.73"	3422	1.34"	4636	1.95"	5593	2.56"	6408	3.17"	7131
.059"	973	.120"	1387	.181"	1704	.242"	1970	.303"	2204	.364"	2416	.74"	3445	1.35"	4653	1.96"	5608	2.57"	6420	3.18"	7142
.060"	981	.121"	1393	.182"	1709	.243"	1974	.304"	2208	.365"	2420	.75"	3468	1.36"	4671	1.97"	5623	2.58"	6433	3.19"	7153
.061"	989	.122"	1399	.183"	1713	.244"	1978	.305"	2212	.366"	2423	.76"	3491	1.37"	4688	1.98"	5637	2.59"	6445	3.20"	7164

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"Complete Air Control and Distribution Solutions"

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