TYPES OF TERMINALS

SINGLE DUCT

This basic terminal consists of casing, a damper, a damper actuator and associated controls. In response to control signals from a thermostat or other source, the terminal varies the airflow through a single duct handling hot or cold air. In some applications the same terminal is used for both heating and cooling; a dual function thermostat, together with the necessary change-over circuitry, makes this possible. Controls can be pneumatic, electric, analog electronic or direct digital electronic. Accessories such as round outlets, multiple outlets and sound attenuators may be added. The single duct terminal is most often used in an interior zone of the building, for cooling only.

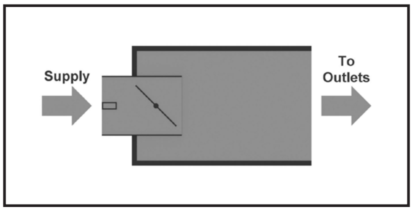

Figure 47. Elevation - Single Duct

DUAL DUCT, NON–MIXING

Essentially the same as two single duct terminals side-by-side, this terminal modulates the flow of hot and cold air in two separate streams supplied by a dual duct central air handling unit. Because there is no provision for mixing the two airstreams, this terminal should not be used for simultaneous heating and cooling, which would result in stratification in the discharge duct. (When stratification occurs, the several outlets served by the terminal may deliver air at noticeably different temperatures.) The non-mixing, dual duct terminal is best used in an exterior zone, in which zero-to-low airflow can be tolerated as the temperature requirement shifts from cooling to heating.

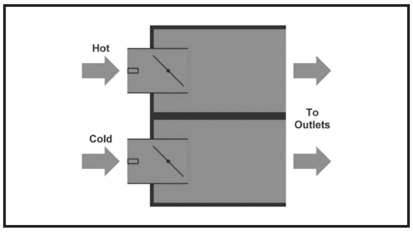

Figure 48. Plan View - Dual Duct, Non-Mixing

DUAL DUCT, MIXING

Here the terminal is designed specifically for mixing hot (or tempered ventilation) and cold air in any proportion. When equipped with pneumatic controls, there is a velocity sensor in the hot air inlet, but none in the cold air inlet. A velocity sensor at the discharge measures the total flow of air and sends the signal to the cold air controller. In the mixing cycle, the hot airflow changes first, and a change in cold airflow follows to maintain a constant total (mixed) volume. When equipped with DDC controls by Titus, both hot and cold inlets have velocity sensors, with the summation of flows computed by the microprocessor. No discharge velocity sensor is used. This dual duct terminal is often used in an exterior zone of a building or to ensure ventilation rates.

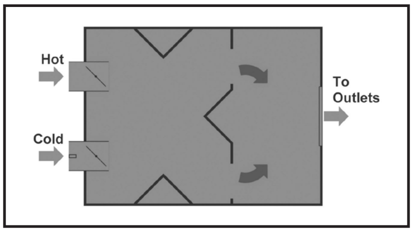

Figure 49. Plan View - Duct Duct, Mixing

SINGLE DUCT, WITH HEATING COIL

This is the single duct terminal described above, with a heating coil added. The coil may be of either the hot water or the electric type. The hot water coil is usually modulated by a proportioning valve controlled by the same thermostat that controls the terminal. Control for the electric coil is either 100% on-off or in steps of capacity, energized by contactors in response to the room thermostat. The single duct terminal with heating coil is most often used in an exterior zone with moderate heating requirements. Since the terminal normally handles its minimum cfm in the heating mode, a dual minimum cfm or "flip-flop" control can be added for increased heating airflow. Separate minimum cfm setpoints are standard with most DDC controls (available optionally on most other control types) and should be considered in design. A higher minimum cfm in heating mode will improve overhead air distribution performance.

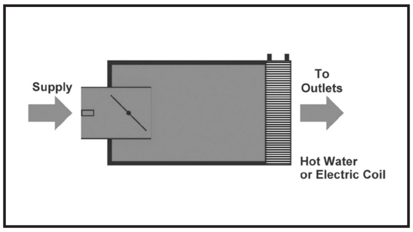

Figure 50. Elevation - Single Duct, with Heating Coil

FAN POWERED, PARALLEL TYPE (VARIABLE VOLUME)

In this terminal a fan is added to recirculate plenum air, for heating only. The heating cycle occurs generally when the primary air is off or at minimum flow. Heat is picked up as the recirculated air is drawn from the ceiling space and the fan motor. Additional heat can be provided by a hot water or electric coil on the terminal. Because the fan handles only the heating airflow, which is usually less than that for cooling, the fan can be sized smaller than in the series flow type terminal (see below). During the cooling cycle, the fan is off and cool primary air is supplied from the central system. A backdraft damper prevents reverse flow through the fan. The flow of the primary air is regulated by variable air volume controls. Used in exterior zones.

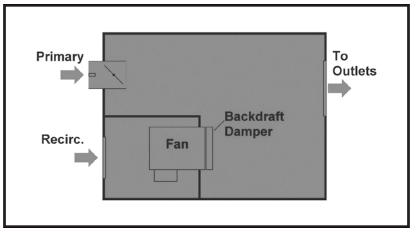

Figure 51. Plan View - Fan Powered, Parallel Type (Variable Volume)

FAN POWERED, SERIES TYPE (CONSTANT VOLUME)

The fan runs continuously, fed by a mixture of primary and plenum air. The more primary air is forced in, the less plenum air is drawn in. The result is variable volume from the central system, constant volume (and sound) to the room. Because the central system need only deliver air as far as the fan, the inlet static pressure can be lower than in the parallel flow terminal (above). The fan, however, is sized to handle the total airflow. These are often used in applications where constant background sound

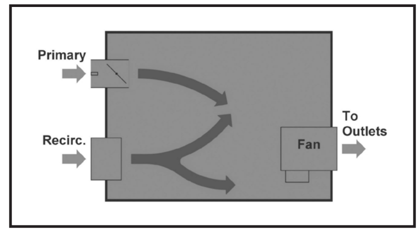

Figure 52. Plan View - Fan Powered, Series Type (Constant Volume)

LOW TEMPERATURE FAN TERMINALS

The fan terminal, with its inherent mixing, is well suited to handle the very cold air delivered by systems designed for air much colder than with conventional 55°F supply systems. In order to use standard diffusers, the primary air must be raised to a conventional supply temperature before it enters the room. A commonly utilized solution is to mix it with recirculated air with a fan powered terminal. The most common application uses a Series Flow unit, but many applications have been utilized with Parallel units with a constant running fan.

FAN POWERED, LOW PROFILE

This series or parallel type terminal has a vertical dimension of only 10.5" for all sizes, to minimize the depth of ceiling space required. Notice in the diagram at the right that the recirculating fan is laid flat on its side, shaft vertical. In localities where building heights are limited, the low profile terminal saves enough space to allow extra floors to be included in a high-rise structure. Ceiling space can be as little as 12" to 14" deep. The low profile terminal is also useful in buildings constructed with precast concrete channel floors. The terminal can fit into the channel space with no extra depth required (Series type shown).

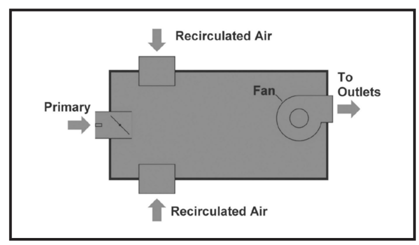

Figure 53. Plan View - Fan Powered, Low Profile

FAN POWERED, ACCESS FLOOR PROFILE (CONSTANT VOLUME)

This series type terminal is designed to fit around the pedestal support grid of access, or raised, floor systems. In a typical access floor the grid is 24" x 24". The terminal can fit into the floor plenum without any modifications to the pedestal system.

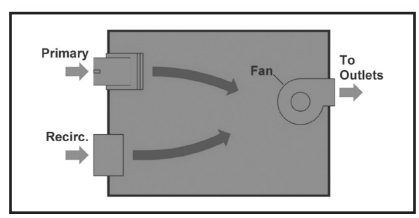

Figure 54. Plan View - Fan Powered, Access Floor Profile (Constant Volume)

REACTION TO DUCT PRESSURE CONTROLS

PRESSURE INDEPENDENT

With this type of control the terminal maintains the flow rate required to handle the heating or cooling load, regardless of system pressure fluctuations. It is the best choice where the system pressure will vary extensively and where precise control is essential. Key components in pressure independent control are the velocity sensor, which furnishes a continuous reading of the air velocity through the terminal, and the velocity controller, which processes this information along with signals from the thermostat. In the chart (**Figure 55**), vertical lines AB and EF represent minimum and maximum cfm settings which are adjustable at the controller. Line CD represents any cfm setting maintained by the controller in response to the thermostat. The damper will open and close as needed to hold the cfm constant up and down this vertical line for the full range of pressure drops shown. Notice that the vertical cfm lines are cut off by the diagonal line AE, which represents the pressure drop from inlet to outlet with the damper wide open. This is the minimum DP shown in our data.

PRESSURE DEPENDENT

A terminal with this type of control is designed for those applications where neither pressure independence nor cfm limit regulation is required. An example is a variable volume makeup air supply in which the downstream duct pressure is held constant by other controls. The terminal consists essentially of a casing, a damper and a damper actuator. There is no controller and no velocity sensor; the damper moves in direct response to the thermostat or other signal input. The line AB (**Figure 56**) shows the typical performance characteristic. It represents a given damper setting, with the flow rate varying as the square root of the static pressure drop through the terminal. This, of course, is typical of any damper or fixed orifice. Lines CD and EF represent random additional settings as the damper opens to the full open position line GH. Line GH is the minimum pressure loss of the assembly.

Most of the control types shown here have certain principle elements in common:

ROOM THERMOSTAT OR SENSOR

The thermostat contains not only a temperature sensing element, but also a means of changing the setpoint. The room sensor used with the direct digital control system is simply an electronic temperature sensor; setpoint changes are handled along with other signal processing in the digital controller.

VELOCITY SENSOR

Mounted in the inlet of the terminal, this device senses air velocity, which can easily be converted to airflow rate. The sensor's signal provides feedback to monitor and directs the operation of the controller and damper actuator.

CONTROLLER

Commands from the thermostat or room sensor, together with feedback from the velocity sensor, are processed in the controller to regulate the damper actuator. Operation is pressure independent.

Note: Excessive airflow may lead to excessive noise. Pressure independent control has less opportunity for variable (and unwanted) sounds in the occupied spaces.

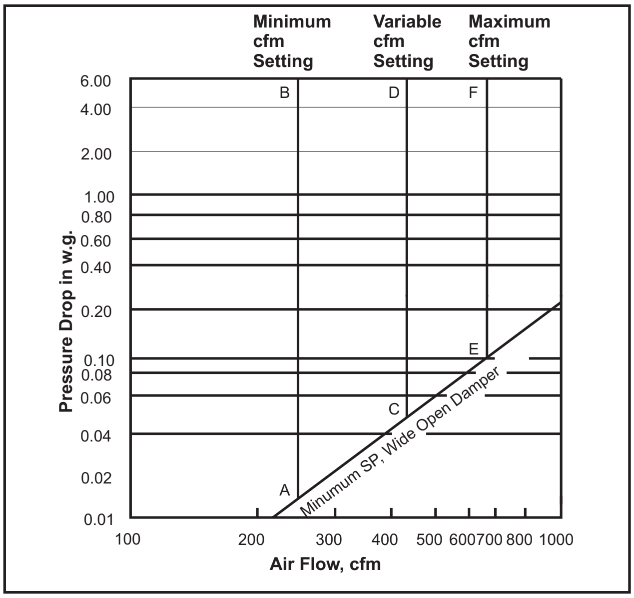

Figure 55. Pneumatic Pressure Independent

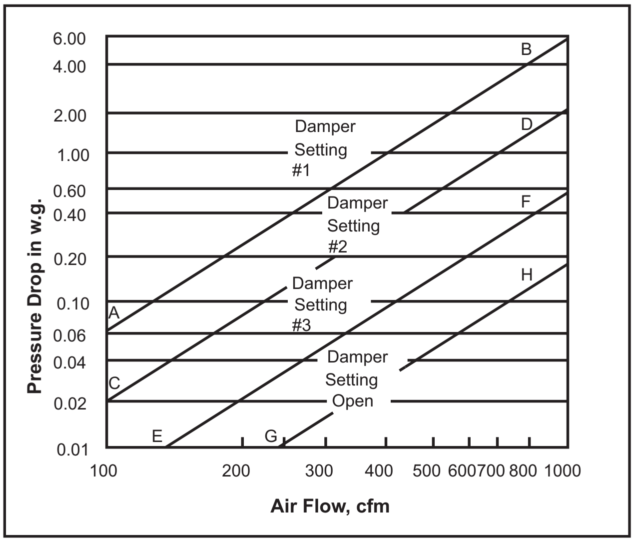

Figure 56. Pneumatic Pressure Dependent

DAMPER ACTUATOR

The damper actuator opens and closes the damper to change the airflow, or to hold it constant, as dictated by the controller.

PNEUMATIC SYSTEMS

In a pneumatic control system, the various components are powered by compressed air, usually at 15-25 psi, from a central system. The thermostat receives air at full pressure directly from the main air supply. In response to room temperature, the air pressure is modulated to the controller, which regulates the damper actuator. The sensor and controller compensate for changes in duct pressure so that operation is pressure independent.

ELECTRIC SYSTEMS - FIGURE 58A

Electric controls operate at low voltage, usually 24 VAC, supplied by a transformer which is often built into the control box of the terminal. The room thermostat has single-pole-double-throw contacts so that (in the cooling mode) a rise in temperature drives the damper actuator in the opening direction; a fall in temperature reverses the actuator. Since the electric system has no velocity sensor and no controller, there is no compensation for duct pressure fluctuations. Operation of the terminal is pressure dependent, the thermostat and room response time are typically much less than the actuator response time, and excessive room temperature variations are a likely result.

ANALOG ELECTRONIC SYSTEMS - FIGURE 58B

Like the electric controls, analog electronic controls operate at low voltage, usually 24 VAC, supplied by a transformer which is often built into the control box of the terminal. These controls, however, also include a velocity sensor of either the thermistor type, or pneumatic velocity sensor with electronic transducer, together with an electronic velocity controller that is pressure independent. The electronic thermostat can control both cooling and heating operations. Because of the pressure independent operation and integrated thermostat, excellent room temperature control can be achieved.

DIRECT DIGITAL ELECTRONIC SYSTEMS - FIGURE 58C

Here again the power source is a low voltage supply. Signals from a pneumatic or electronic velocity sensor, together with signals from the room temperature sensor, are converted to digital impulses in the controller, which is a specialized microcomputer. The controller not only performs the reset and pressure independent volume control functions, but it also can be adjusted and programmed either locally or remotely for multiple control strategies, including scheduling. In addition, it can link to other controllers and interface with security, lighting, and other equipment. Control can be centralized in one computer.

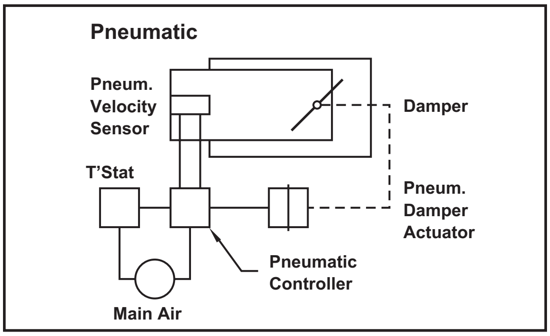

Figure 57. Pneumatic System

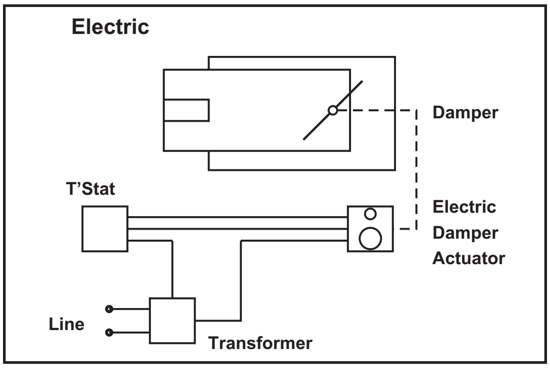

Figure 58A. Electric Pressure Dependent System

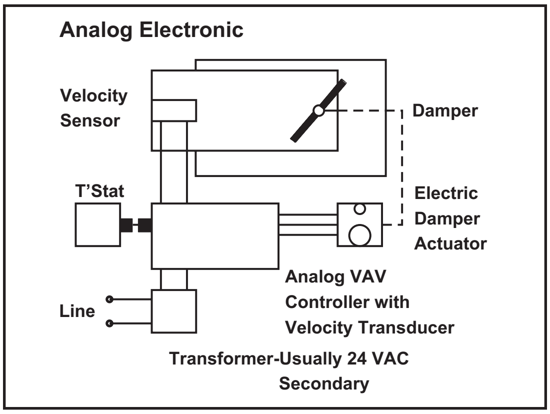

Figure 58B. Electric Pressure Independent System

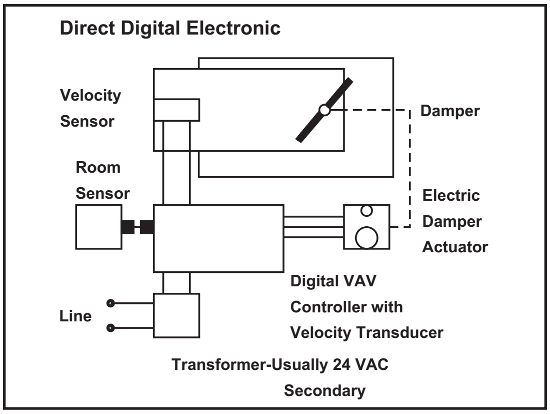

Figure 58C. Electric Pressure Independent System

CONTROL OPERATION IN TERMINALS

DAMPER OPERATION

Linearity (**Figure 61**) is the ideal characteristic for most damper applications. How nearly linear the operation is depends upon the percentage of the overall system pressure drop contributed by the wide open damper. Pressure independent control operations eliminate the effect of nonlinear dampers, but simulate the effect of a true linear damper to the system. For a linear damper characteristic, the damper is sized to contribute about 10% of the overall system resistance. Also (**Figure 62**), actuator torque must be sufficient to close the damper under all design conditions. In Titus terminals, the torque is always more than adequate.

DIRECT ACTING/REVERSE ACTING PNEUMATIC THERMOSTAT ACTION

In the direct acting pneumatic thermostat (**Figure 63**), a room temperature increase causes a corresponding increase in thermostat output. In the reverse acting thermostat (**Figure 64**), the sequence is the opposite. Because of these characteristics, direct acting thermostats are often used for cooling, reverse acting for heating. (With electronic systems, this term has no application.)

DIRECT RESET/REVERSE RESET PNEUMATIC VELOCITY CONTROLLER ACTION

In the direct reset pneumatic velocity controller (**Figure 65**), an increase in thermostat output pressure causes a corresponding increase in controller cfm setting. The damper will open and close to maintain this cfm when duct pressures change. In the reverse reset controller (**Figure 66**) the same action results from a decrease in controller cfm setting.

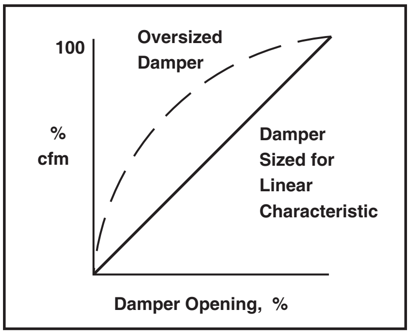

Figure 61. Linear Damper Operation

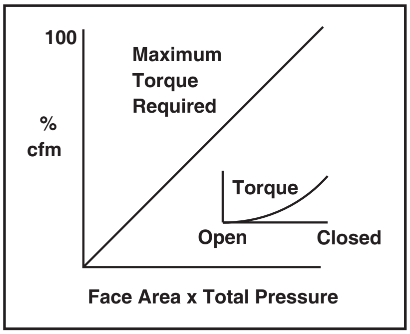

Figure 62. Damper Torque Requirement

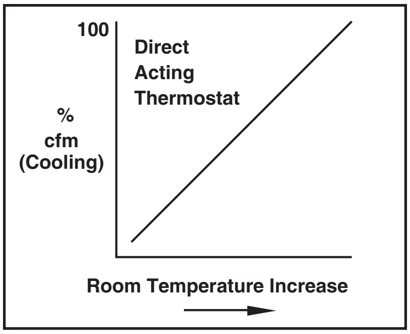

Figure 63. Direct Acting Thermostat Action

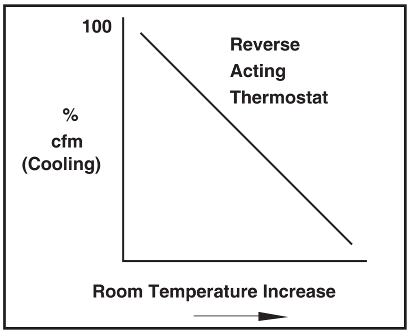

Figure 64. Reverse Acting Thermostat Action

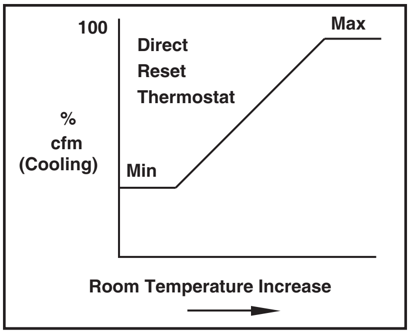

Figure 65. Direct Reset Pneumatic Velocity Controller

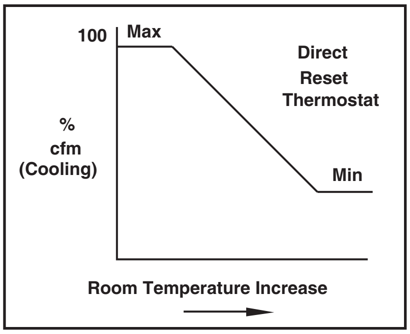

Figure 66. Reverse Reset Controller

PNEUMATIC THERMOSTAT–CONTROLLER COMBINATIONS

For systems supplying cold air when a direct acting pneumatic thermostat signals a direct acting controller (**Figure 67**), an increase in room temperature produces an increase in cfm 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 (**Figure 68**) will produce a decrease in cfm as the room temperature increases. With warm supply air, the logic is reversed.

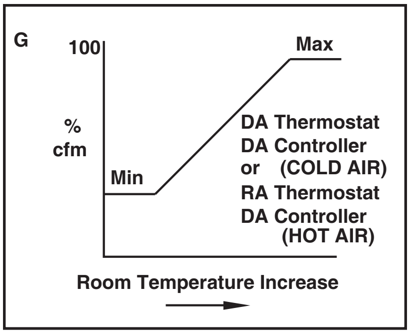

Figure 67. DA Pneumatic Thermostat Signaling DA Controller Combination

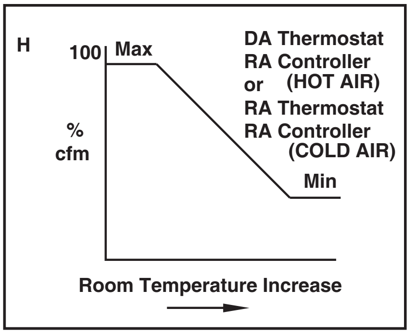

Figure 68. RA Thermostat with Reverse Reset Controller or RA Thermostat with Direct Reset Controller Combination

ACTUATOR TERMINOLOGY

Pneumatic actuators have an internal spring which is overcome by control air pressure. When air pressure is less than the spring tension, the actuator will retract. Depending on how it is connected to a damper, the damper may open or close on increase in control signal. Electronic actuators, however, are typically “fail stopped” unless they have a return spring which is activated by a loss of control signal. These are several times the cost of “fail stopped” actuators. When normally open or normally closed actuators are specified in an electronic control project, the requirement is most often in error.

NORMALLY OPEN

This describes a pneumatic operator which is configured so that on loss of air pressure the damper in the unit will open fully. These applications are typically ones where all like units are desired to be open for control purposes such as smoke removal or to prevent excessive pressure on system start-up.

NORMALLY CLOSED

When air pressure is removed, the actuator will cause the damper in the unit to go fully closed. This is typically specified when an area is to be isolated.

PNEUMATIC CONTROL / ACTUATOR COMBINATIONS

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

VELOCITY CONTROLLER OPERATION

DEFINITIONS OF TERMS

The controller setpoint is the cfm setting that the control system is calling for at any given moment. At that setpoint the damper opening may vary widely to compensate for any duct pressure changes reported by the inlet sensor, and thus hold the cfm constant.

With pneumatic systems, the setpoint, 11 psi in the example (**Figure 69**), can be reset by the action of the thermostat anywhere between the maximum and minimum cfm settings of the controller. The corresponding thermostat output pressures are called the start and stop points. The range of possible setpoints between the start and stop points is called the reset span, 8 to 13 psi in the example shown here.

The thermostat may also control an auxiliary piece of equipment, such as a proportioning valve on a hot water coil, shown here 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 or total throttling range, 3 to 13 psi in this example.

THERMOSTAT SENSITIVITY

This is the change in output signal caused by a change in room temperature. This rating (**Figure 70**) is usually 1°F = 2.5 psi for pneumatic systems. Electronic systems have a wide variance in output responses.

HYSTERESIS

This is the failure of an object to return to its original position after a force has moved or deflected it. For example, in some velocity controllers (**Figure 71**) the cfm setting increases along the lower curved line and decreases along the upper curved line. At the setpoint, the cfm may be either A or B.

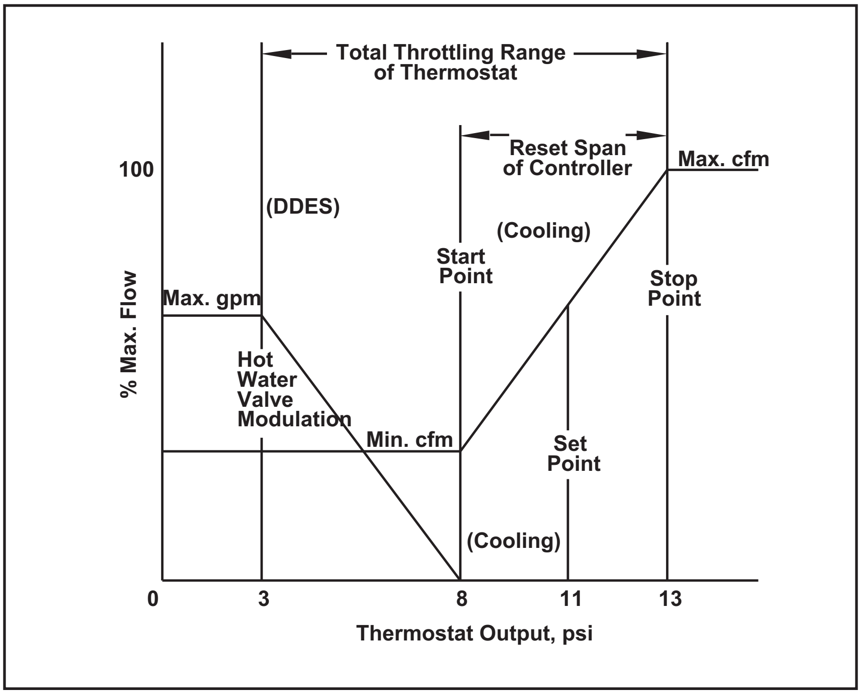

Figure 69. Set Point Example

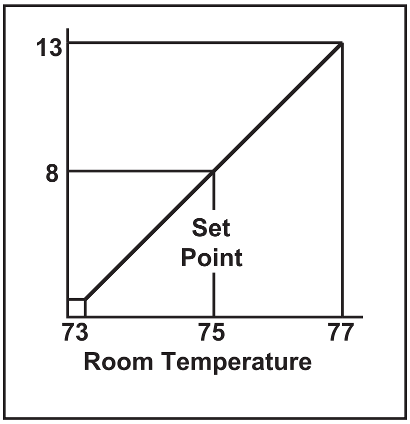

Figure 70. Thermostat Sensitivity Example

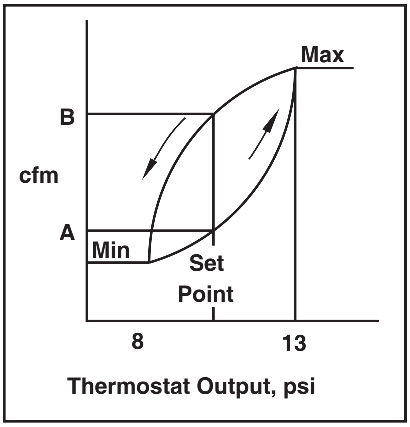

Figure 71. Hysteresis Example

PNEUMATIC FEEDBACK

Signals from the thermostat determine the cfm setpoint of the controller. The duct velocity acting on the velocity sensor forms a feedback (closed) loop (**Figure 72**) that allows the controller to monitor the airflow resulting from its settings and make corrections continuously. This is a form of closed loop control and is used on both pneumatic and electronic pressure independent systems.

In the Titus II pneumatic controller there is also an internal feedback loop that works in conjunction with a positive positioning reset mechanism to eliminate hysteresis (**Figure 71, page B47**).

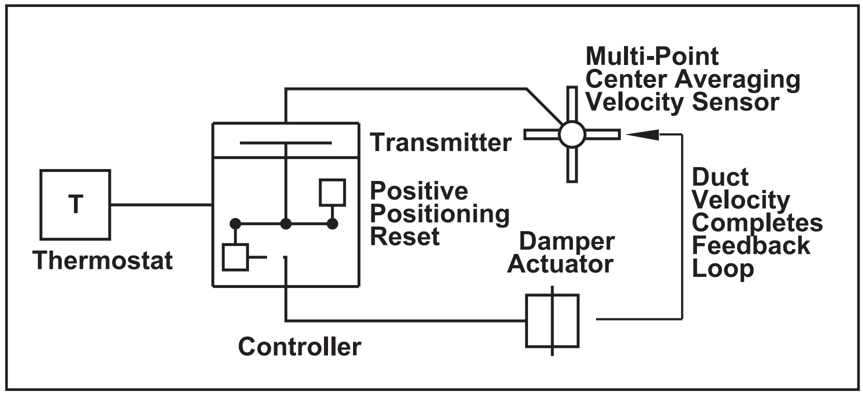

Figure 72. Pneumatics Feedback

FAN TERMINAL FLOW CONTROL

Engineers designing air systems try to match the airflow capacity of fan powered terminals to the needs of the space. Exact matches are rare, however. The design may not allow an exact match, a product other than the one which is the subject of the design might be selected, or system balancing might require a different airflow to meet field conditions. The two commonly used methods of trimming fan airflow are:

- Mechanical Trimming
- Voltage Adjustment

SERIES FAN SHIFT

With Series fan terminals, the fan output is intended to remain constant over a range of primary inlet damper flow rates. With proper design, this is normally so. With improper design, or with additional inlet attenuators added to a terminal, the fan may see a different external pressure when in full induction mode than when in full cooling. This results in a variation in the quantity of air delivered to the space, or "Fan Shift." The consequences of fan shift depend on individual zone characteristics and building design. If diffusers are selected such that they may add background masking sound at design flow, variations in flow may be an annoyance to the occupants. If a designed ventilation rate is assumed, this may vary if fan shift happens. (Titus terminals are designed to minimize fan shift.)

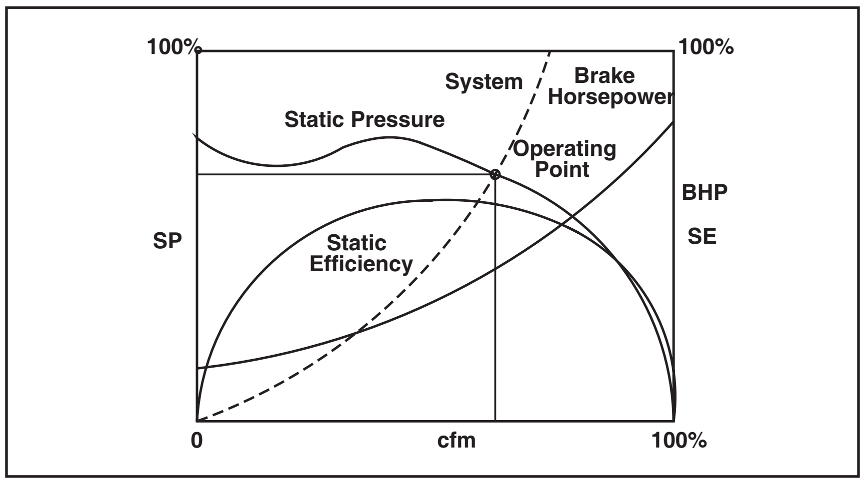

Figure 73. Forward Curved Fan Performance Curve

MECHANICAL TRIMMING

Mechanical trimming involves the use of a mechanical device, such as a damper, to adjust the fan airflow to meet the design requirements. Typically, these are used in conjunction with a multi-tap motor to provide a greater operating range and keep the energy consumption and sound levels as low as possible. Mechanical trimming offers a lower first cost versus a voltage adjustment, but at increased operating costs and increased sound. Multi-tap motors are not always effective in changing flow.

In operation, the mechanical device will raise the static pressure the fan operates against by either restricting the free area downstream of the fan or restricting the free flow of air drawn into the fan. A forward curved fan riding the fan curve will reduce airflow accordingly (**Figure 73**).

Although the rpm of the fan will increase, less work will be performed. This will result in a reduction of the amp draw of the fan motor. Since voltage remains constant, the overall power consumption of the fan is reduced. The power reduction from mechanical trimming is less, however, than the power reduction from voltage adjustment. When mechanical trimming is used, the sound levels of the fan terminal will increase. When the dampering occurs downstream of the fan, the velocity of the discharge air must rise, thereby increasing the discharge sound power levels. Additional sound contributions are made by the fan. The increased rpm of the fan results in greater tip speed. This occurs with either dampering method, raising the level of both the radiated and discharge sound.

VOLTAGE ADJUSTMENT

Voltage adjustment of fan powered terminals typically involves the use of a silicon controlled rectifier (SCR). An SCR uses a triac to phase proportion (chop) the electrical sine wave.

In effect, the SCR switches power off 120 times a second on a 60 Hertz cycle. This reduces the voltage to the motor, slowing its speed. In operation, the SCR responds to the current but controls voltage. Thus, while an SCR's triac may be energized at zero current, the current sine wave generally lags the voltage sine wave with an induction motor. This results in the idealized voltage sine wave (**Figure 74**). As the SCR is used to further reduce fan speed, the true RMS value of the voltage is reduced.

As voltage to the motor is reduced, the motor tries to compensate and the motor's amp draw rises slightly. The amperes will continue to increase until 50% of the current sine wave is phase proportioned. After this point, the amp draw will decrease. The increased amp draw is small relative to the reduction in voltage. As a result, comparing power consumption of the mechanical trimming method with the voltage adjustment method is analogous to comparing the power consumption of inlet guide vanes on central air handlers with speed inverters (**Figures 75 and 76**).

FAN SPEED CONTROL

The rpm of the motor is reduced by the SCR, lowering the tip speed of the fan. Since the free area downstream of the fan is not reduced, the velocity either meets design conditions or is lowered if the airflow is reduced below design for balancing purposes. There is no increase in sound from air disturbances.

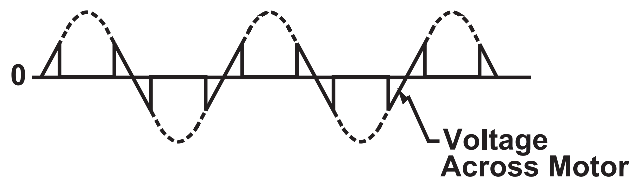

Figure 74. Idealized Voltage Sine Wave Resulting from an SCR

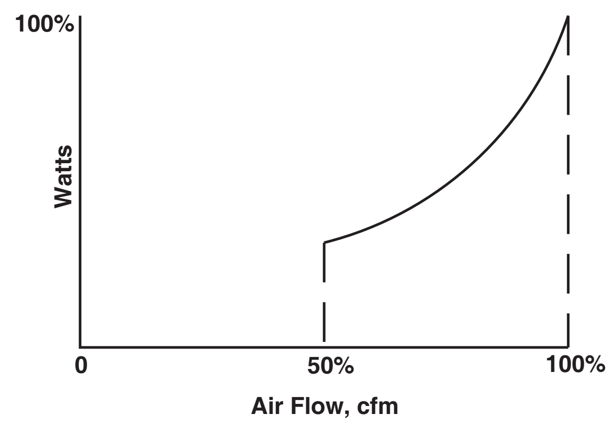

Figure 75. Watt Reduction Versus cfm

A Note on Nameplate Ratings
The amp draw can increase above the nameplate rating of the motor! The motor's nameplate specifies the amp draw for one set of design conditions. Since the voltage to the motor is reduced, the nameplate rating is no longer applicable. If proper care is taken in the design, specification and selection of the motor by the terminal manufacturer, the increased amp draw will pose absolutely no problem in operation or longevity. Thousands of fan powered terminals shipped with SCRs over the years serve as confirmation.

Titus accounts for the increased amp draw in the specification and selection of motors used for fan powered terminals. As a result, Titus specifies unit fusing adequate to handle the maximum amp draw possible under all operating conditions. This differs from the motor nameplate; it is essential that electric circuit fuses/overcurrent protection are sized according to the nameplate of the terminal, not the motor nameplate.

CATALOG FAN CURVES

The fan curves in a catalog represent the operating range of the fan powered terminal. Fan operation is dependent on the static pressure on the fan, so fan curves show airflow vs. static pressure. As the static pressure increases, airflow decreases. A typical fan curve will show maximum and minimum airflow for a fan powered terminal.

In (**Figure 77**), the top curve represents the maximum airflow that the fan and motor can provide. This corresponds to the recommended maximum operating rpm of the motor. The bottom curve shows the minimum airflow that the fan and motor can provide. This corresponds to either the minimum operating rpm of the motor or the minimum voltage of the SCR fan speed controller.

The SCR minimum is designed to protect the motor from operating below its recommended rpm. Most standard fan powered terminal motors must operate above a manufacturer's specified rpm to effectively self-lubricate.

However, the relationship between rpm and SCR voltage is dependent of static pressure. At minimum voltage on the SCR, the motor rpm will be different at different static pressures. Because of this, there is a possibility that at minimum SCR voltage, the rpm will be below the motor minimum recommended operating rpm. When this happens, the cataloged fan curve will use minimum rpm to set the minimum fan curve, not minimum SCR voltage.

To ensure proper motor operation, always operate a fan powered terminal with the cataloged fan curve.

A Note on Meter Usage
Many Digital Multi-Meters (DMMs) will provide erroneous readings when attempting to measure current or voltage near an SCR. These meters are designed for normal, smooth sine waves. The SCR, by changing the shape of the sine wave, throws off the readings from these meters. To measure the current voltage, a true RMS DMM designed for these conditions must be used.

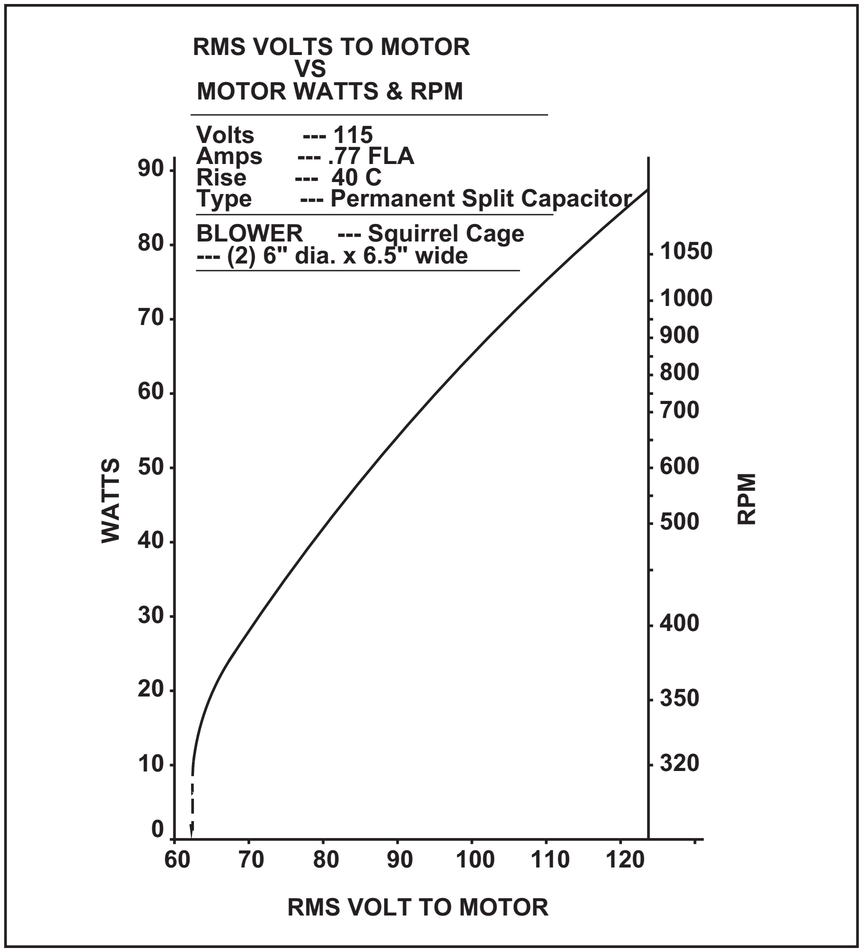

Figure 76. Watt, Volt and rpm Relationships

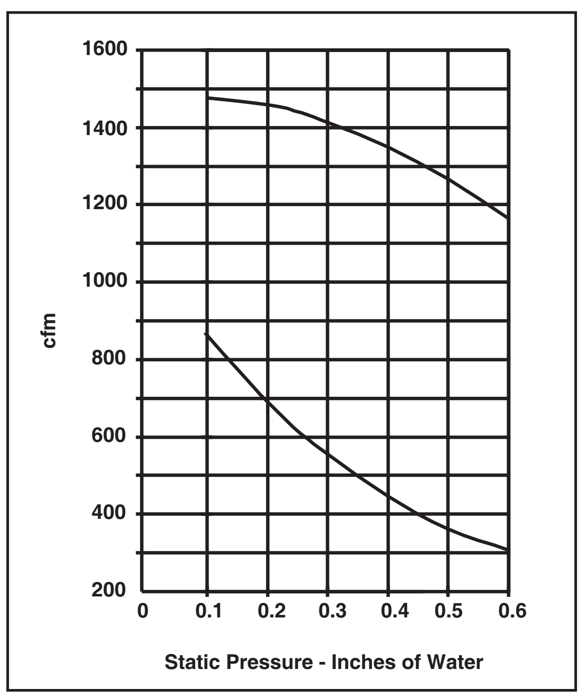

Figure 77. Typical Fan Curve

PRESSURE INDEPENDENT – ENERGY EFFICIENT ANALOG SPEED SETTINGS

ECM MOTOR TECHNOLOGY

The ECM motor is an ultra-high efficiency, brushless DC motor with a unique microprocessor based motor controller. Motor efficiencies of 70% or better across the entire operating range of the motor saves considerable electrical energy when compared to conventional induction motors. The motor controller, when tuned to the fan powered terminal, provides a large turn down ratio and constant volume airflow regardless of changes in downstream static pressure operating against the fan. With the introduction of the ECM motor, factory setting of the fan cfm is now possible.

Separate controls are required to enable field adjustment of fan speed. The fan speed control allows adjustments to be made three ways.

- Manually with a screwdriver, similar to the SCR control
- Remotely (as an option) through the DDC controls using a laptop at the unit
- Remotely through the Building Management System

HARMONICS

Power for a given motor is drawn through the line in the form of a pure sine wave. This sine wave contains a fundamental frequency, in the US typically 60 Hz. When there exists other pure sine waves, each with individual frequencies, other than the fundamental frequency, they are called harmonics.These waves cause distortion or “noise” in the power line. Therefore, harmonic distortion is a collection of pure sine waves, including the 60 Hz fundamental frequency, which when summed together point by point in time creates distortion in the incoming line.

Due to the way a standard split capacitor motor draws power, they have slightly fewer harmonic frequencies as compared to the ECM motor. The ECM motor, unlike the standard split capacitor motor, draws peak power only when needed, resulting in less electrical noise generation.

As of 2011, the most stringent of limitations for harmonics is published in the CAN/CSA - CEI/IEC 61000-4-3-07 (R2011). These values set the ceiling for allowable harmonic levels. The critical maximum or peak amp values for a given harmonic level occur in the third harmonic closely followed by that of the fifth harmonic. Published data for a 1hp ECM without filtering capability violates these CEI limits. Titus has developed technology to decrease the harmonic frequencies while continuing to deliver peak power as it is requested. The Titus ECM motor meets the criteria, as well as specified national and international harmonic limitations.

ENERGY SAVINGS POTENTIAL

The ECM motor, as applied to the Titus TQS fan powered terminal, offers significant energy savings over time to the owner when compared to conventional induction motors. Titus has evaluated an actual field trial and confirmed through bench testing an example of the potential energy savings when using the ECM motor. The following charts show the watt reduction associated with the ½ hp and 1 hp ECM motor when compared to standard TQS units of equivalent application range.

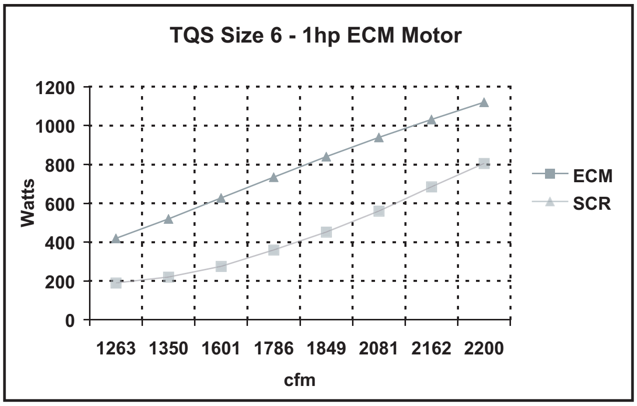

Figure 78. Watt Reduction with ½ hp ECM Motor

Note: TQS Size 6 with 1 hp ECM motor watt comparison to standard permanent split capacitor motor. The average watt reduction over the above range is 335 watts.

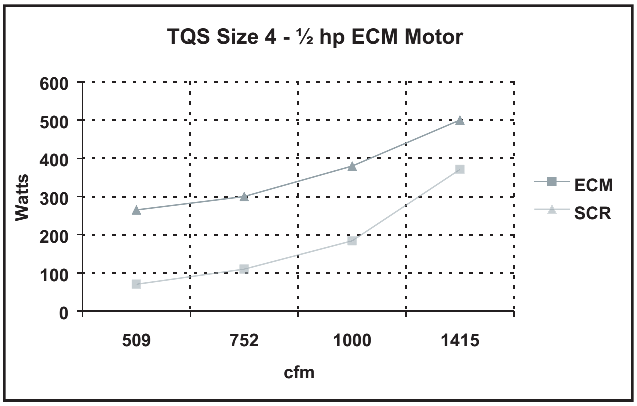

Figure 79. Watt Reduction with 1 hp ECM Motor

Note: TQS Size 4 with ½ hp ECM motor kW comparison to standard permanent split capacitor motor. The average watt reduction over the above range is 178 watts.

When evaluating this reduction in watts for energy usage the following table shows, at various usage rates, the annual savings per motor. Annual savings assume a run time of 3000 hours per year (250 days at 12 hours/day).

Table 9. Annual Savings per Motor

Usage	KW/hr reductions		
Rate	0.28	0.35	0.40
$0.05	$43.08	$52.50	$60.75
$0.06	$51.70	$63.00	$72.90
$0.07	$60.31	$73.50	$85.05
$0.08	$68.93	$84.00	$97.20
$0.10	$86.16	$105.00	$121.50
$0.12	$103.39	$126.00	$145.80
$0.14	$120.62	$147.00	$170.10

Also, reduction in demand charges must also be considered. Typically, demand charges are calculated during a 15-minute peak window. Some utilities will qualify the peak demand to only the summer months and use this peak as the monthly charge throughout the remainder of the year while other utilities will calculate demand charges using that months peak kW requirement. The savings associated with reduced demand charges are substantial, as demand charges are usually several dollars per kW. As an example, a typical multi-story office application may require 200 fan terminals.

Each fan terminal equipped with an ECM motor may have approximately 0.4 kW reduction in power. This translates to an 80 kW reduction in demand and with a demand rate of $10.00 per kW equates to a potential $800 per month reduction in the demand charges. While this model is simplistic, it is indicative of the payback potential of the motor. Utilities will vary not only in price but also in calculation methods with contract kW's versus actual kW usage so actual savings must be calculated according to local market conditions.

Coupling the usage and demand savings associated with the ECM motors can provide a substantial savings throughout the life of the building.

DIRECT DIGITAL CONTROL

APPLYING COMPUTERS TO CONTROL

With many years of experience, design engineers have established the basic principles of temperature control for heating, ventilating, and air conditioning (HVAC) systems. These control strategies have been applied utilizing conventional pneumatic, electric or analog electronic devices.

Recent advances in micro-technology have made it possible to apply the power and precision of computers to HVAC control. Microprocessors, which cost less than ever before and offer superior computing power, are now suitable for application to individual air handlers, packaged heating/cooling units, VAV terminals or the entire HVAC system.

DIRECT DIGITAL CONTROL

Microprocessor-based controllers inherently perform direct digital control (DDC) and typically replace the conventional pneumatic or analog electronic controls. Digital controllers measure signals from sensors (input), process these signals in software (through the microprocessor), and initiate a corrective action to a controlled device (outputs) (Figure 80). A more technical definition is provided in the ASHRAE Applications Handbook.

ADVANTAGES OF DDC

DDC systems offer several potential advantages over conventional counterparts.

- DDC systems provide improved comfort and greater energy efficiency through precise and accurate control. Pneumatic and Analog systems utilizing proportional (P) control have the inherent characteristic of offset (**Figure 81**). Microprocessor based controls can eliminate offset by adding the integral (I) or reset action. Furthermore, addition of the derivative (D) action can result in a faster response and greater stability (**Figure 82**), but requires significant tuning.
- DDC systems require less maintenance than conventional systems. Since there are no moving parts, periodic preventive maintenance (PM) tasks such as calibration, lubrication, cleaning and adjustments are seldom required.
- Control strategies can be modified quickly and easily without the need to rewire, repipe or install additional components

A direct digital controller receives electronic signals from the sensors, converts the electronic signals to numbers and performs mathematical operations on these numbers inside the computer. The output from the computer takes the form of a number, and can be converted to a voltage or pneumatic signal to operate the actuator.

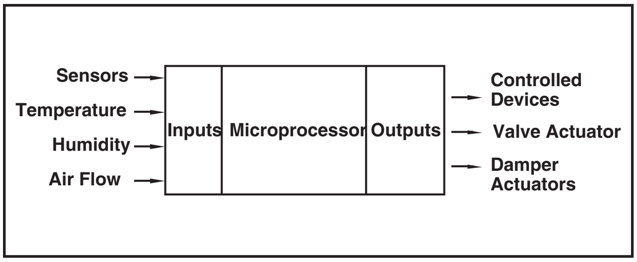

Figure 80. Direct Digital Controller

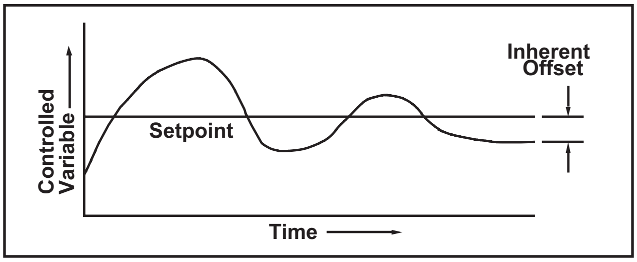

Figure 81. Inherent Offset - Lost Energy Dollars and Sacrificed Comfort

- Since microprocessor controllers are software based, multiple control sequences can be preprogrammed in memory thus allowing a single controller to be fully interchangeable between different equipment. For example, an application specific VAV controller may be used to control single duct, dual duct or fan powered terminals by simply choosing the appropriate operating sequence from a software library maintained on board every controller (**Figure 83**).
- While functioning completely independent, digital controllers perform all essential functions necessary to control different pieces of HVAC equipment without interconnecting to other computers. In this way each piece of HVAC equipment has its own digital controller in the same way conventional systems would provide individual control panels.

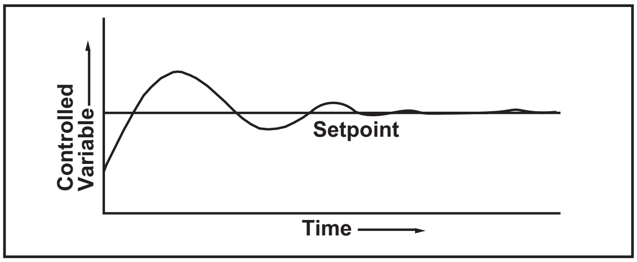

Figure 82. Offset Completely Eliminated - Improved Comfort and Less Energy Usage

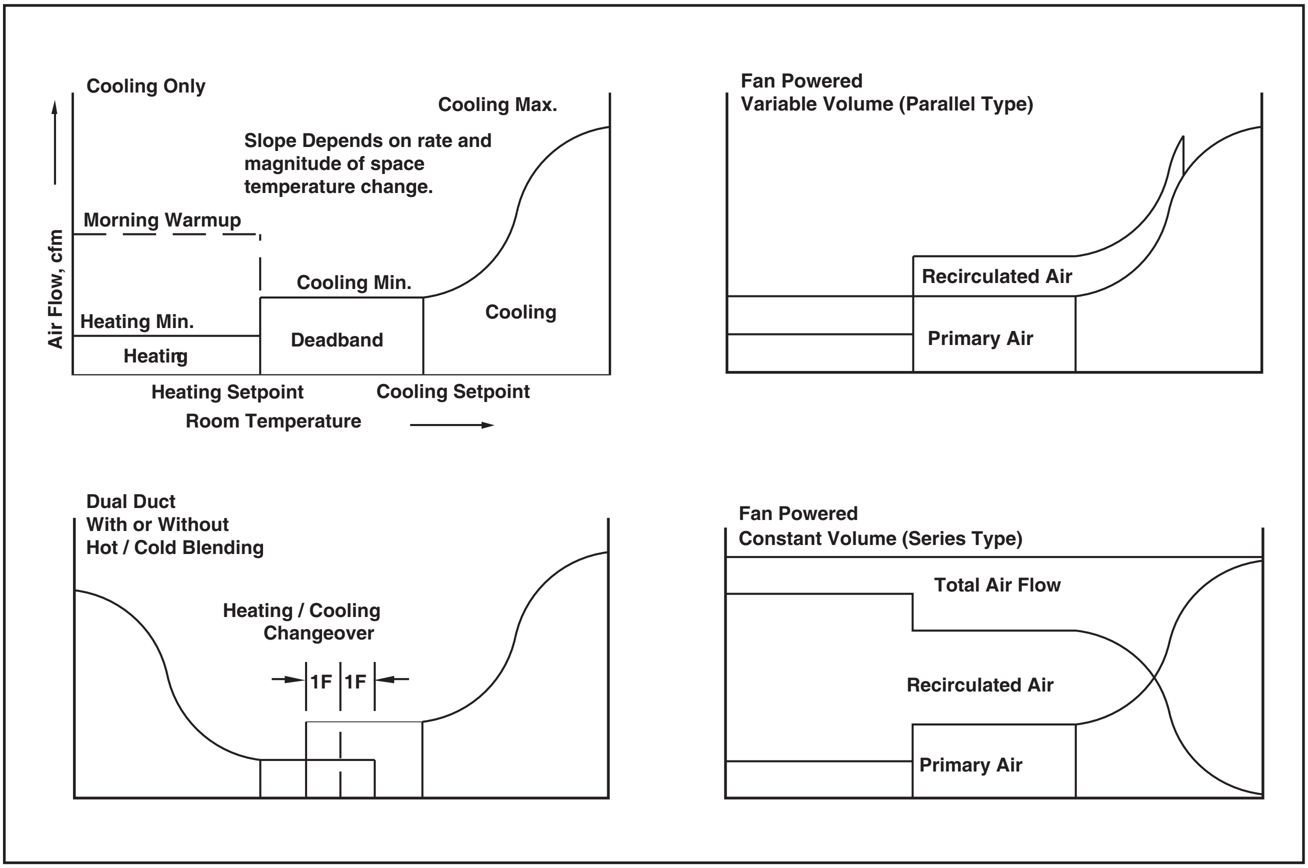

Figure 83. Frequently Used Control Sequences

DDC DISTRIBUTED PROCESSING

Using a concept commonly referred to as distributed processing, DDC controllers can function as standalone devices. In this way if one controller fails, others throughout the system can continue to function unaffected. The controllers are connected over a system communication bus or local area network (LAN) for system wide sharing of information. This information is used to perform sophisticated building control strategies not possible with conventional noncommunicating systems. The network also allows system access locally through a personal computer or remotely via modem over telephone lines (**Figure 84**).

SIZING BASIC TERMINALS FROM CAPACITY TABLES

CERTIFIED AIR TERMINALS

To provide engineers with sound power data which can be compared on an even basis, leading air terminal manufacturers joined together under the Air-Conditioning, Heating and Refrigeration Institute (AHRI) to develop an industry standard for rating air terminals and certifying performance data. The result was AHRI Standard 880, Air Terminals, and the 880 Certification Program. Standard 880 specifies the procedure, using a reverberant chamber, for developing sound power data. The certification program ensures manufacturers' equipment performance meets their claims.

Compliance with 880 is assured through third party testing. If a manufacturer fails to match claimed performance, the manufacturer must immediately rerate the terminal or lose the ability to use the AHRI Standard 880 seal. Another standard, AHRI Standard 885, was developed at the same time to assist the engineer in using certified product data.

Terminal selection involves a series of trade-offs. The designer needs to try to balance all of the constraining factors and select the terminal which meets overall needs best.

> Engineers who specify AHRI Certified air terminals are assured that the manufacturer's performance meets the manufacturer's claims. This is protection for the engineer, the building owner and the building occupant.

SIZING SINGLE DUCT TERMINALS

The starting point for sizing single duct terminals is to identify the type and model of controller. This is necessary because some controllers are more accurate at lower velocities than others.

Once the type of control is identified, the minimum and maximum primary airflows should be considered against the published cfm range. The trade-offs start here. Some engineers will select terminals near the bottom of the cfm range to reduce sound levels since large inlets reduce face velocity. Others select terminals near the top of the cfm range to hold down equipment costs. Still other engineers believe that one should remain comfortably in the middle to avoid potential control problems resulting from low velocities and sound problems occurring at high velocities.

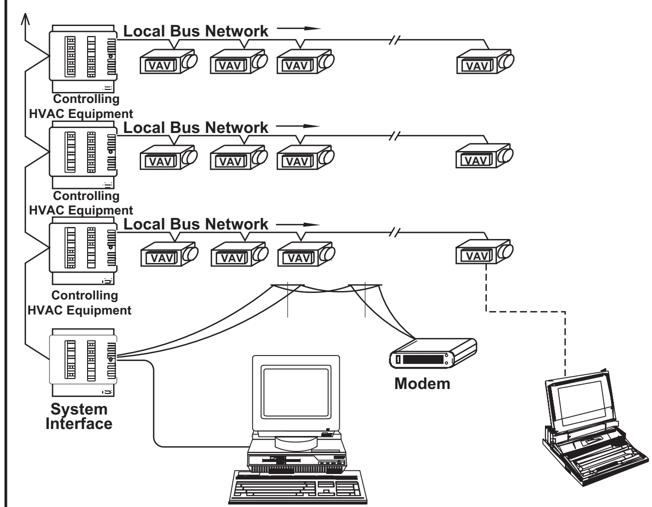

Figure 84. System Access via Network

All Titus products operate extremely well within the published cfm ranges. Therefore, low velocity control concerns can be eliminated. This leaves sound and first cost as the key issues. If the terminal is relatively small to begin with and will be located over a kitchen or hallway, sound will probably not be of concern and the designer may choose to slightly undersize the terminal. If, on the other hand, the terminal is located over office space, the designer may slightly oversize the terminal.

The selection of an appropriate water coil should also be considered at this time. In some cases, a terminal may need to be increased in size in order to obtain the desired heat output from the coil. With single duct units, the water coil air pressure drop should be subtracted from the duct pressure when determining sound generation. The sound produced by the damper is proportional to the pressure drop across the damper and discharge water coils may reduce that pressure drop. Other significant downstream pressure drops should be considered, and their pressure drop subtracted as well.

SIZING PARALLEL FAN POWERED TERMINALS

Parallel flow (variable volume) fan powered terminals are selected based on their capacity to handle the primary airflow. The same rules which apply to the selection of single duct terminals can be used, except that water coils are not in the primary airstream path, and will not affect sound levels. The pressure drop of the water coils, however, which are on the fan inlet in Titus parallel fan units, must be added to the expected discharge pressure at the fan flow rate when entering the fan curve tables.

The fan is selected based on the minimum airflow requirements for the space or the heating load required. In most cases the fan can be downsized from the cooling flow requirement considerably, reducing both first cost and operating cost. The fan is selected from the fan curves. The downstream static pressure of the secondary air may not be the same as the primary air, however. If the secondary airflow requirements are less than the primary air requirements, the static pressure will be reduced. The following equation can be used to determine the static pressure at reduced airflows. **(Do not forget to add water coil pressure drops to the fan requirement)**.

$$Ps_2 = Ps_1 (V_1 / V_2)^2$$

Where:
- Ps_1 = Primary Air Static Pressure
- Ps_2 = Secondary (Fan) Air Static Pressure
- V_1 = Primary Air Velocity
- V_2 = Secondary (Fan) Air Velocity

To select a Titus parallel fan powered terminal, refer to the published fan curves and primary air pressure drop curves, together with the application and sound power data.

In the parallel flow type of unit, when the primary air is ON, the fan is typically OFF, and vice versa. As shown in the (**Figure 86**), the primary air and the fan discharge air follow parallel paths into a common plenum. Therefore both airflows will encounter the same downstream resistance at a given flow rate.

Since the primary and secondary airflows come from two different sources-and often at two different specified flow rates-the volume vs. pressure relationship in each of these airflows must be checked to ensure adequate flow rates under actual job conditions.

Example: Select a Model DTQP for a maximum of 1400 cfm of primary air with 1.00” wg inlet static pressure. The fan airflow required is 1150 cfm. The downstream resistance offered by the duct and diffusers has been determined to be 0.30” static pressure at 1150 cfm.

Primary Air: From the chart on **page N75**, a size 4 with a 12” inlet will handle 1400 cfm of primary air with a minimum static pressure drop of 0.23” through the primary air section. But since the downstream resistance is 0.30” at 1150 cfm,

$$\left(\frac{1400}{1150}\right)^2 \times 0.30'' = 0.44'' \text{ sp}$$

The overall primary air static pressure drop is

$$0.23'' + 0.44'' = 0.67'' \text{ sp}$$

Since a 1.0” static pressure is available at the inlet, the selection will work.

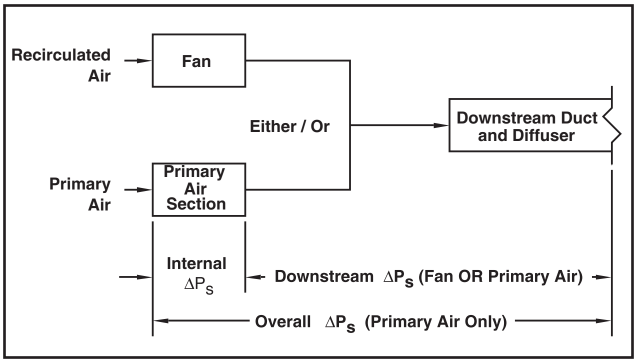

Figure 85. Schematic Diagram of Airflow in Parallel Flow (Variable Volume) Models

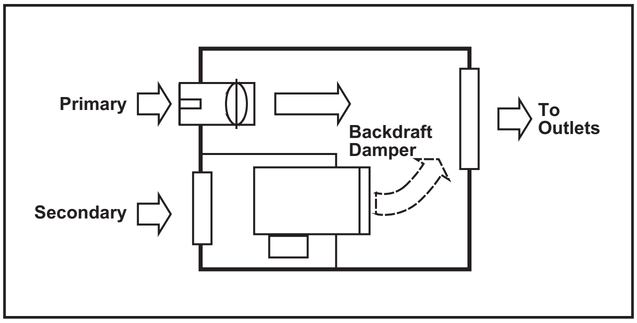

Figure 86. Actual Arrangement of Components Shown in the Previous Schematic Diagram

The damper in the primary air section will do some throttling to hold the maximum air flow to 1400 cfm.

Secondary Air (Fan): From the fan curves, a size 4, without coils, terminal will handle 1150 cfm at 0.30” static pressure, with the proper setting of the standard SCR speed control.

SIZING SERIES FAN POWERED TERMINALS

Compared to single duct terminals, series flow (constant volume) fan powered terminals add the additional factor of fan cfm requirements. The designer must consider both the primary airflow and the fan. Series terminals are selected based on the capacity of their fans. The secondary (or fan) cfm should be equal to or slightly more than the primary air to ensure primary air does not short circuit through the induced air port into the plenum, thereby wasting energy. Before selecting the fan, the static pressure downstream of the terminal must be determined. This is the resistance of the ducts and diffuser(s) at design airflow rates.

Once the downstream static pressure is known, the designer can select the fan based on the fan curves (these are shown throughout the catalog with the performance data for each fan powered terminal). The designer should find the intersection of the static pressure line on the horizontal axis and the fan cfm on the vertical axis. Selecting toward the upper end of the range will ensure that first costs are kept low and the fan motor efficiency is high. Selecting below the indicated minimum flow will result in shortened motor life as the bearings in the motor are centrifugally lubricated. If a water coil is needed, the designer must use the curves provided for a one or two row coil. These curves account for the additional static pressure generated by the coil. The static pressure added for an electric coil is negligible and may be disregarded. Neither has an appreciable effect on sound levels.

Inlet size must also be selected. Fan powered terminals come with varying inlet sizes. In general, inlets should be selected toward the bottom of the range. This reduces the face velocity of the inlet and minimizes the sound generated by the primary air valve.

To select a Titus series fan powered terminal unit, refer to the published fan curves and primary air pressure drop curves together with the application and sound power data. An abbreviated table is shown at the right for use with the example discussed here.

In the series flow type of unit, the fan runs continuously in the standard version. With the optional night shutdown and night setback controls, the fan can be cycled ON and OFF when the primary air is OFF.

As shown in the diagrams below, the primary air is drawn into the fan inlet along with secondary (recirculated) air from the room. The maximum primary airflow must always be equal to, or less than, the total airflow through the fan. When the primary air section reduces its airflow in response to a reduced demand for cooling, the fan makes up the difference by drawing more recirculated air from the room. As a result, the flow rate to the room is constant.

The primary air section discharges into the unit casing near the fan inlet, where the static pressure is slightly below atmospheric. For this reason, the available inlet pressure need only be enough to overcome the internal pressure drop through the primary air damper itself.

Example: Select a Model DTQS for a maximum of 1200 cfm of primary air at 0.50" wg inlet static pressure. The fan airflow is 1200 cfm. The downstream resistance offered by the duct and diffusers is 0.30" at 1200 cfm.

Primary Air: From the table on **page N21**, a size 4 will handle 1200 cfm of primary air with a minimum static pressure drop of .18" through the primary air section. Since 0.50" static pressure is available at the inlet, the selection will work.

Secondary Air (Fan): From the published fan curves, a size 4 terminal will handle 1200 cfm at 0.30" static pressure, with the proper setting of the standard SCR speed control.

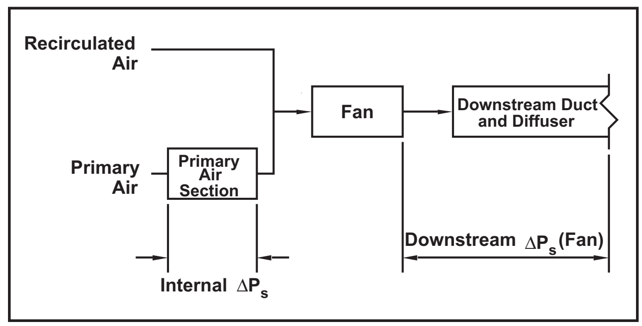

Figure 87. Schematic Diagram of Airflow in Constant Volume (Series Flow) Models

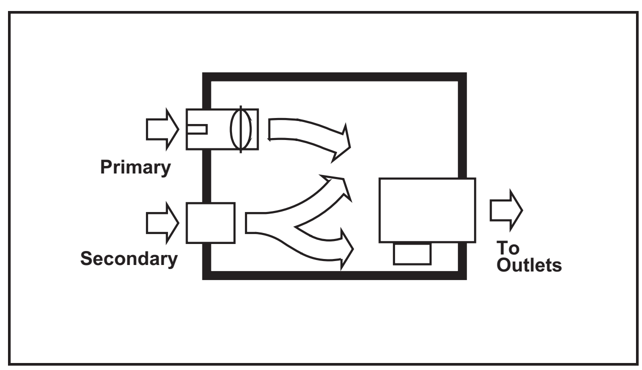

Figure 88. Actual Arrangement of Components Shown in the Previous Schematic Diagram

OVERSIZING TERMINAL

The direct result of oversizing is low air velocity. With the velocity too low, the damper must operate in a pinched-down condition most of the time, making control difficult. The inlet velocity can also be too low for effective operation of the sensor and controller. Too low a velocity through an electric heater will cause the safety airflow switch to shut down the heater. Oversizing fan terminals results in low fan motor rpm and the potential for under-lubrication of the motor bearings, resulting in shortened motor life and additional sound from larger motors (**Figure 89**).

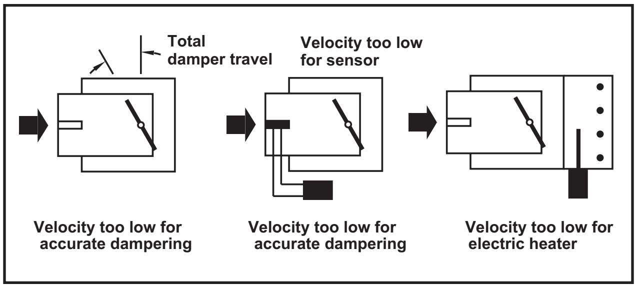

Figure 89. Low Velocity Effects

CAPACITY CONCENTRATED IN TOO FEW TERMINALS

When one large terminal serves a space that should be served by two or more smaller ones, comfort problems can result. There may be noticeable temperature differences between rooms, since the thermostat is located in just one room as at the right. Also, for a given air velocity, the larger the terminal the more sound power it generates (**Figure 90**).

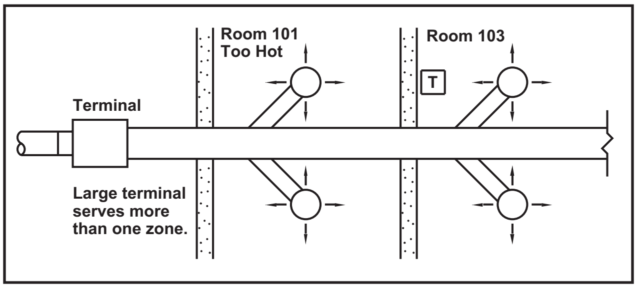

Figure 90. Too Few Terminals Effect

INSUFFICIENT SPACE

Carefully planning the locations of the terminals avoids problems with installation, performance, and maintenance. In the example shown at the right, the control side of the terminal is against the wall, making connections difficult and service almost impossible. The cramped location also creates the need for close-coupled duct elbows, which reduce performance (**Figure 91**).

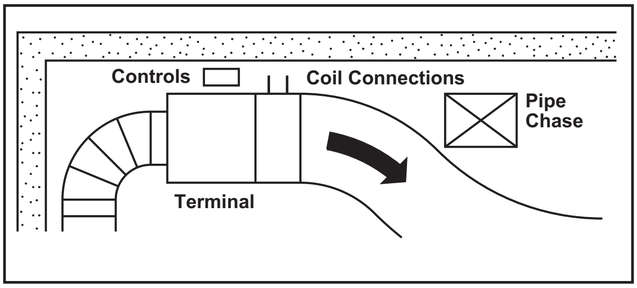

Figure 91. Installation Affecting Performance

IMPROPER DISCHARGE CONDITIONS

The duct connections at the discharge end of the terminal have a major effect on pressure drop. A tee close to the discharge is especially to be avoided, along with transition pieces and elbows. Another common error is running too much flex duct, as at the right. It would have been better to continue the rectangular duct to the last diffuser, then install short flex branches (**Figure 92**).

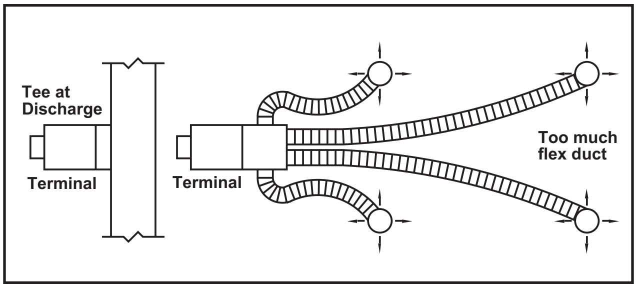

Figure 92. Improper Discharge Conditions

IMPROPER INLET CONDITIONS

The arrangement of duct at the terminal inlet affects both pressure drop and control accuracy.

The conditions shown at the right will create turbulence at the inlet. This makes it difficult for the sensor to measure airflow accurately. Although Titus velocity sensors correct for a considerable amount of turbulence, the best practice is to use straight duct at the inlet the same size or larger than the inlet (**Figure 93**).

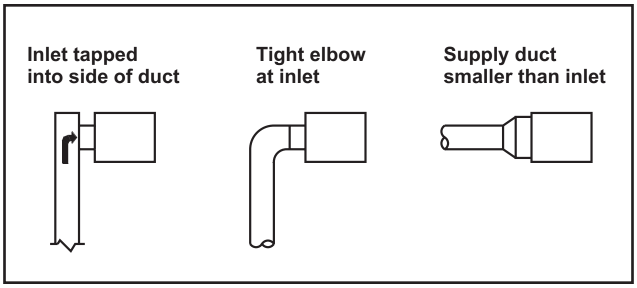

Figure 93. Improper Inlet Conditions

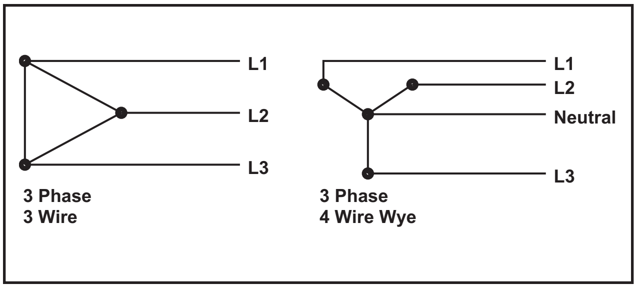

Figure 94. Power Source Compatibility

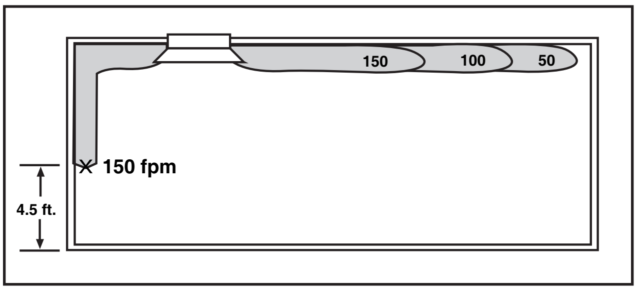

Figure 95. Overhead Heated Air

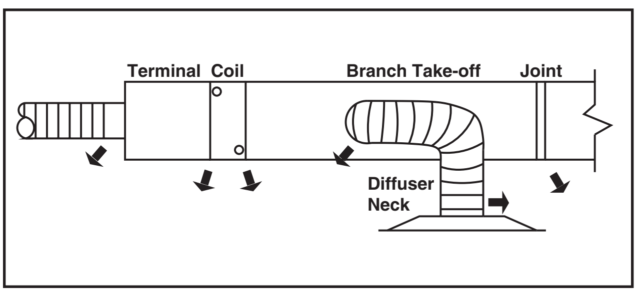

Figure 96. Possible Air Leakage

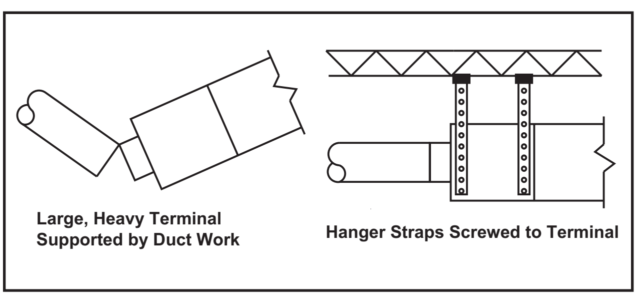

Figure 97. Terminal Support

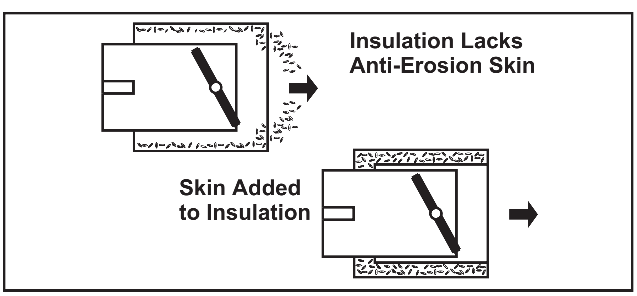

Figure 98. Anti-Erosion Skin Effects

INCOMPATIBILITY WITH POWER SOURCE

In fan powered terminals, electrically or electronically controlled terminals, and all terminals with electric heating coils, the order to the factory should be carefully checked against the electrical characteristics of the power source at the point of connection.

Not only must the voltage, phase, and frequency match, but the distinction between 3 phase-3 wire and 4 wire wye must be observed (**Figure 94**).

EXCESSIVE AIR TEMPERATURE RISE AND AIR CHANGE EFFECTIVENESS

The discharge temperature for terminal units should be selected so the maximum temperature difference between the room and the diffuser discharge is no greater than 15°F. This can be found in the ASHRAE Handbook of Fundamentals. According to ASHRAE 62.1, ceiling diffusers which have ceiling returns and are used for heating, should be mounted as shown to allow the supply air jet of 150 fpm to come down the exposed wall to within 4.5 ft. of the floor level. This reduces the short circuiting of warm air at the ceiling level and can be used to achieve an E_z air change effectiveness value of 1.0 as determined in ASHRAE Standard 129 for all air distribution configurations except unidirectional flow (**Figure 95**).

EXCESSIVE AIR LEAKAGE

Leakage from the branch duct upstream and downstream from the terminal, as well as from the terminal itself, can be serious. In some installations it is found to be as much as 10% or more of the total airflow.

Most of this leakage can be avoided by careful fabrication and installation and the use of top quality terminals (**Figure 96**).

IMPROPER SUPPORT OF TERMINAL

Many terminals are light enough to need no support other than the duct work itself. However, the larger sizes, units with electric coils and fan powered models, are heavy enough to require additional support. A practical method is to use hanger straps screwed to the sides of the terminal. The bottom should be left clear where there are access panels (**Figure 97**).

WRONG TYPE OF INSULATION

Installations in hospitals, clean rooms, and laboratories often require a special insulation liner to prevent air erosion or microbial growth. In the past, Mylar and Tedlar were often specified in these installations. Neither, however, meet current safety codes in many cities. Foil faced insulations, such as foil-faced EcoShield and Steri-Loc, provide the required covering, meet all safety codes and actually provide some sound attenuation. Titus Fibre-Free insulation provides both sound attenuation and resistance to erosion and mold growth (**Figure 98**).

NON-COMPLIANCE WITH LOCAL CODES

Some localities have stringent codes of their own, with requirements beyond those of NEC, UL, and CSA. An example is the primary fusing in the control transformer at the right (**Figure 99**).

INSTALLATION TECHNIQUES–DUCT CONNECTIONS

The inlet duct slips over the inlet collar of the terminal. It should be fastened and sealed according to the job specifications.

The diameter of the inlet duct must be equal to the listed size of the terminal. For example, a duct that measures 8″ in diameter must be fitted to a size 8 terminal. The inlet collar of the terminal is made 1/8″ smaller than nominal size in order to fit inside the duct (**Figure 100**).

Note: A duct should never be inserted inside the inlet collar of the terminal.

For optimum control accuracy, a straight section of unrestricted duct at least 1½ diameters long should be installed at the inlet (**Figure 101**). Where this condition does not exist, field adjustment of the airflow setting on the velocity controller may be required.

If space does not permit using the 1½ diameter length of straight duct, a hard duct elbow up to 90° can be installed at the inlet of the Titus terminal without altering the factory maximum or minimum airflow setting by more than 10% (**Figure 102**).

The outlet end of the Titus terminal is designed for a slip and drive connection. Unless a round duct adapter is furnished, a rectangular outlet duct should be fitted to match the size of the terminal casing. It should be fastened and sealed according to the job specifications.

If a round outlet adapter is furnished, it should be fastened and sealed by the same method used for the inlet. Close coupling the terminal inlet to the side of the main supply duct is not recommended. Where this condition is unavoidable, a flow straightening device (**Figure 103**) should be installed between the main supply duct and the inlet to the terminal. Even with the flow straightening device, the terminal may still require some field adjustment of the factory airflow settings at the velocity controller.

Air leakage adds significantly to the operating cost of an HVAC system. Important savings are realized by carefully fitting and sealing all duct joints and specifying tightly constructed Titus terminals. The Titus box has very low damper and casing loss leakage.

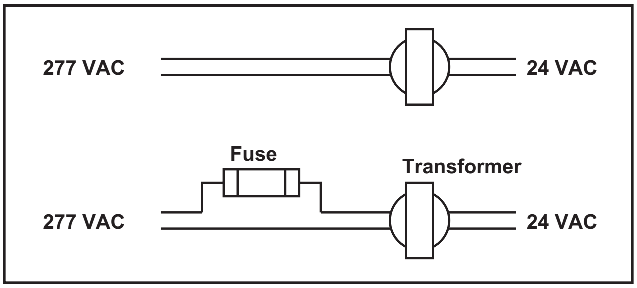

Figure 99. Primary Fusing in the Control Transformer

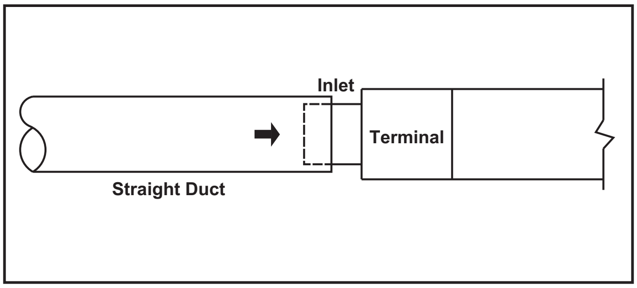

Figure 100. Terminal Inlet Collar Fitting Properly

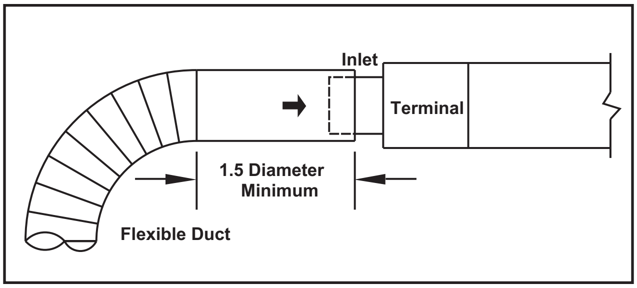

Figure 101. Unrestricted Duct Properly Install at the Inlet

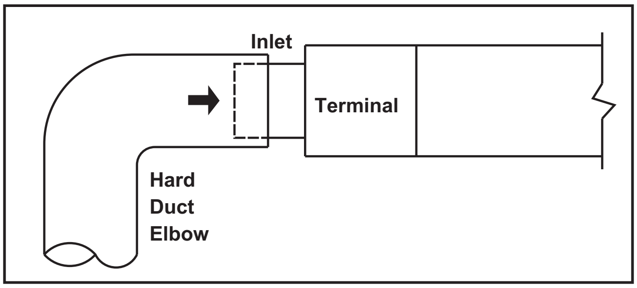

Figure 102. 90 Degree Hard Elbow Duct Installed to Inlet

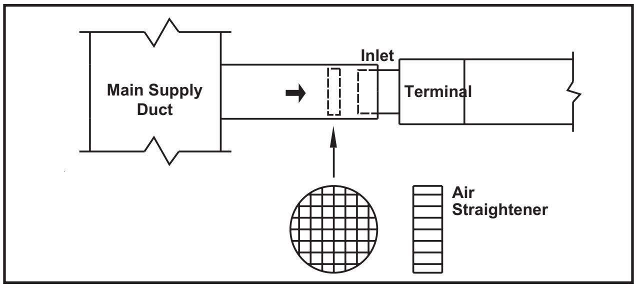

Figure 103. Flow Straightening Device Placement

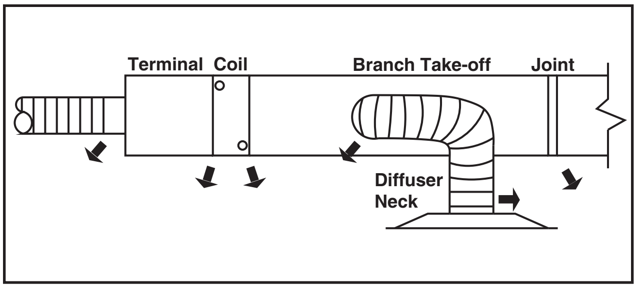

Figure 104. Possible Air Leakage

The example in (**Figure 104**) shows how many dollars can be lost in the leakage from just one terminal together with its connected duct work. Multiply that amount by the hundreds or thousands of terminals that may be in one building, and the seriousness of the loss is apparent.

This is a conservative example, in that the leakage is only 5%; a much higher percentage is found in many installations. Also, the compressor, pumps, and fans may not run as efficiently as indicated here, and the cost of electric power in many parts of the country is greater than $0.06 per kilowatt hour.

Example of Leakage Costs

A 10" terminal handles 1150 cfm. The central system cools air from 80°F dry bulb / 67°F wet bulb to 53°F dry bulb / 51.5°F wet bulb before sending it to the terminal.

Cost of Refrigeration

The total heat removed from the system is:

31.65 Btu/# at 80/67 minus 21.10 Btu/# at 53/51. = 10.45 Btu/# of dry air.

The amount of leakage given in this example is 58 cfm.

The loss of refrigeration energy through leakage is:

58 cfm x 10.45 Btu/# x 4.5 = 2727 Btu/hr.

Assuming a cooling system EER of 7.5 overall (reference ASHRAE Standard 90), in a space where the system operates 24 hours a day, 365 days a year, (worst case)
At a power cost of $0.06 per kwh, 0.3636 x $0.06 x 24 x 365 = $191.11

$$\frac{2727}{7.5 \times 1000} = .0.3636 \text{ kWh input}$$

If the system operates at 40% capacity, averaged over:one year, $191.11 x 0.40 = $76.44, the cost of wasted refrigeration power alone, again worst case, assuming continuous operation.

The amount of leakage in the branch duct serving the terminal, the connections to the terminal, the terminal itself, and the duct downstream from the terminal is 5% of the 1150 cfm being handled, or about 58 cfm.

Cost of Fan Operation

If the static pressure across the fan is 5" wg and the fan static efficiency averages 75%, the leakage converts to:

$$\frac{58 \text{ cfm} \times 5}{6356 \times 0.75} = .0.061 \text{ bhp}$$

Assuming that the motor efficiency multiplied by the power factor averages 0.80,

$$\frac{0.061 \times 746}{0.080 \times 1000} = .0.0569 \text{ kw}$$

0.0569 x $0.06 x 24 x 365 x 0.40 = $11.96, the cost of wasted fan power.

Combined cost equals:

$76.44	
$11.96	
$88.40	per year for one terminal.

PRESSURE MEASUREMENT

Three categories of pressure are connected with air handling:

1. Static pressure may be thought of as the pressure in a tire or storage tank. It is exerted in all directions equally.
2. Velocity pressure, as its name implies, is entirely a function of air velocity and its direction. It is the pressure you feel against your hand if you hold it outside the window of a moving car.
3. Total pressure is the sum of static pressure and velocity pressure. It and static pressure are the pressures actually sampled by velocity sensors in terminals and by commonly used measuring devices, as described next.

The interaction of static, velocity, and total pressures is illustrated by (**Figure 105**). The Pitot tube, which is used to measure velocities and pressures, is really a tube within a tube. The inner, or impact, tube senses both the velocity pressure and static pressure combined (total pressure). The outer tube, which communicates with the airstream through small holes in its wall, avoids the impact of the air movement and senses only static pressure.

The U-tube manometer, connected to both parts of the Pitot tube, has the effect of subtracting static pressure
from total pressure to give a reading of velocity pressure.

Once the velocity pressure is known,
the velocity can be calculated easily:

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

where V = Air Velocity
and
P_v = Velocity Pressure

Knowing both the velocity and the cross-sectional area of the duct, the flow rate is then:

cfm = Area x Velocity

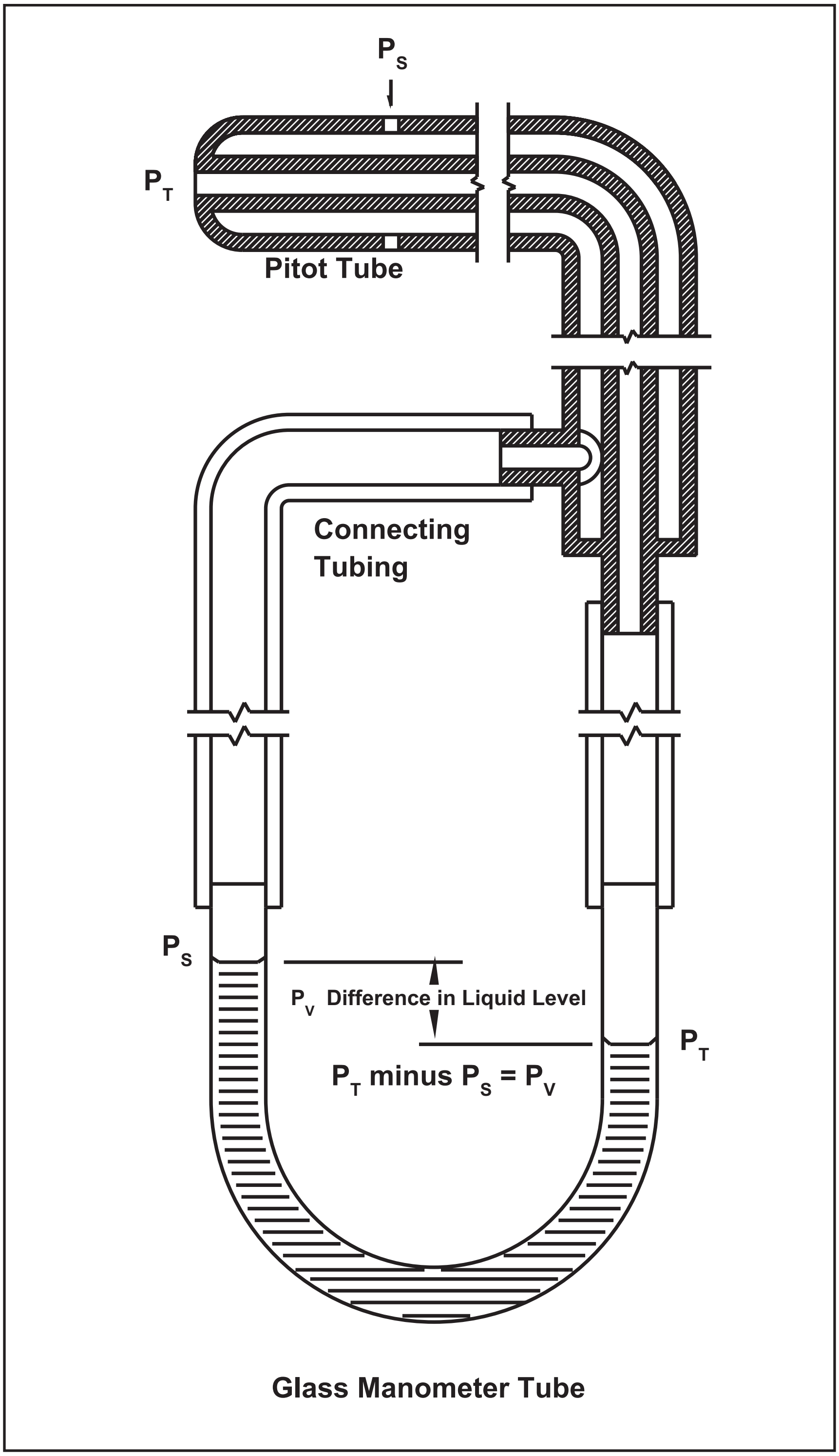

Figure 105. Static, Velocity and Total Pressures Interaction

The Fan Laws are basic tools in air handling. Three of the most common relationships are illustrated as follows.

Example:

A fan handles 40,000 cfm at 2” static pressure. It runs at 760 rpm and draws 18 brake horsepower. The fan is increased to 800 rpm. What are the new cfm, sp, and bhp?

1. Airflow rate varies directly with shaft speed.

$$\frac{cfm1}{cfm2} = \frac{rpm1}{rpm2}$$

$$cfm2 = (cfm1 \times rpm2) / rpm1$$
$$= (40{,}000 \times 800) / 760$$
$$= 42{,}105$$

2. Pressure varies as the square of shaft speed.

$$\frac{P_1}{P_2} = \left(\frac{rpm_1}{rpm_2}\right)^2$$

$$P_2 = P_1 \times \left(\frac{rpm_2}{rpm_1}\right)^2$$
$$= 2 \times \left(\frac{800}{760}\right)^2$$
$$= 2.22''$$

3. Horsepower varies as the cube of shaft speed.

$$\frac{bhp_1}{bhp_2} = \left(\frac{rpm_1}{rpm_2}\right)^2$$

$$bhp_2 = bhp_1 \left(\frac{rpm_2}{rpm_1}\right)^3$$
$$= 18 \left(\frac{800}{760}\right)^3$$
$$= 21.0$$

The relationships stated here apply when the air density remains constant and when there is no change in the fan or the system. They are based on Fan Laws 1, 2 and 3. For a complete presentation of the Fan Laws, see the ASHRAE Handbook, Systems and Equipment.

Each fan design has its characteristic set of performance curves. Those shown in (**Figure 109**) are typical of a centrifugal fan with forward curved blades in the wheel, as commonly used in fan powered terminals. For a full discussion of the characteristics of the various types of fans, see the ASHRAE Handbook, Systems and Equipment.

The solid curve represents a fan running at constant speed, as it is throttled from free delivery to close-off. The broken line square curve represents the pressure drop through the complete air handling system in which the fan operates. Intersection (A) is the operating point of the fan.

The dashed line represents another system pressure curve which intersects at point B. This point is a poor operation point as instability will likely reset.

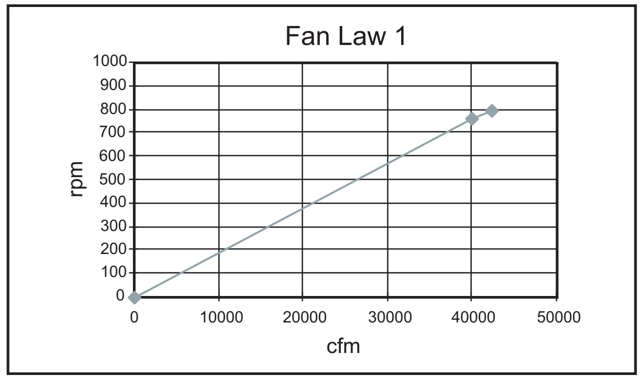

Figure 106. Fan Law - Airflow

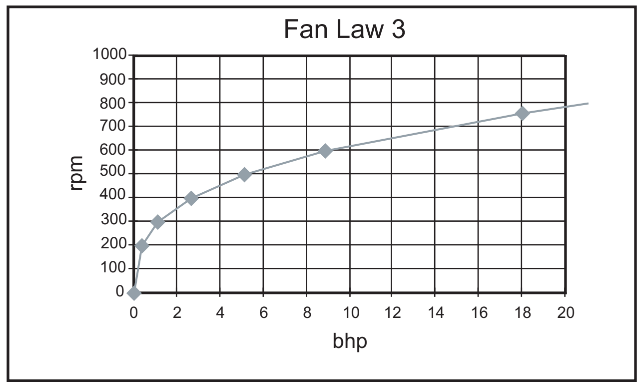

Figure 107. Fan Law - Pressure

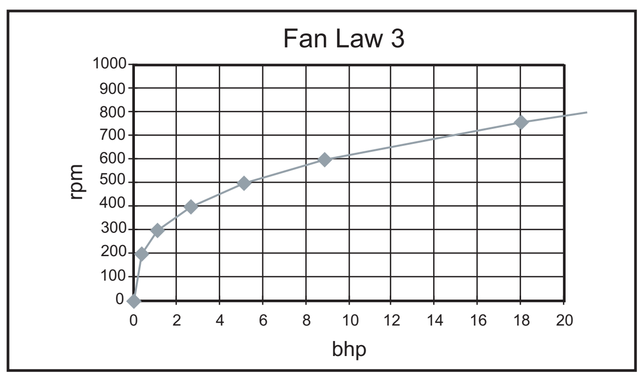

Figure 108. Fan Law - Brake Horsepower

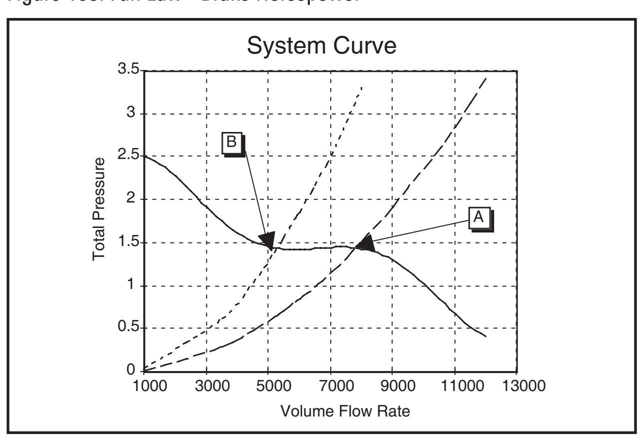

Figure 109. Centrifuge Fan Performance Curves

Table 10. Units of Measurements

Formulas and Definitions			Power		
VP	=	$(fpm / 4{,}005)^2$	W	=	Watts
(Q) cfm	=	Cubic Feet per Minute	A	=	Amps
TP	=	Total Pressure	hp	=	Horsepower
SP	=	Static Pressure	V	=	Volts
VP	=	Velocity Pressure	E1	=	Efficiency
(V) fpm	=	Feet per Minute	PF	=	Power Factor
ΔP	=	Differential Pressure			
ΔPs	=	Static Differential Pressure	Power DC Circuits		
ΔPT	=	Total Differential Pressure	W	=	V x A
(A) Area Factor	=	Dimension in Square Feet	A	=	W / V
VP	=	TP - SP	hp	=	V x A x E / 746
TP	=	SP + VP	E	=	746 x HP / W
SP	=	TP - VP			
cfm	=	fpm x Area Factor	Power AC Circuits (Single Phase)		
ΔPT	=	TP1 - TP2	PF	=	W / (V x A)
ΔPs	=	SP1 - SP2	A	=	746 x HP / (V x E x PF)
ΔP	=	$(cfm / K)^2$	E	=	746 x HP / (V x A x PF)
fpm	=	cfm / Area Factor	kW	=	V x A x PF / 1,000
K	=	$cfm/ \sqrt{(DP)}$	hp	=	V x A x E x PF / 746
Water Coils					
MBH	=	1,000s of Btus per Hour	Power AC Circuits (3 Phase)		
Btu	=	British Thermal Unit	PF	=	W / (V x A x 1.732)
gpm	=	Gallons per Minute	A	=	746 x HP / (1.732 x V x E x PF)
ΔT	=	Temperature Differential	E	=	746 x HP / (V x A x PF x 1.732)
Air ΔT	=	927 x MBH / cfm	kW	=	V x A x PF x 1.732 / 1,000
H20ΔT	=	2.04 x MBH / gpm	hp	=	V x A x 1.732 x E x PF / 746
Electric Coils					
kW	=	Kilowatts			
Air ΔT	=	Temperature Differential, Leaving Air - (minus) Entering Air Temperature			
kW	=	cfm x ΔT / 3,160			
ΔT	=	kW x 3160 / cfm			

REHEAT COILS:

Several types of terminal devices are available with reheat coils, both hot water and electric. When determining the heat requirement for a terminal, the engineer will often start with the known zone heating demand, typically expressed in BTUH, or more conveniently, MBH (thousands of Btu's). The room load requirements for heating are then used to determine the Room Entering Air temperature (EAT_r) by the equation:

$$\text{Btuh (room)} = 1.085 * (EAT_r - T_r) * Q$$

Where;

EAT_r = Temperature (°F) entering the room
T_r = Room setpoint temperature or average temperature
Q = Flowrate (cfm) (typically 30 - 50% of the cooling cfm)

By solving for the EAT_r, the coil Btuh requirements can then be determined. The room entering air temperature (EAT_r) now becomes the required LAT of the VAV box (ignoring any duct heat losses). The coil can now be sized according to:

$$\text{Btuh (coil)} = 1.085 * (LAT - EAT_c) * Q$$

Where;

LAT = Coil leaving air temperature
EAT_c = Coil entering air temperature (primary or mixed air)
Q = Flowrate (cfm)

Now that the coil requirements are known, published catalog data may be used to select the proper hot water or electric coil.