APPLICATION GUIDE

Variable Air Volume for Rooftop Units

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Johnson Controls

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Introduction

This application guide is an overview of variable air volume (VAV) for rooftop units (RTUs). The guide describes RTU VAV's structure, components, and uses. It also covers service and maintenance, refrigeration, and controls. It outlines how common and important standards apply to RTU VAVs and recommends particular installations and configurations for them. Properly applied, this is a guide to optimizing control of building air and temperature through RTU VAV.

Rooftop Unit (RTU)

An RTU is a single package unit containing all the components necessary for cooling or heating air in a building's occupied spaces. Commercial units range in size from small 5 ton units supplying a single office area of 2,000 square feet (sq. ft) to over 150 ton units supplying multiple floors of a building totaling 60,000 sq. ft.

RTUs are typically used in single story buildings or multistory buildings with four floors or less. Common applications include schools, places of worship, office buildings, restaurants, shopping centers, and big box retailers. For each of these buildings, RTUs can have various roles. They can provide cooling only, both cooling and heating, or contain a heat pump for limited sizes.

RTUs are roof mounted with the supply and return duct openings penetrating the roof. These openings can be arranged in a number of ways:

- **Vertical supply and return** – Directly below the unit
- **Horizontal discharge and return** – A location some distance away from the unit, with the ductwork running parallel to the roof and then down through the opening. Running the ductwork parallel to the roof requires the supply and return opening in the RTU to be on either the end or side of the unit.
- **Combination of vertical and horizontal** – The supply or return opening is directly below the unit (vertical supply and return) and the other is some distance away (horizontal discharge and return)

RTUs have two main parts to their structure:

- A condensing section comprised of compressors, condenser coils, and condenser fans
- An air handling section comprised of an evaporator coil, a supply air (SA) fan, a heat source, and air filters

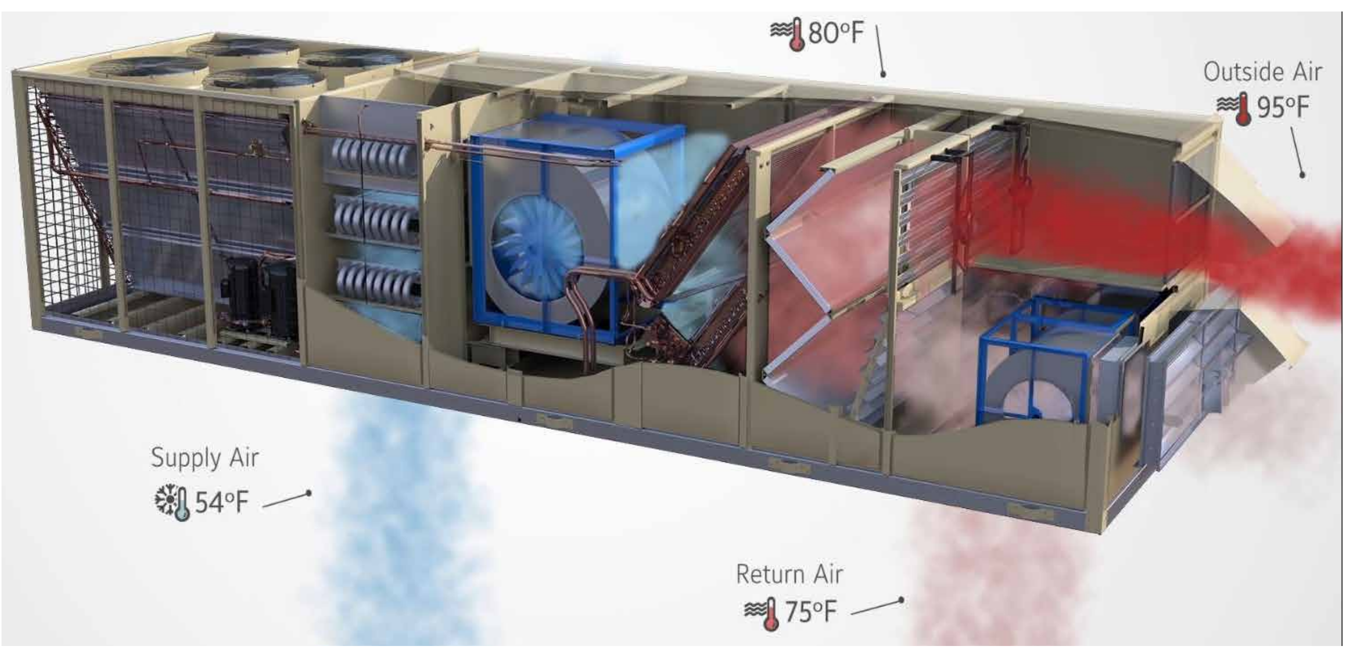

Figure 1: Rooftop Unit (RTU)

Many RTUs also include a wide range of optional equipment. Here are some of the more common options:

- An economizer that uses air from outside to cool the building during moderate weather
- An airflow monitoring station (AFMS) that is mounted on the economizer and measures ventilation air quantity
- A piezometer ring mounted at the SA fan inlet to measure SA volume
- An exhaust/return fan that equalizes building pressure when ventilation air is in use
- An energy conservation wheel that recovers heat from exhaust air (EA)
- A hot gas reheat (HGRH) coil that raises the SA temperature (SAT) during dehumidification

Finally, RTUs have several different heating options that can vary depending on their location:

- A gas fired system using natural or propane (LP) gas. This is a common option where gas is readily available
- Electric resistance heat

  In some areas, this option is restricted by codes. For example, in California, it is restricted by Title 24
- Steam and hot water coils when the building has an available heat source
- Heat pumps with smaller RTUs

Matching RTU Capacity to the Load

An RTU offers economy of scale. A single large unit can serve the cooling needs of several thousand square feet of space. However, the application of an RTU can be a challenge for the designer. The RTU is sized for the maximum load, that is, the hottest days of the year and the maximum internal load of people, lights and equipment. These conditions may only occur for a few days out of the entire year. For much of the time, the RTU is oversized, especially when occupancy or space cooling fluctuates throughout the day, such as classrooms, conference rooms, auditoriums, gymnasiums, places of worship, warehouses, and retail shops.

On days that match the design cooling criteria, the system runs at full capacity, providing needed cooling to the occupied space. When the demand for cooling is low, the unit is oversized for the load. The cooling system comes on, runs for a short period, and then turns off. When humidity is high, this can be problematic because the unit does not run long enough to provide adequate dehumidification.

Multiple cooling stages and multispeed fans can help match capacity to the load while hot gas bypass (HGBP) can keep the unit running longer for dehumidification. HGBP creates a false load on the compressor to keep it running longer. However, there is an energy penalty. Recent revisions to the energy code require fundamental changes to the way we think of comfort cooling (ASHRAE 2016a).

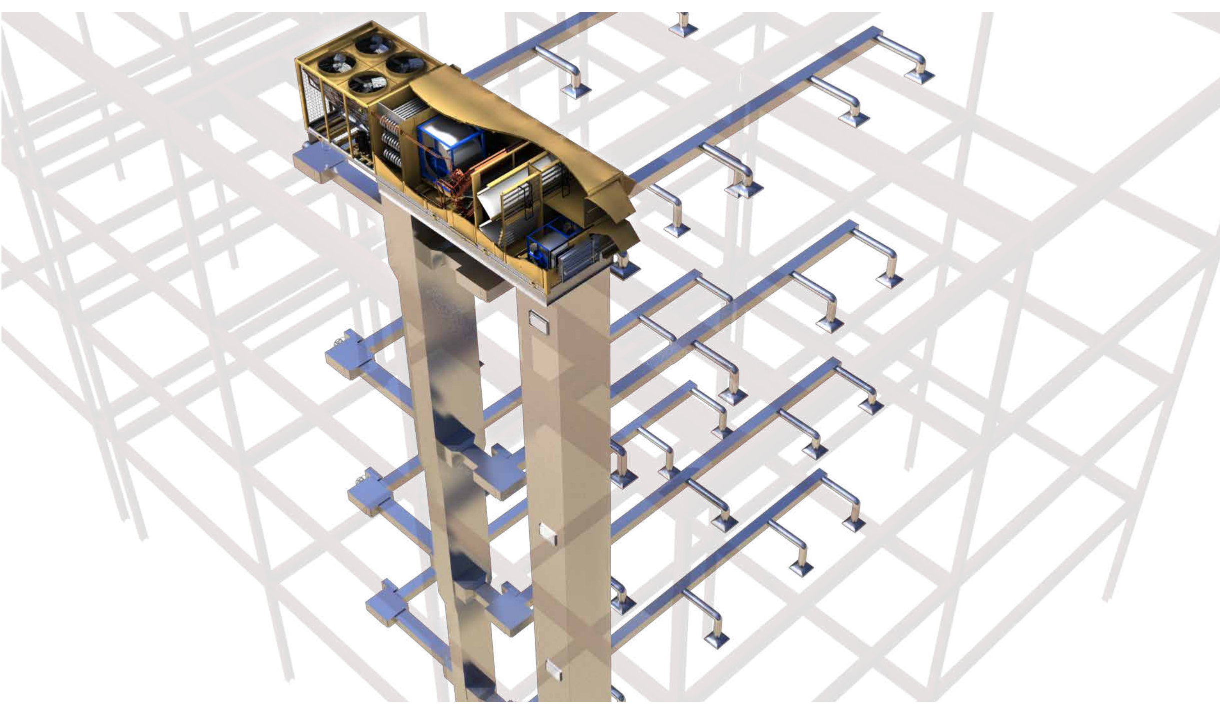

Figure 2: RTU and Ductwork to Variable Air Volume (VAV) Boxes (Building Cutaway)

Constant Volume (CV) and Single Zone VAV (SZVAV) Systems

RTUs supplying a large open area are typically set up with either Constant Volume (CV) or Single Zone VAV (SZVAV) control logic. In a CV system, the amount of air supplied to the occupied space is the same, independent of the cooling load, while the SA temperature varies with the load. When cooling demand is satisfied, the compressors turn off and await the next demand for cooling (The fan can be kept running to supply ventilation).

As a SZVAV, the RTU provides supply air at a constant temperature (55.0°F) and varies the airflow to match the system capacity to the room load. By using a variable frequency drive (VFD) on the supply fan and keeping a constant SAT, the cooling capacity can closely match the load in the space. This matching avoids wide temperature swings, allowing adequate run time for humidity control and providing better comfort to the occupants.

SZVAV also has the benefit of saving energy. By fan laws, fan power is a cube function of the fan speed. Reducing the airflow by decreasing fan speed results in less energy consumption compared to a CV system.

Relative to CV, SZVAV requires several changes to the RTU design. It has a room temperature sensor instead of a room thermostat. The sensor provides a proportional signal to the RTU controller, regulating airflow via the VFD drive on the SA fan.

A discharge air sensor measures the SAT, providing a proportional signal to the system controller. To maintain the discharge air temperature, additional cooling stages energize as required. A HGRH coil can be incorporated to keep SA at a comfortable temperature during low load dehumidification operation.

SZVAV design was originally and primarily for large RTUs to supply large common zone spaces. However, the wider industry have recognized the possible energy efficiency gains. The design has been applied across the HVAC equipment spectrum. Requirements for SZVAV were added to the 2010 edition of ANSI/ASHRAE/IESNA Standard 90.1, Energy Standard for Buildings Except Low-Rise Residential Buildings. The US Department of Energy (DOE) also mandates that each state update its commercial building codes to meet or exceed this edition of Standard 90.1 by October 18, 2013.

According to ASHRAE 2010, HVAC systems shall have variable airflow controls:

a. Air-handling and fan-coil units with chilled-water cooling coils and supply fans with motors greater than or equal to 5 hp shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50%, the supply fan controls shall be able to reduce the airflow to no greater than the larger of the following:

    1. One-half of the full fan speed, or

    2. The volume of *outdoor air* required to meet the *ventilation* requirements of AS-Standard 62.1.

b. Effective January 1, 2012, all air-conditioning equipment and air-handling units with direct expansion cooling and a cooling capacity at AHRI conditions greater than or equal to 110,000 Btu/h that serve single zones shall have their supply fans controlled by two-speed motors or variable-speed drives. At cooling demands less than or equal to 50%, the supply fan *controls* shall be able to reduce the airflow to no greater than the larger of the following:

    1. Two-thirds of the full fan speed, or

    2. The volume of outdoor air required to meet the ventilation requirements of Standard 62.1.

(ASHRAE 2010, 44)

Operational Sequence

Units configured for SZVAV operation include a VFD for the supply fan. Depending on the zone temperature, the unit switches between cooling mode, heating mode, and standby mode. In cooling mode, the supply fan speed varies based on zone temperature. If the zone temperature value increases, the supply fan speed increases. Conversely, if the zone temperature value decreases, the supply fan speed decreases. In heating mode, the supply fan runs at full speed.

Figure 3 on page 10 details a SZVAV system's control sequence of a VFD controlling a variable speed supply fan. If the zone is at maximum sensible cooling load (OCC cooling high), the system delivers 100% SA flow at the designed SAT for cooling (usually 55.0–58.0°F).

As the zone cooling sensible load decreases, SA flow is reduced (the VFD varies the supply fan speed). to maintain the desired temperature in the zone. The unit cooling capacity is modulated (staging compressors) to maintain the active SAT setpoint.

For units with an airside economizer, the economizer operates with compressors to provide all or part of the cooling to achieve the SAT setpoint. The zone sensible cooling load decreases until SA flow reaches the minimum limit (SZVAV minimum setpoint).

The zone temperature drops to the heating setpoint and the fan continues to operate at minimum airflow (OCC standby). To maintain the active SA temperature setpoint, heating capacity stages (electric) or modulates (gas/hydronic). The zone heating load increases until the SAT is below a predetermined zone temperature setpoint limit (OCC heating low); to maintain the SAT, the SA flow increases to 100% and heating capacity stages or modulates.

Benefits of SZVAV

SZVAV was developed to overcome the negative aspects of the CV system. Generally, SZVAV offers a number of benefits over the alternatives. These benefits include energy saving, reduced noise, and reduced humidity.

Energy Saving – SZVAV units can reduce supply fan speeds. When a space is not fully occupied, ventilation needs are well below design-day conditions. In that case, fans do not need to run as fast as when the space is fully occupied. Fan power input is proportional to the cube of the fan rotational speed. For example, cutting the fan speed in half uses an eighth of the energy required for higher rotational speeds. As a result, lower fan speeds need less energy to run.

The compressors of SZVAV units operating at lower supply fan speeds may also cycle on and off less often than a unit with constant fan speeds. Each time a motor or compressor cycles on, there is an inrush of current. The amount of inrush current exceeding normal operating current can vary for each unit design or size. However, direct expansion (DX) compressors that cycle more often have more periods of high current. These compressors tend to use more energy. The single VAV compressor cycling reduction uses less energy.

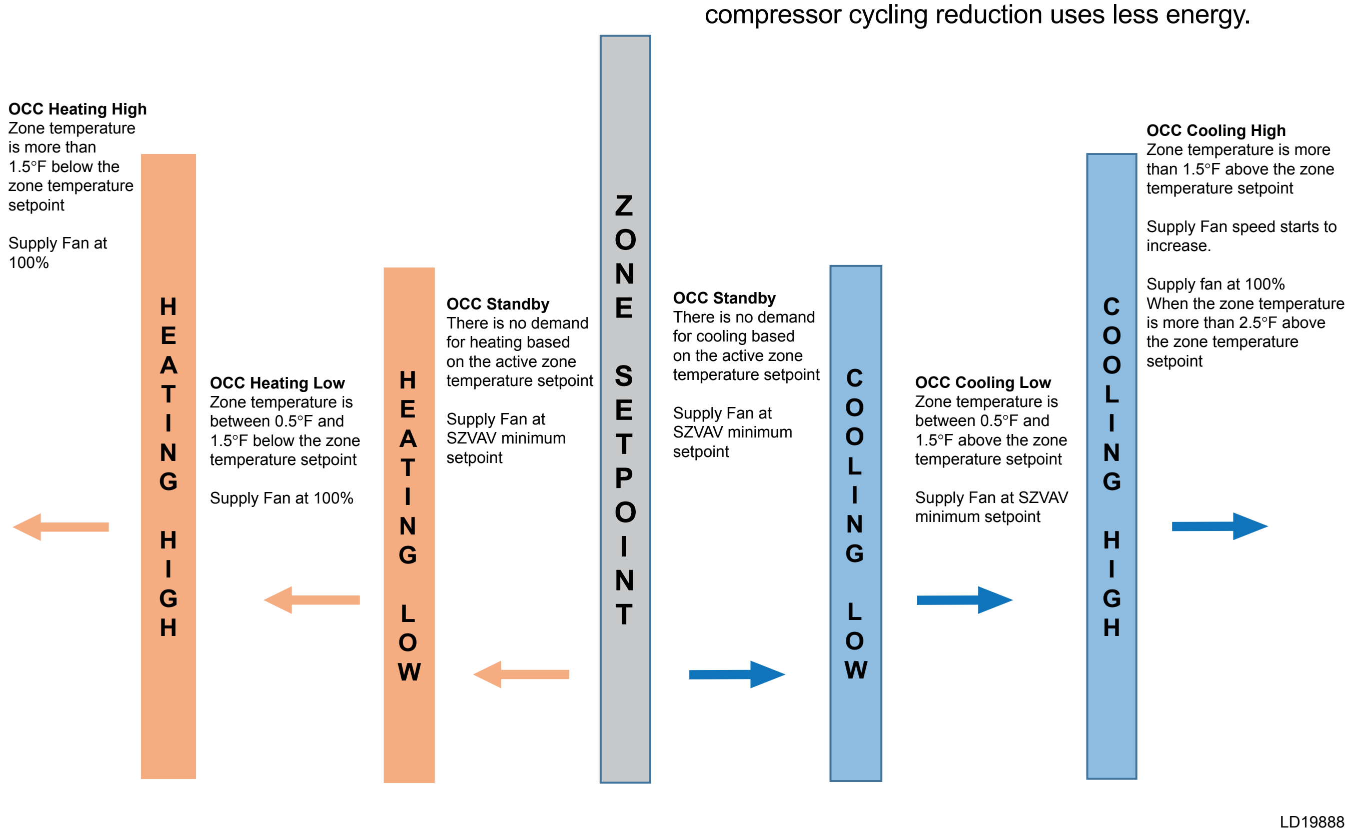

Figure 3: RTU SZVAV Operation

Notes

1. Whenever the unit enters an active cooling or heating mode, the unit controller utilizes as many or as few stages of cooling or heating that it needs to achieve and maintain the active SA temperature setpoint.
2. An unoccupied sequence is the same as 1 except the zone temperature setpoints are the unoccupied setpoints' values.
3. The unit modes stage down when the zone temperature is 0.5°F under cooling setpoint and 0.5°F over heating setpoint.

Quietness – Supply fans produce more noise in the zone when running at higher speeds. The fans in SZ-VAV units run at lower speeds for longer than the alternatives, resulting in less noise.

Reduced Humidity – SZVAV units passively dehumidify the environment. They maintain a constant zone temperature by varying fan speed while keeping the compressors running. For example, if the zone calls for less cooling, the SZVAV unit maintains the zone's temperature by keeping its compressor engaged and lowering the fan speed. The SAT remains low even though it passes over the evaporator coil at a slower rate. The cooler SA holds less water, resulting in more dehumidification.

In comparison, in a CV system, the same amount of air is moved through the evaporator coil whether or not the compressor is running. When the sensible cooling load of a space decreases, a constant volume of SA moves through the warmer evaporator coil (the compressor being cycled off). Warmer air is delivered to the conditioned space. Warmer air holds more water than colder air, resulting in less dehumidification of the supplied air.

VAV Systems

A VAV system allows a single RTU to provide the appropriate amount of cooling to multiple zones with different cooling loads. The RTU provides air at a constant temperature and airflow varies based upon the pressure of the SA duct. Downstream of the RTU, VAV boxes control the airflow to the related space based upon the signal from a zone air sensor. Most VAV systems are single duct. Many others are either series (constant) flow fans or parallel (variable) flow fans. Less common are dual duct VAV systems (about 5% of VAV systems).

All VAV systems have the following features:

- An RTU with SAT and static pressure (SP) controlled by an RTU controller
- Analog, pneumatic, or direct digital control (DDC) VAV controller for each zone

The RTU typically maintains about 1 inches of water gauge (iwg) SP downstream of the supply fan and inside the longest run of ductwork. This pressure ensures each VAV terminal unit has enough pressure at its inlet to deliver the maximum required flow of air into the space. As each VAV box opens and closes in response to temperature changes, the SP in the air handling system's main duct changes.

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Figure 4: VAV Examples

The function of the RTU's controller is to modulate the supply fan, increasing or decreasing the fan speed to maintain the desired duct SP. By maintaining the SP setpoint, this fan provides the required amount of airflow to each VAV box.

A VAV box's performance also depends on the correct box sizing for each zone. A VAV box's maximum cooling capacity depends on the VAV box's size. If an installed unit is too small, there is insufficient cooling. At high flow rates, it emits audible noise. If the unit is too large, small changes in damper position cause excessive changes in airflow. It can be difficult to control.

Single Duct Systems – Many single duct systems are only for cooling. When the zone temperature is below the setpoint, the damper slightly opens to provide the minimum fresh air volume requirement. In colder climates, exterior zones may require some form of heating. The system can heat the air entering the zone with staged or proportional electric heat, two position, or a modulated hot water coil.

Dual Duct Systems – All dual duct systems have a separate hot deck and cold deck. Air supplied to the zone may come from the hot or cold deck. Within the comfort zone, the box may supply a blend from both decks.

VAV Terminal Unit

Commonly called a VAV box, a VAV terminal unit modulates the airflow to the space with a VAV controller. The box is commercially manufactured with the following components:

- A control damper
- Inlet and outlet connections
- Options such as flow pickups, a return air (RA) plenum inlet, a heating coil, or a fan
- A dual duct box can also have inlets (or control) dampers for warm and cold air

Usually, the control damper is a butterfly type blade. The control damper rotates its shaft through a full stroke of 90°, 60°, or 45°. The degree of the rotation varies according to the manufacturer. Box manufacturers rate their boxes for a range of airflows based on inlet size and 1 iwg inlet duct SP. There are two control strategies for VAV terminal units:

- Pressure dependent
- Pressure independent

Pressure Dependent – The amount of air delivered to the space depends on the inlet duct SP and control damper position. Pressure dependent control does not use a device to measure inlet pressure as a means to determine flow. The space temperature control loop directly positions the damper.

The system has the following drawbacks:

- The effect of the damper position on space temperature is nonlinear
- The space temperature controller has no control over the actual airflow to the space

For example, if some boxes on a branch duct are closing, the resulting inlet pressure at the boxes that remain open increases. This causes more air to flow into the served spaces. The VAV box flow depends on duct SP.

Pressure Independent – This control strategy employs cascaded proportional/integral control loops. The zone temperature loop samples space temperature and resets the airflow setpoint between the minimum and maximum flow settings. The airflow loop uses this airflow setpoint. It samples airflow through a differential pressure transmitter (DPT) in the box inlet and modulates the damper to control the flow. The VAV box flow is independent of duct SP.

The engineering basis for this control method is that a space's temperature with a constant load is linearly proportional to the flow of conditioned air into the space. The consulting engineer must accurately determine the required maximum and minimum flows for the space based on heating and cooling loads.

Airflow Measurement

The VAV terminal unit control has some common methods for flow measurement or sensing:

- **Differential Pressure** – Based on the pressure difference created by the motion of air. A DPT senses the pressure difference across a multiple port airflow pickup. This typically amplifies the velocity pressure.
- **Thermal** – (For example, hot wire) Based on the rate of cooling and the flow of air over a hot body.

  There are two basic types:

  - A single point duct insertion probe
  - A flow-through device that samples through a multiple port airflow pickup

The manufacturer recommends differential pressure sensing. Reasonable flow measurement accuracy can be obtained at velocities above 400 FPM and down to approximately 200 FPM. Given today's technology, the temperature effect of the pressure sensor is by far the greatest contributor to error in the indicated flow. The preferable pressure sensor has a minimal effect from temperature and maintains a relatively constant ambient temperature.

For example, with the following parameters:

- 1.5 iwg sensor with a temperature coefficient of offset of 0.06% of span per °F
- Temperature variation of +/- 3.0°F
- Airflow pickup gain (or K-factor) of 2.78

The flow indication error from temperature has one of the following values:

- Less than 5% at 400 FPM
- Less than 10% at 200 FPM

The largest effect is also on the sensor zero. An auto zero algorithm compensates this effect.

Although thermal types initially have better accuracy at low velocity, as dirt accumulates on the sensor, they sustain a shift in calibration over time. Hot wire insertion probes also have the disadvantage of sensing at a single point in the airstream.

Flow through types may use inline air filters. However, as the filters become loaded with dirt, their pressure drop increases, causing an apparent shift in sensor calibration. Further, this shift can affect both the sensor sensitivity and auto zero. An auto zero algorithm cannot compensate for a change in sensitivity. It requires verification at two or more recalibration points.

Airflow Pickup

Usually, an airflow pickup is located in the inlet of a pressure independent box to sample the airflow. The pickup can have a variety of shapes:

- A molded plastic cross shape
- A pair of rings
- Straight sections of 1/4-inch diameter aluminum or copper tubing

For examples, see *Figure 5.*

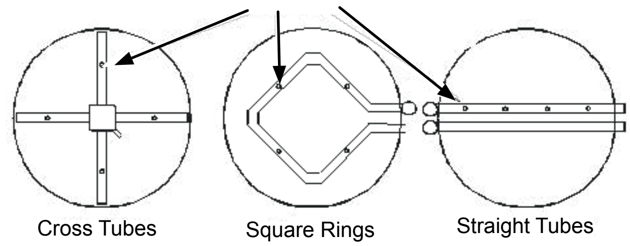

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Figure 5: Common Flow Pickups (Box Inlet View)

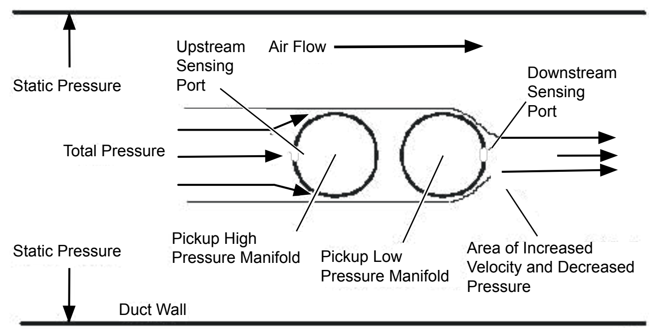

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Figure 6: Pickup and Air Stream Interaction

Functionally, the pickup consists of two manifolds with an equal number of symmetrically located ports. The high side manifold ports face upstream and the low side ports open downstream. Each manifold averages the samples from its multiple ports.

When the air velocity is non-uniform across the duct area, these manifolds give a better indication than a single port pickup of average pressure. Non-uniform velocity is common in VAV installations. Usually, it is caused by turns, other transitions, or sagging flex duct within three diameters upstream of the flow pickup.

Usually, among multiport devices, the cross and ring types perform better than straight tubes because the sensing ports are more distributed across the duct area.

As shown in *Figure 6*, the upstream ports are exposed to total pressure. In order to sense true SP, the pickup must have openings that are perpendicular to the direction of flow. Instead, the low pressure ports open downstream and the passing air exerts a pull on these openings. This pull results in a pressure less than static.

The velocity pressure is the total pressure minus the SP:

$$P_{velocity} = P_{total} - P_{static}$$

The differential pressure is the total pressure minus the downstream pressure:

$$P_{differential} = P_{total} - P_{downstream}$$

If downstream pressure is less than SP, then the pressure difference across the pickup must be greater than the velocity pressure. As a result, the velocity pressure is amplified by this effect. The box inlet's aerodynamic design and flow characteristics determine the pickup's amount of amplification or gain. It varies among box manufacturers in the range of 1.5 to 3.5.

Flow Multiplier

Usually, users of digital systems expect an accurately calculated and displayed flow. As a result, the exact pickup gain (or K-factor) must be provided to the control algorithm. In VAV applications, the term flow multiplier is used for pickup gain. Velocity pressure is expressed by the following equation:

$$P_{velocity} = \frac{P_{differential}}{\text{Flow Multiplier}}$$

Flow is expressed by the following equation:

$$\text{Flow} = \text{Area x } 4{,}005 \text{ x } \sqrt{P_{velocity}}$$

Flow is CFM, area is sq. ft, and $P_{velocity}$ is iwg. The combination of these gives the following equation:

$$\text{Flow} = \text{Area x } 4{,}005 \text{ x } \sqrt{\left(\frac{P_{differential}}{\text{Flow Multiplier}}\right)}$$

NOTE: For metric equivalents of these equations (in l/s), see *Pressure Independent Decks on page 19*.

For most currently manufactured VAV boxes, flow multipliers are used with dead-ended devices (for example, differential pressure transducers).

Box designs change over time. Some controls companies specify flow pickups other than what is normally supplied by the box manufacturer. The published gains may not apply to existing boxes that are retrofitted with new controls. In these cases (or when using a flow-through device such as a hot wire sensor), contact the box manufacturer and calculate the correct pickup gain.

Instead of pickup gain, box manufacturers provide a number representing the flow in CFM at 1 iwg differential pressure. This number is a combination of the gain, inlet area, and the constant 4,005 (derived from Bernoulli's equation).

This number can also be estimated by the graph normally attached to the VAV box's side. These graphs plot flow against differential pressure. However, it is often incorrectly labeled velocity pressure. The flow multiplier is calculated by the following equation:

$$\text{Flow Multiplier} = P_{differential} \text{ x } \left(\frac{4{,}005 \text{ x Area}}{\text{Flow}}\right)^2$$

Flow is CFM, area is sq. ft, and $P_{differential}$ is iwg.

During test and balance, using the balancer's reading, the flow multiplier can be adjusted to match the controller flow indication. However, if the two readings differ by more than 20%, it is better to investigate the cause of the difference.

Airflow Test and Balance Concerns

Pressure independent VAV control jobs frequently require an accuracy within 5–10% of flow, both actual and indicated. The balancing contractor must adjust and certify the flow rates specified by the consulting engineer. Sometimes, the balancer's readings do not correspond with the VAV controller's indicated flow.

When airflow readings disagree, there may be a problem or there may be some fact of the air delivery system that is unknown or not understood. There are margins for error in measuring equipment used by both the controller and the balancer. Therefore, it is important that both contractors (controls and balancing) understand each other's equipment, techniques, and expectations.

Single Duct Application Control Sequence

One single duct application is a user-defined, pressure-independent single control. *Figure 7* illustrates the logic of this pressure independent control.

For the selected operation mode, its generator selects zone cooling and heating temperature setpoints. The generator also selects the flow setpoint schedule. This schedule supplies the flow proportional (or integral) loop during the following modes:

- Occupied
- Unoccupied
- Warmup
- Standby
- Shutdown
- Auto Zero

Pressure independent controls have the following sequence:

1. The temperature control loop sequencer compares the zone temperature to the zone setpoint. It produces a 0–100% output command.
2. The output command feeds into the flow setpoint reset schedules. It provides a supply flow setpoint during the following modes: Occupied, Unoccupied, and Warmup.
3. The flow control loop compares the supply flow setpoint from the reset schedule to the actual flow calculated from the differential pressure input. It produces a 0–100% command to the damper.
4. The user defines the flow sensor type and ranging through a user-defined flow path. In addition to differential pressure measurement, this definition provides the user linear and non-linear sensors with output ranges in flow or velocity.
5. To linearize the sensor, the user enters the appropriate constants for the sixth order polynomial.
6. The user enters the flow coefficient, box area, and sets whether or not the calculation uses the box area. The box area assists the calculation of flow loop tuning parameter values.

   As a result, even if the area is not required to calculate flow, the area must be accurately entered. Setting the flow coefficient to zero disables square root extraction.

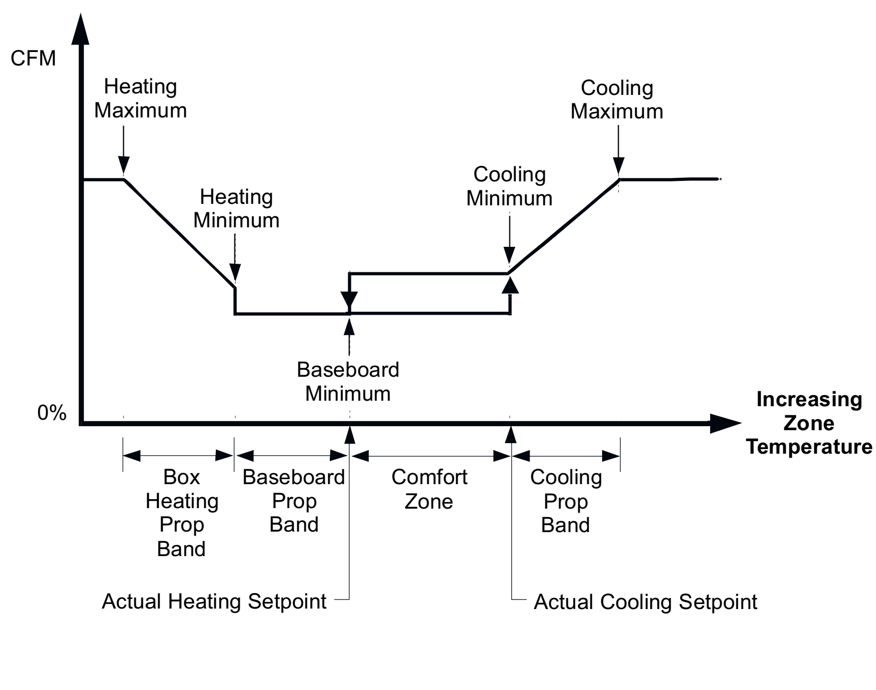


Figure 7: Pressure-Independent Sequence

Fan Operation

Some VAV boxes have small fans and a heating coil. These fans serve two purposes:

- When heating is required, they produce a flow of plenum air through the heating coil. They produce this flow even if the box damper is fully closed to the primary air source.
- They maintain a constant airflow through the diffuser independent of the box damper's position. This constant airflow improves occupant comfort by providing a better mix of delivered air and room air. As the box damper closes, the fan pulls more air from the plenum.

VAV box fans have two types:

- Series fans
- Parallel fans

For illustration of these fan types, see *Figure 8 and Figure 9. For descriptions of the types, see Table 1 on page 17.*

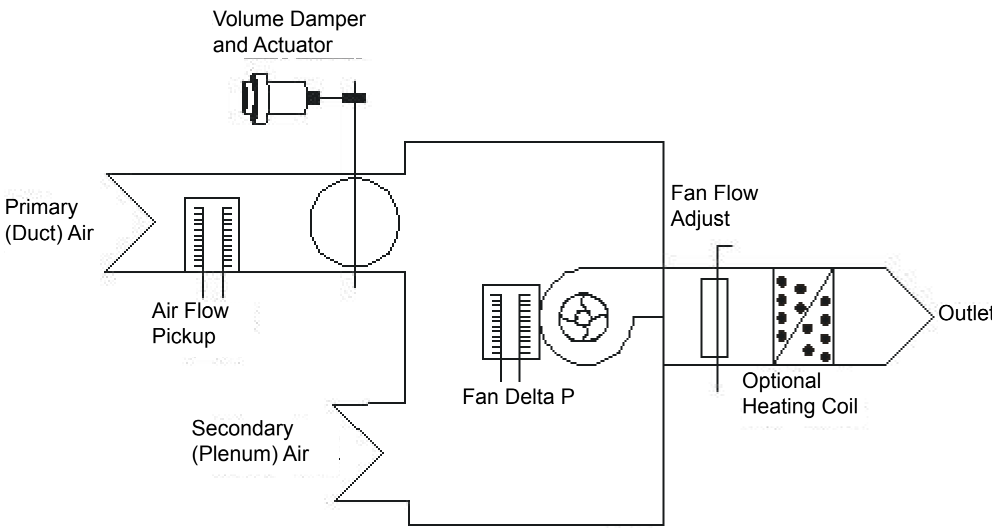
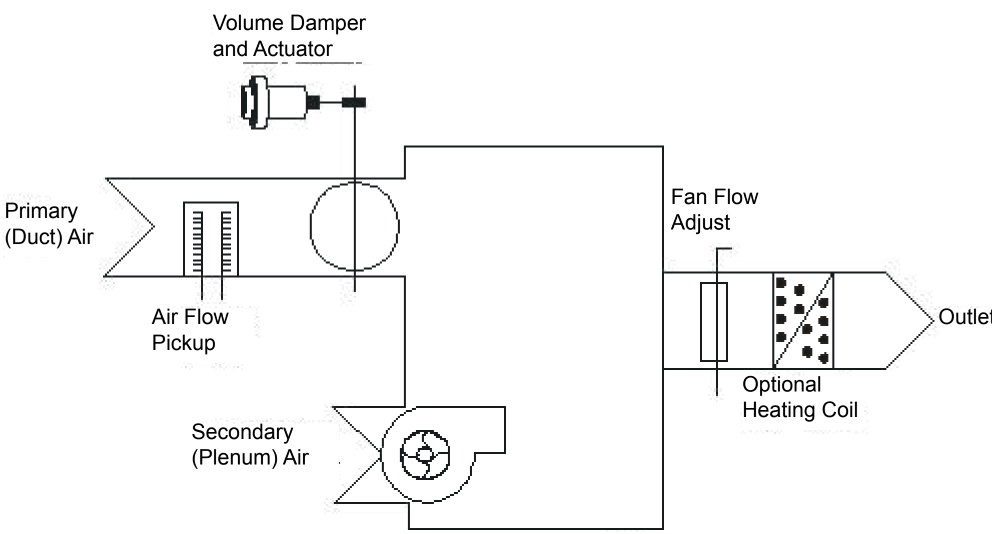


Figure 8: Single Duct VAV Box, Series Fan



Note

When used, the Fan Delta P pickup is attached to the fan inlet cone.

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Figure 9: Single Duct VAV Box, Parallel Fan

Table 1: Single Duct VAV Box Fan Types

Fan Type	Description
Series	The series fan is off during the shutdown and auto zero modes. The fan is always on during the occupied and standby modes and is cycled on during the unoccupied and warm-up modes when the zone requires heating. Before the fan is turned on, the damper is driven closed for the auto zero duration time to ensure that the fan is not spinning backward. The on/off series fan is controlled by a single binary output with minimum on/off timers that can be set in the binary output modify screen.
Parallel	The parallel or variable Fan is also referred to as fan assist. The fan is off during the shutdown and auto zero modes. The parallel/temp fan is cycled on when warm-up is inactive and the internal zone heating command is greater than the value of the fan start setpoint parameter defaulted to one percent. The parallel/flow fan is also cycled on during occupied and standby mode whenever the flow setpoint is below the parallel fan/flow parameter value. The supply deadband is used as a differential to turn the fan off. *Table 2* summarizes the relationship between the parallel and series fan types with the different modes of operation: • If the parallel fan is defined without setpoints, the fan starts during unoccupied when the heating command is greater than one percent and it stops when it is below one percent. • If the parallel fan is defined with setpoints, then the fan starts when the heating command is greater than the fan start setpoint and stops when the heating command is less than the fan start setpoint – parallel fan differential. • The parallel fan setpoint defaults to 1%, and the parallel fan differential defaults to 0%. These parameters must be adjusted to the required values.

Table 2: Parallel and Series Fan Status per Mode of Operation

Fan Type	Operation Mode		
	Occupied and Standby	**Unoccupied**	**Shutdown**
Parallel Fan/Temp	Cycled per Box[1] Heating Temperature Setpoints	Cycled at Unoccupied Box Heating Setpoint	Off
Parallel Fan/Flow	Cycled per Flow Setpoint	Cycled at Unoccupied Box Heating Setpoint	Off
Series Proportional	On	Cycled at Unoccupied Box Heating Setpoint	Off
Series On/Off	On	Cycled at Unoccupied Box Heating Setpoint	Off

Notes
1. This is on when the heating command is greater than the Fan Start Setpoint.
2. Fan and heat control outputs operate independently when overridden.

Box Heat

Box heat support includes the following types:

- Incremental
- Proportional
- Two-position valve actuators
- One to three stages of electric heat

Electric heat control for non-fan powered boxes has logic to avoid heat operation with inadequate airflow. Otherwise, the operation can trip electrical overload protection.

Usually, in the absence of inlet SP, VAV box manufacturers provide a pressure switch to lock out electric heat. However, this does not ensure adequate airflow.

The logic is for staged and two-position heating options because both can control electric heat. The provided function depends on selection of the main box strategy.

Patented FlowStar™ Sensor Control

The air valve features the FlowStar™ airflow sensor. This brings new meaning to airflow control accuracy. The multi-axis design uses between 12 and 20 sensing points that sample total pressure at center points within equal concentric cross-sectional areas, effectively traversing the air stream in two planes. Before being sent from the sensor to the controlling device, each distinct pressure reading is averaged within the center chamber.

The sensor adds a new dimension to signal amplification. Most differential pressure sensors provide a signal equal to 1.5 times the equivalent velocity pressure signal. The FlowStar provides a differential pressure signal that is 2.5 to 3 times the equivalent velocity pressure signal. At low airflow capacities, this amplified signal allows more accurate and stable airflow control. Low airflow control is critical for indoor air quality, reheat minimization, and, during light loads, preventing over-cooling.

Unlike other sensors that use a large probe surface area to achieve signal amplification, the FlowStar uses an unprecedented streamline design to generate amplified signals unrivaled in the industry. The streamlined design also generates less pressure drop and noise.

The VAV schedule specifies several values for the sensor:

- Minimum and maximum airflow setpoints
- Maximum sound power levels
- For each terminal, maximum air pressure loss

For a VAV terminal, the specification must detail the required performance of the airflow sensor. For maximum building occupant satisfaction, the VAV system designer specifies the airflow sensor as suggested in *VAV Terminal Unit on page 12.*

A system using FlowStar sensing to amplify the airflow signal can have lower minimum airflow setpoints. Many VAV controllers require a minimum differential pressure signal of 0.03 iwg. The airflow sensor can generate this signal with only 400–450 FPM air velocity through the inlet collar.

Conventional airflow sensors without the ability to amplify require approximately 700 FPM to generate a 0.03 iwg signal. If 700 FPM represents a 20% minimum condition, at the maximum airflow setpoint, the inlet velocity is 3500 FPM. Some consequences of this velocity are extremely noisy conditions. Over the operating range of the terminal unit, the airflow sensor also generates a differential pressure range of at least 1 iwg.

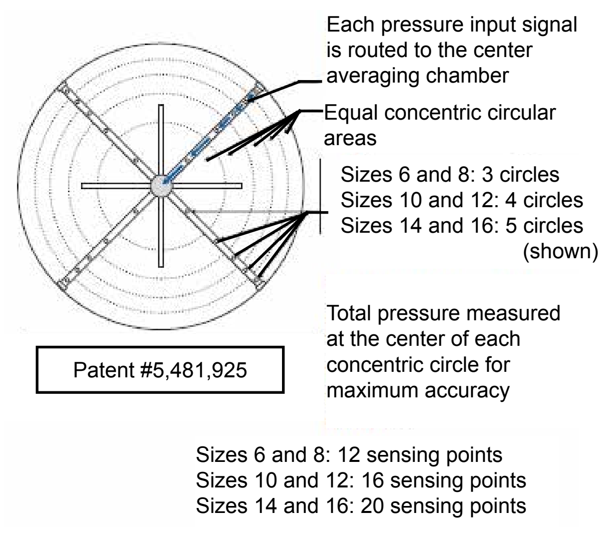


Figure 10: FlowStar Airflow Sensor

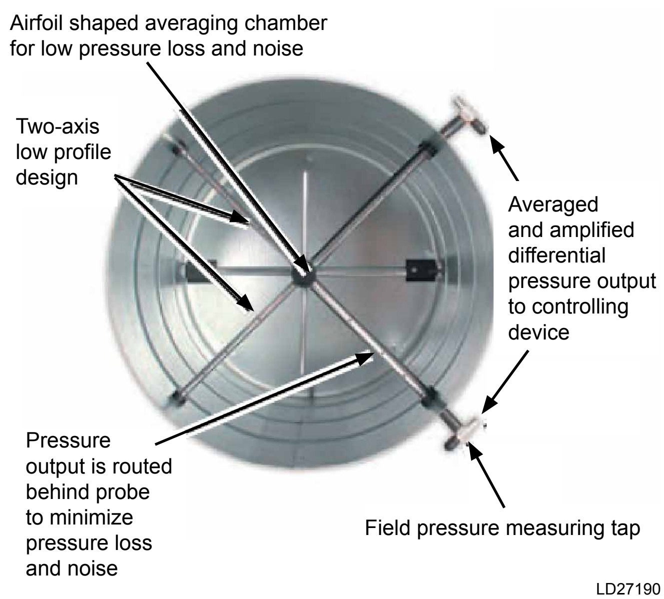


Figure 11: FlowStar Features

VAV Dual Duct Applications

There are a number of dual duct applications, including applications for pressure independent decks, CV separate dampers, zone control, the patented FlowStar™ sensor control, and noise criteria (NC).

Pressure Independent Decks

Figure 12 illustrates pressure independent control logic. For the selected operation mode, the operation mode generator selects the zone cooling and heating temperature setpoints.

The mode generator also selects the flow reset schedule (supplying both the hot and cold deck damper actuator) during the following modes:

- Occupied
- Unoccupied
- Warmup

Pressure independent controls have the following sequence:

1. The zone proportional/integral loop compares the zone temperature to the zone setpoint. It produces 0–100% output commands.
2. The output commands for heating and cooling feed into separate hot and cold deck reset schedules. These provide flow setpoints.

   To modulate the damper and as a result maintain the flow setpoint, the damper control uses the following parameters:

   - Prop band
   - Integration time
   - Deadband
   - Stroke time
3. The hot and cold deck flows reset between the zone heating and cooling setpoints.

   The cold deck automatically resets between its minimum cooling flow setpoint and minimum heating flow setpoint. The hot deck automatically resets between its minimum heating flow setpoint and minimum cooling flow setpoint.

   This allows for smooth transitions in airflow as the zone temperature requirements switch between heating and cooling.

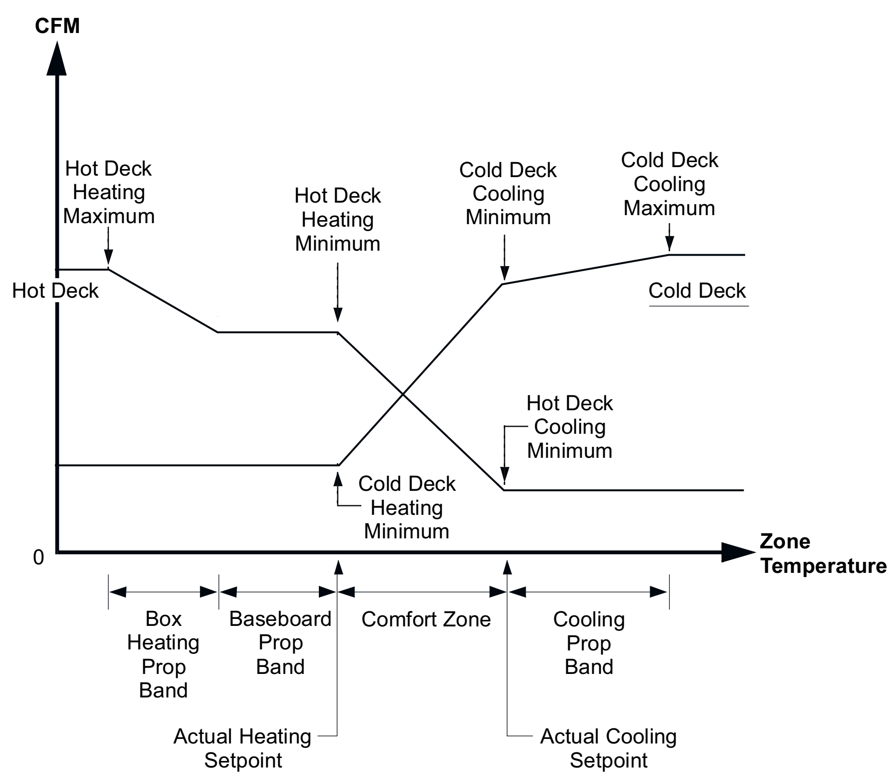


Figure 12: Control Sequence for Pressure Independent Hot and Cold Decks

4. The user defines the flow sensor type and ranging through a user-defined flow path.

   In addition to differential pressure measurement, this definition provides the user linear and non-linear sensors with output ranges in flow or velocity.

5. To linearize the sensor, the user enters the appropriate constants for the sixth order polynomial.
6. The user enters the flow coefficient, box area, and sets whether or not the calculation uses the box area.

   The box area assists the calculation of flow loop tuning parameter values. As a result, even if the area is not required to calculate flow, the area must be accurately entered. Setting the flow coefficient to zero disables square root extraction.

CV Separate Dampers

Figure 13 illustrates the separate damper, constant volume control strategy. This strategy uses separate damper actuators to modulate the hot and cold deck.

The control algorithm controls the temperature in the zone. It maintains a constant flow by proportionally resetting the hot and cold deck flow setpoints in response to the zone temperature. For full hot deck or cold deck flow, the zone heating and cooling setpoints set the zone temperature limits.

Whenever the zone temperature is between these limits, the hot and cold deck flow setpoints reset equally. However, in response to the zone temperature change, the setpoints reset in opposite directions. As a result, they maintain a constant volume airflow.

For example, for the hot and cold deck, a 500 CFM constant volume setpoint establishes a 0 CFM to 500 CFM range. Depending on the zone temperature, the hot deck control point could be 300 CFM. For the cold deck, this dictates a 200 CFM control point. When the zone temperature is half way between the heating and cooling setpoints, the control points for the hot and cold decks are both 250 CFM.

This strategy does not provide minimum flow setpoints for either deck. To provide this setpoint, we use the pressure independent strategy. Similar to previous strategies, this strategy involves a number of steps:

1. The user defines the flow sensor type and ranging through a user-defined flow path.

   In addition to differential pressure measurement, this definition provides the user linear and non-linear sensors with output ranges in flow or velocity.

2. To linearize the sensor, the user enters the appropriate constants for the sixth order polynomial.
3. The user enters the flow coefficient, box area, and sets whether or not the calculation uses the box area.

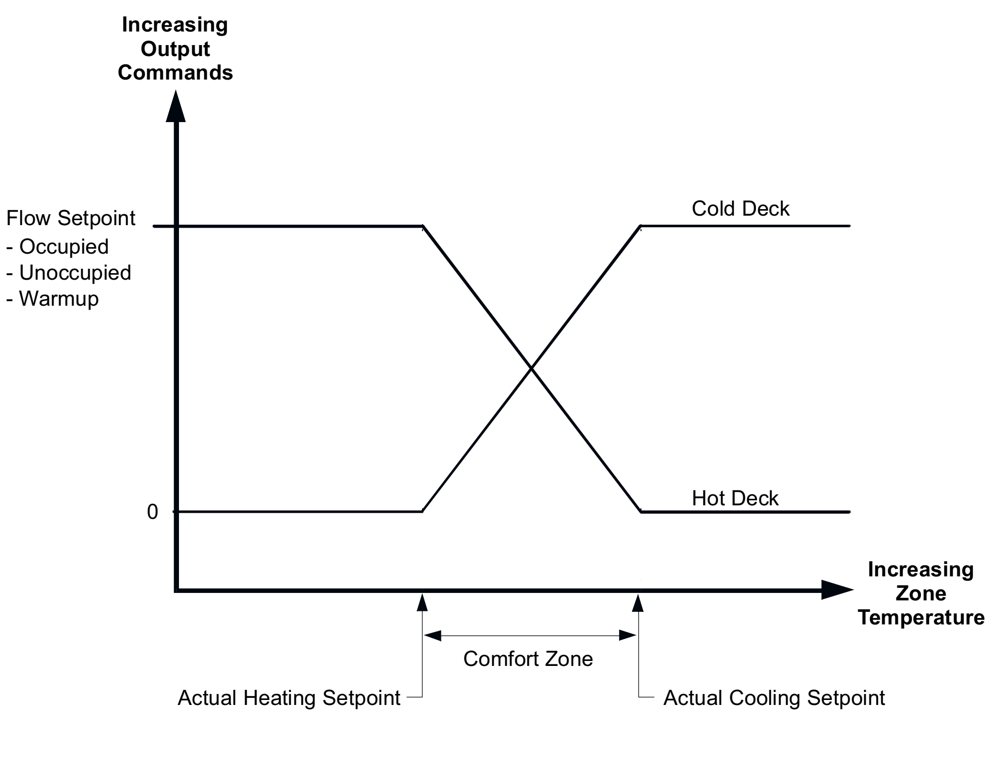


Figure 13: Control Sequence for Constant Volume Separate Dampers

The box area assists the calculation of flow loop tuning parameter values. As a result, even if the area is not required to calculate flow, the area must be accurately entered. Setting the flow coefficient to 0 (zero) disables square root extraction.

To provide tight zone control and stable discharge over a wide range of hot and cold deck SATs and space loads, this strategy uses the following features:

- A terminal unit discharge air temperature sensor
- Flow and temperature reset schedules

The reset strategy is particularly well-suited to cold air systems when the cold deck temperature may be less than the minimum desired air temperature delivered to the occupied space. Optionally, to eliminate the effects of cold walls in exterior zones, baseboard radiation control may be integrated.

The strategy uses proportional and integral control in the zone temperature loop and in the individual hot and cold deck flow control loops. Multi-variable control logic provide control of discharge temperature and flow.

Maximum heating and cooling airflow setpoints are in terms of discharge flow. For example, to produce 1,000 CFM of 55.0°F airflow to the space, if cooling design calls for 800 CFM of 42.0°F cold deck and 200 CFM of 107.0°F hot deck, the maximum cooling flow setpoint should be set to 1,000 CFM.

To meet ventilation requirements, a minimum airflow may be established on just one deck, on both decks, or on discharge flow.

For measurements, the manufacturer recommends the hot and cold inlet locations. However, flow can be measured at any two of the following three locations:

- Cold deck
- Hot deck
- Discharge

The unmeasured variable is internally calculated. Unless the flow pickup is located at least three duct diameters downstream from the box outlet, discharge flow measurement at low velocity may be less reliable. This makes the box manufacturer's installation of the discharge flow pickup impractical.

Zone Control

A single setpoint provides for a zone temperature setpoint for each of the following modes:

- Occupied
- Unoccupied
- Standby

The system also provides a bias for each of the three modes. In this configuration, the bias value establishes the zone control's deadband.

A deadband is the range of zone temperatures above and below the setpoint, range in which no control action takes place. This allows the zone to float. As a result, energy savings may be realized, for example, by using a larger bias during unoccupied periods.

The controller sets the deadband to be the zone setpoint with a margin equal to the bias. For stable control and expected component life, the manufacturer recommends using a bias of at least 0.1. Internally, to allow the control to function above and below setpoint, the control offset is set to 50%.

There are also the options of zone control loop tuning parameters for proportional band and integration time. The proportional band uses a positive value for direct action. The output from the zone control loop (or zone command) is a value of 0% to 100%. The discharge temperature and flow reset schedules use this output.

When the system includes baseboard control and requires heating, before resetting the discharge flow from heating minimum to heating maximum, the system sequences the radiation valve. A proportional band facilitates tuning. The band must have a negative value to produce the reverse acting heating ramp. The radiation control loop also uses the zone integration time.

Acoustical Concepts

The focus on indoor air quality also affects the proper selection of air terminal equipment for acoustics.

Sound

At the zone level, the terminal unit generates acoustical energy that can enter the zone along two primary paths. First, sound from the primary air valve can propagate through the downstream duct and diffusers before entering the zone. This is discharge or airborne sound. Second, acoustical energy radiates from the terminal casing and travels through the ceiling cavity and ceiling system before entering the zone. This is radiated sound.

To properly quantify the amount of acoustical energy emanating from a terminal unit at a specific operating condition (for example, SP), manufacturers must measure and publish sound power levels.

Units of sound measurement (decibels) actually represent units of power (watts). The terminal equipment sound power ratings provide a consistent measure of the generated sound, independent of the environment in which the unit is installed. This allows a direct comparison of sound performance between equipment manufacturers and unit models.

Noise Criteria (NC)

For most projects, the bottom line acoustical criterion is the NC level. The NC level is derived from resulting sound pressure levels in the zone. The causes of these sound pressure levels is acoustical energy (sound power levels) entering the zone from the terminal unit and other sound generating sources. Other sources include a central fan system, office equipment, and the environment. The unit of measurement is decibels. However, because ears and microphones react to pressure variations, in this case the decibels represent units of pressure (Pascals).

There is no direct relationship between sound power levels and sound pressure levels. We must base predictions of the zone's resulting sound pressure levels (or NC levels) in part on the terminal equipment's published sound power levels. The NC levels depend, architecturally and mechanically, on the project specific design. For a constant operating condition (fixed sound power levels), the zone's resulting NC level varies from one project to another.

AHRI 885

A useful tool for predicting space sound pressure levels is the application Standard AHRI 885. AHRI 885 provides information (for example, tables and formulas) required to calculate the attenuation of the ductwork, ceiling cavity, ceiling system, and conditioned space below a terminal unit.

These attenuation values are known as the "transfer function." They assist transfers from the manufacturer's sound power levels to the resulting estimated sound pressure levels in the space that is below or served by the terminal unit.

AHRI 885 does not provide all of the necessary information to accommodate every conceivable design. However, for most applications, it provides enough information to approximate the transfer function. Manufacturers have different assumptions about a typical project design. It is impractical to compare product performance by looking at the published NC values (AHRI 2008).

Quiet Design – General Recommendations

For quiet design, the following steps are generally recommended.

The RTU

Frequently, the zone's sound levels are impacted by central fan discharge noise that either radiates from the ductwork or travels through the distribution ductwork and enters the zone as airborne (discharge) sound. Achieving acceptable sound levels in the zone begins with a properly designed central fan system that delivers relatively quiet air.

Supply Duct Pressure

A primary contributor to noisy systems (including single duct applications) is high SP in the primary air duct. As the primary air valve closes to reduce the pressure, high SP causes higher sound levels from the central fan and also higher sound levels from the terminal unit. When a flexible duct is used at the terminal inlet, this contribution is compounded. It allows the central fan noise and air valve noise to break out into the ceiling cavity, then enter the zone below the terminal.

Ideally, system SP reduces to the following point: on the duct run associated with the highest pressure drop, a terminal unit has the minimum inlet pressure for design airflow to the zone.

In the main trunk, many current HVAC systems experience a 0.5 iwg pressure drop or less. For systems that have substantially higher pressure variances from one zone to another, special attention is paid to the proper selection of air terminal equipment.

To date, the most common approach is to select (to size) all the terminals based on the worst case condition (the highest inlet SP).

Usually, this method results in 80% or more terminal units being oversized for their application, with the following problems:

- Much higher equipment costs
- For each unit, greatly reduced operating efficiency
- Reduction in the unit's ability to provide the zone's comfort control
- Inadequate control of the minimum ventilation capacity required for heating

A more prudent method is, on those few zones closest to the central fan, to use a pressure-reducing device upstream from the terminal unit. If the device is located far upstream from the terminal inlet, this device can simply be a manual quadrant-type damper.

In tight quarters, perforated metal can quietly reduce system pressure. Perforated metal allows all terminal units to have similar, lower inlet pressures. At lower inlet pressures, they can be selected in a consistent manner. This selection allows more optimally sized units (as illustrated in *Figure 14*).

Zoning

On projects that do not permit internal lining of the downstream duct, acceptable noise levels require special considerations. In these cases, a greater number of smaller zones help reduce sound levels.

Where possible, the designer uses the following configuration:

- The first diffuser takeoff is located after an elbow or tee.
- There is a greater number of small-necked diffusers (rather than fewer large-necked diffusers).
- To avoid noise regeneration, the downstream ductwork is carefully designed and installed.
- To provide an established flow pattern downstream of the fan, bull-head tee arrangements are located sufficiently downstream from the terminal discharge.
- After the airflow completely develops, diffusers are located downstream from the terminal.

Downstream splitter dampers can cause noise problems if placed too close to the terminal or when there are excessive air velocities. If there are tee arrangements, volume dampers are used in each branch of the tee, and balancing dampers are provided at each diffuser tap. This configuration offer maximum flexibility in quiet balancing of the system.

Usually, casing radiated sound determines the overall room sound levels directly below the terminal. As a result, the location of these terminals and the size of the zone require special consideration. Larger zones have the terminal located over, for example, a corridor or open plan office space and not over a small confined private office. Fan powered terminals are never installed over small occupied spaces where the wall partitions extend from slab-to-slab (for example, fire walls or privacy walls).

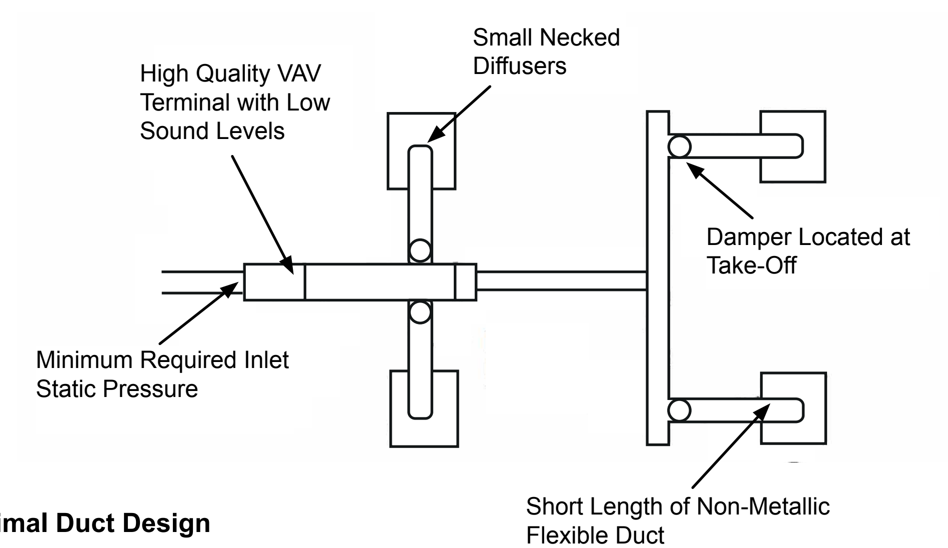


Figure 14: Optimal Duct Design

VAV Terminal Unit Types

There are several types of VAV terminal unit, each with its own features and benefits. The types include the following models:

- Single duct
- Dual duct
- Parallel fan powered
- Serial fan powered

Single Duct VAV

The basic unit is a single duct. It is composed of a sheet metal casing and an air valve. The air valve modulates the air delivered into the occupied zone. Air enters the air valve inlet and exits into the sheet metal casing. It is distributed to the occupied zone through ductwork attached to the discharge of the unit.

The single duct can be ordered with either a factory-mounted hot water heating coil or an electric heater. When the load in the occupied space drops off, the system uses these re-heat units to primarily reheat the air-to-zone temperature. Rotating the damper blade through the FlowStar air valve modulates the primary air.

The air valves come in two types:

- Rectangular
- Round

The round valves only come in diameters of 4, 5, 6, 8, 10, 12, 14, and 16 inches. Metric ductwork must have an adapter.

LD27192

Figure 15: Single Duct VAV

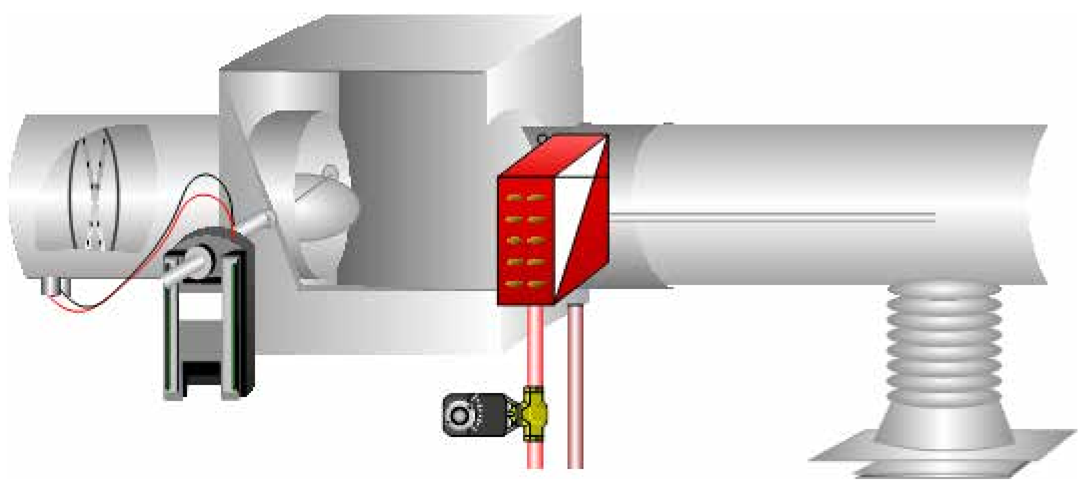
Precise Zone Control

Single duct VAV terminals provide VAV control beyond the typical single duct box. Independent of the installed inlet conditions, they are specifically designed for precise air delivery throughout the entire operating range. For a wide variety of HVAC applications, they also offer improved space comfort and flexibility.

Single duct VAV terminals have the advantage of single duct unit benefits, while performing at extremely low sound levels. This is critical in today's buildings, where occupants place more emphasis on indoor acoustics.

For any VAV terminal, the measurement of quality is the ability to provide occupant comfort. Comfort is achieved through quiet and precise control of airflow to the occupied space.

The single duct VAV terminal provides the ultimate airflow control with the patented FlowStar™ airflow sensor. No other sensor in the industry matches the FlowStar sensor's ability to quietly and precisely measure airflow. The basis for airflow control is accurate airflow measurement.



LD27193

Figure 16: Single Duct VAV With Hot Water Reheat Coil

Design Flexibility

The single duct VAV terminal has a flexible system design. The compact cabinet and quiet operation gives the system designer the versatility to place units directly above occupied spaces. The single duct VAV does not need to be located in a crowded space above a hall or corridor. This reduces lengthy and expensive discharge duct runs.

The FlowStar sensor ensures accurate control, even when space constraints do not permit long straight inlet duct runs to the terminal.

To handle airflow capacities between 45 CFM and 8,000 CFM, single duct VAV terminals come in ten unit sizes.

Convenient Installation

All single duct VAV terminals are thoroughly inspected during each step of the manufacturing process. To maintain the highest quality product, this inspection includes a comprehensive pre-shipping inspection.

All single duct VAV terminals are packaged to minimize damage during shipment.

Dual Duct VAV

Dual duct VAV terminals provide VAV control beyond the typical dual duct box. They are specifically designed for precise air delivery throughout the entire operating range, independent of the installed inlet conditions. These units can be ordered with or without a direct digital controller. The digital controller can operate as a stand-alone, on a BACnet® trunk, or on a LON trunk.

Dual duct VAV terminals have benefits common to all dual duct units and also perform at extremely low sound levels. This is critical in modern buildings where occupants place more emphasis on indoor acoustics. The measurement of quality for any VAV terminal is its ability to provide occupant comfort. This comfort comes from a quiet and precise control of airflow to the occupied space.

LD27194

Figure 17: Dual Duct VAV

The dual duct VAV terminal is manufactured and assembled with two airflow sensors. These sensors provide a signal to each respective controller, allowing them to quietly and precisely measure airflow. Accurate airflow measurement is the basis for airflow control.

The dual duct VAV terminal series is available in the following arrangements:

- Heating and cooling with mixing (constant volume)
- Heating and cooling with mixing (variable volume)
- Heating and cooling with reheat (optional)

Design Flexibility

Dual duct VAV terminals have a flexible system design. The compact cabinet design and quiet operation allow the system designer to place units directly above occupied spaces. It is not necessary to locate the unit in the crowded space above a hall or corridor, reducing lengthy and expensive discharge duct runs. The flow sensor ensures accurate control, even when space constraints prevent long straight inlet duct runs to the terminal.

To handle airflow capacities between 45 and 4100 CFM, dual duct VAV cold deck and hot deck air valves come in 8 unit sizes.

Convenient Installation

All dual duct VAV terminals are thoroughly inspected during each step of the manufacturing process. To maintain the highest quality product, this inspection includes a comprehensive pre-shipping inspection. All dual duct VAV terminals are packaged to minimize damage during shipment.

Quick Installation

All dual duct VAV terminals have as standard a single point electrical main power connection. This provides all the electronic controls and electrical components, which are located on the same side of the casing for quick access, adjustment, and troubleshooting. The flow sensor ensures accurate airflow measurement independent of the field installation conditions. For quick reference during start-up, a calibration label and wiring diagram is located on the terminal. The terminal is constructed to allow installation with standard metal hanging straps. There is also the option of hanger brackets for use with all-thread support rods or wire hangers.

The dual duct VAV terminal has many other features that, unique to it, come as standard. For other manufacturers, most of these features are expensive options. *Figure 18 on page 26* shows some of these features.

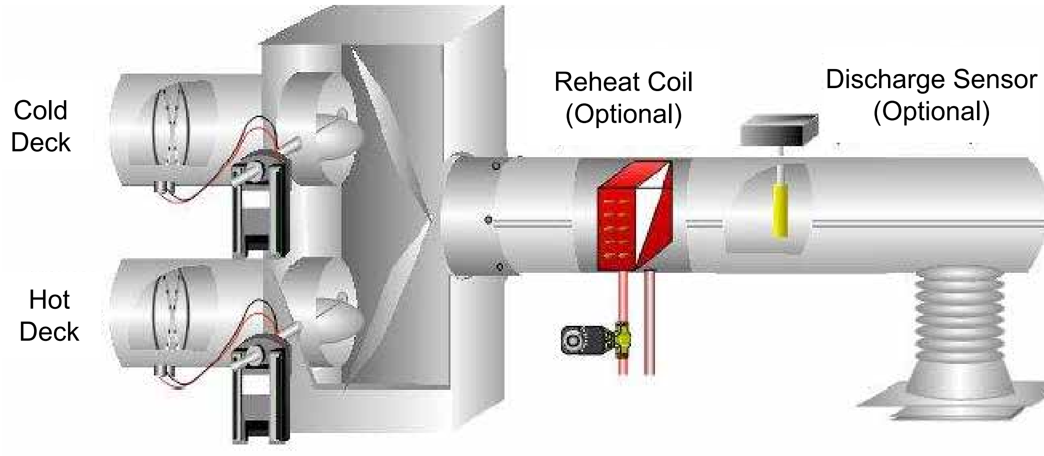


Figure 18: Dual Duct VAV Standard Features

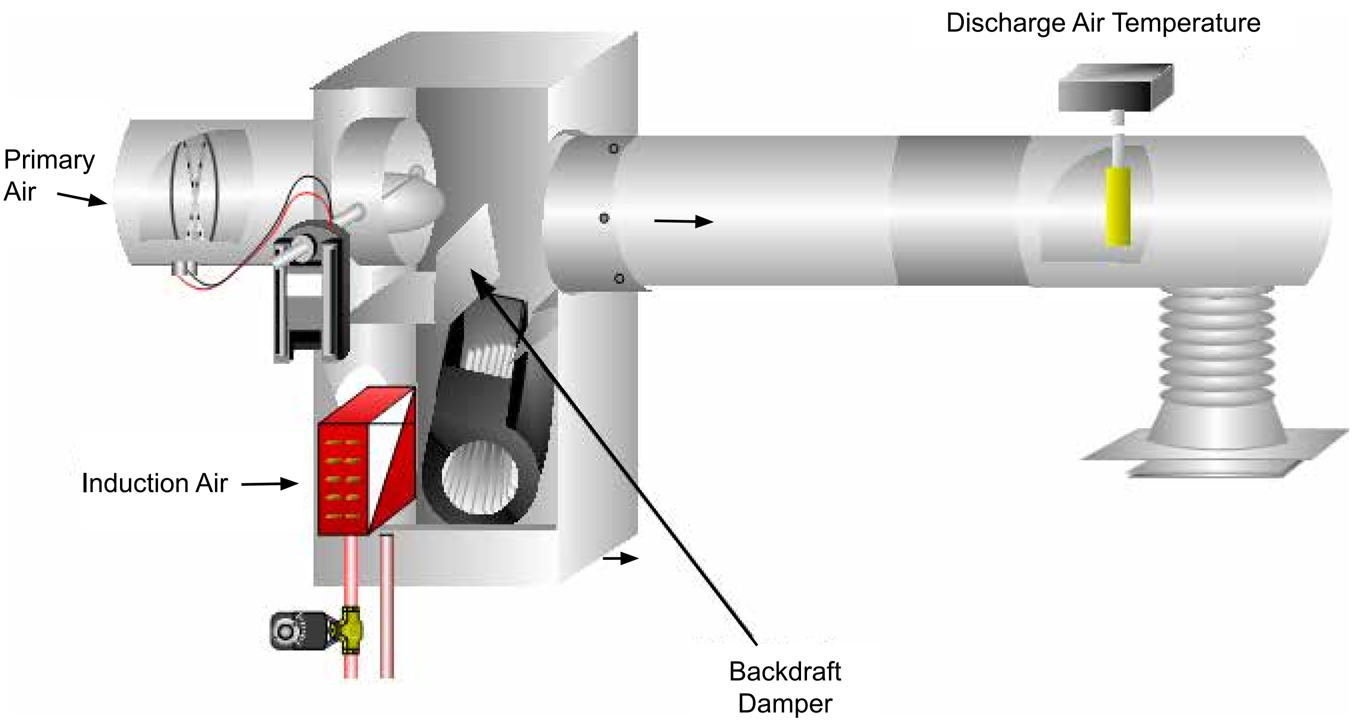


Figure 19: Parallel Fan Powered VAV Standard Features

Parallel Fan Powered VAV

Parallel fan powered terminals offer improved space comfort and flexibility in a wide variety of applications. They offer substantial operating savings through waste-heat recovery and night-setback operation.

Figure 20: Parallel Fan Powered VAV

Heat Recovery

To offset heating loads in perimeter zones, the parallel fan powered VAV recovers heat from lights and core areas. Additional heat is available at the terminal unit using terminal unit from electric, steam, or hot water heating coils. There are controls to energize remote heating devices such as a wall fin, fan coils, radiant panels, and roof load plenum unit heaters.

Sequences of Operation

In the cooling mode, the parallel fan powered VAV provides variable volume, constant temperature air, and, in the heating mode, constant volume and variable temperature air.

At the design cooling condition, while the unit fan is off, the primary air valve handles the maximum scheduled airflow capacity. As the cooling load decreases, the primary air valve throttles toward the minimum scheduled airflow capacity. A further decrease in the cooling load causes the unit fan to start, inducing warm air from the ceiling plenum. This warm air increases the discharge air temperature to the zone. When the heating load increases, the optional hot water coil or electric heater is energized to maintain comfort conditions.

Indoor Air Quality (IAQ)

The parallel fan powered VAV terminal enhances the indoor air quality (IAQ) of a building. In the heating mode, it provides higher air volumes than usually provided by single duct VAV terminals. The higher air capacity provides greater air motion in the space and lowers the heating discharge air temperature.

This combination improves air circulation, preventing the concentration of carbon dioxide (CO_2) in stagnant areas. Increased air motion also improves occupant comfort and the higher air capacity improves diffuser performance (minimizing what is sometimes called diffuser “dumping”).

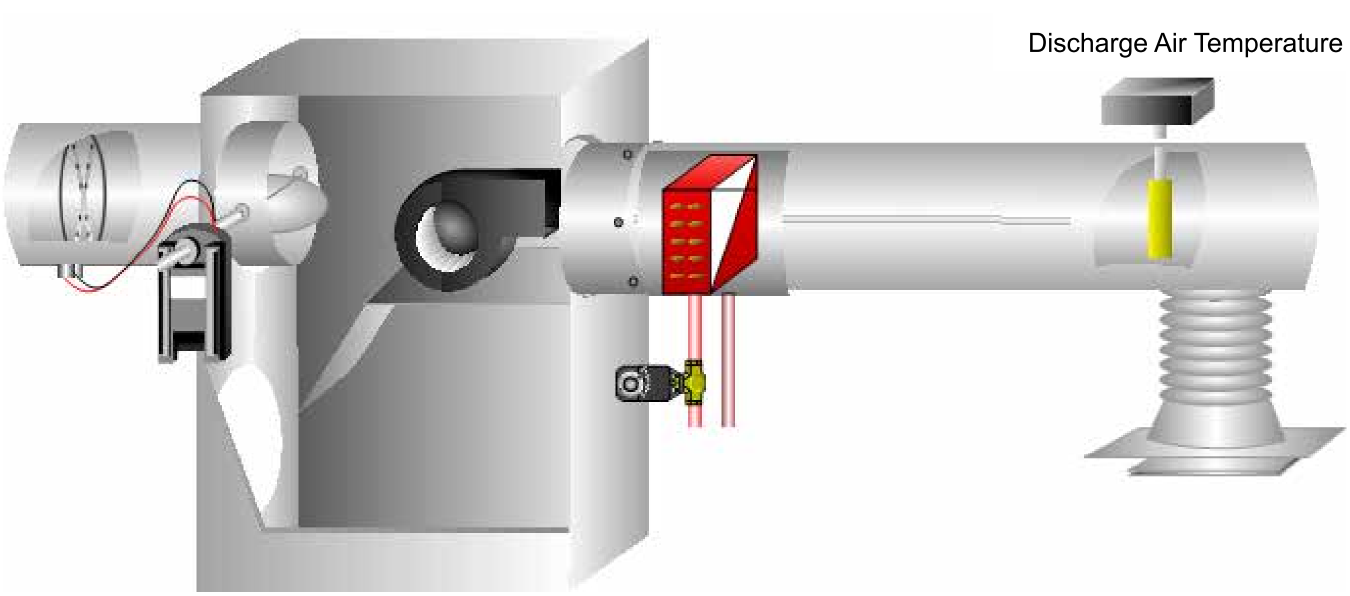
Series Fan Powered Terminal Units

Series fan powered terminals offer improved space comfort and flexibility in a wide variety of applications.

They offer substantial operating savings through waste heat recovery, a reduction of central fan horsepower requirements, and night setback operation.

LD27198

Figure 21: Series Fan Powered VAV



LD27199

Figure 22: Series Flow Fan Powered Standard Features

Heat Recovery

To offset heating loads in perimeter zones, the series fan powered VAV recovers heat from lights and core areas. Additional heat is available at the terminal unit using electric, steam, or hot water heating coils. Controls are available to energize remote heating devices such as wall fin, fan coils, radiant panels, and roof load plenum unit heaters.

IAQ

The series fan powered VAV enhances a building's IAQ. In the heating mode, it provides constant air motion and higher air volumes than usually provided by single duct VAV or parallel fan powered VAV terminals. The higher air capacity provides continuous air motion in the space and lowers the heating discharge air temperature. This combination improves air circulation, preventing the concentration of CO_2 in stagnant areas. Increased air motion also improves occupant comfort and the higher air capacity improves the diffuser performance (minimizing diffuser "dumping").

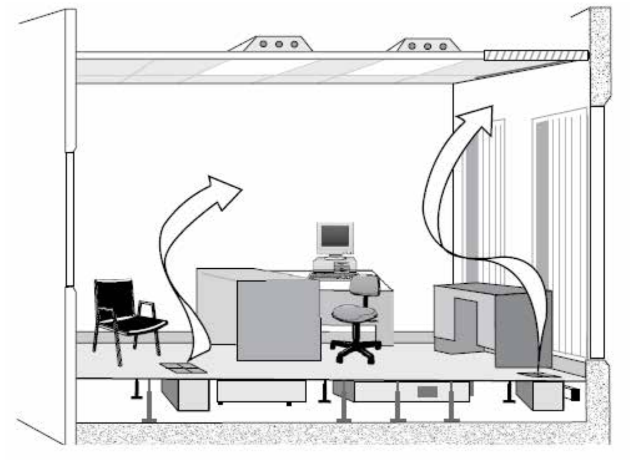
Fan Terminal Isolation

To prevent the need for additional external isolation, series fan powered VAV terminals are sufficiently equipped with internal vibration dampening. Flexible duct connectors at the unit discharge usually do more harm than good. The sagging membrane causes higher air velocities and turbulence, which translates into noise. Furthermore, the discharge noise breaks out of this fitting more than with a hard sheet metal fitting.

Underfloor

The traditional approach to HVAC design in commercial buildings has been to supply conditioned air through extensive overhead duct networks to an array of diffusers spaced evenly in the ceiling. Ceiling plenums must be large enough to accommodate the supply ducts that run through them.

RA is usually configured as ceiling plenum return without any ductwork. This type of air distribution, known as well-mixed air distribution, is the most commonly used system. This conventional HVAC system is designed to promote complete mixing of SA with room air, maintaining the entire volume of all air in the space (from floor to ceiling) at the desired space setpoint temperature.



LD27236

Figure 23: Sample Underfloor Air Delivery System

In addition, to meet IAQ requirements, an adequate supply of fresh outside air (OA) must be introduced to the mix. This control strategy has a significant disadvantage: it has no provisions to accommodate different temperature preferences among the building occupants or to provide preferential ventilation in the occupied zone.

In a good underfloor air delivery system, conditioned air from the RTU is ducted to the underfloor plenum. *Figure 23* shows this conditioned air flowing freely throughout the plenum to individual supply discharge outlets. Unlike the larger single supply duct outlets typical of overhead systems, underfloor systems are configured to have a large number of smaller supply outlets in close proximity to the building occupants.

These adjustable outlets provide an opportunity for nearby occupants to have some amount of control over thermal comfort conditions in their local environment. Air is returned from the room at ceiling level (the figure shows an unducted plenum return). The resulting overall floor-to-ceiling airflow pattern exploits the natural buoyancy of heat sources in the space; it more efficiently removes heat loads and contaminants from the space, particularly in cooling applications.

Some of the most important advantages of underfloor systems over ceiling-based systems occur during cooling conditions. In many parts of the US, cooling conditions are required year-round in the vast majority of interior office spaces.

Controls

A number of controls are important for efficient and safe rooftop unit (RTU) variable air volume (VAV) service, maintenance, and operation. See the *Service* section for information on the *Start-Up Wizard on page 86.* This section covers other important controls.

Morning Warm-Up

To start warm-up or cool down, a system can have three different settings or commands:

- Occupied
- Optimal Start
- Schedule

Occupied

Under the occupied command, the supply fan turns on for a 5-minute stabilization period and the system compares the return air (RA) temperature sensor to the warm-up setpoint. If the value is equal to or below the warm-up discharge air temperature setpoint, the system controls the heating to meet the warm-up setpoint. If the value is above the warm-up setpoint, the control goes into occupied mode.

The system continues in warm-up mode until the RA temperature sensor is above the warm-up setpoint for 5 minutes or the early start period expires. From then on, the control goes into occupied mode. For cool down mode, the system operates the same as warm-up mode, except that there are cooling stages instead of heating stages.

Schedule

Under the schedule command, during the early start period before scheduled occupancy, the supply fan turns on for a 5-minute stabilization period and the temperature control sensor is compared to the warm-up setpoint. If the value is equal to or below the warm-up setpoint, the system controls the heating to meet the warm-up discharge air temperature setpoint. If the value is above the warm-up setpoint, the control goes into occupied mode. For cool down mode, the system operates the same as warm-up mode, except that there are cooling stages instead of heating stages.

Optimal Start

Under the optimal start command, when unoccupied, the controller calculates an early start time based on the control temperature deviation from a setpoint. This ensures that the space is at conditioned levels when the occupied period starts. There are a number of conditions for the optimal start:

- Historical data determines when optimal start begins. However, users can also limit the early start time.
- The optimal start time is always within the same calendar day.
- Until the demand is satisfied, the system uses warm-up or cool down discharge air temperature setpoints. The control uses the heating or cooling setpoint to determine when the demand is satisfied. Once the demand is satisfied, the control goes into occupied mode.

Static Pressure (SP) Reset

When the fan energizes, the system uses the output from the controller to maintain the supply duct pressure to the duct pressure setpoint.

If the duct pressure is above that setpoint, the controller output decreases. If the duct pressure is below the setpoint, the controller output increases. The deviation from setpoint and length of time away from setpoint determine the rate of change of the controller output.

The system can use the local user interface (UI) or a building automation system (BAS) to adjust the duct pressure setpoint. When using a BAS, the RTU duct SP setpoint can be reset based on corresponding VAV box demand.

For systems with direct digital control (DDC) at the zone level, ASHRAE Standard 90.1 and California's Title 24 Energy Standards require the discharge air SP setpoint be reset based on the zone requiring the most pressure. To maximize energy savings, the BAS resets the following pressures according to season mode:

- In summer mode, the discharge air SP
- In winter mode, the discharge air temperature

Exhaust Air (EA) System

To control building pressure, users can select three options for power exhaust control:

- Modulating exhaust fan
- Modulating exhaust damper with fan
- External control

Modulating Exhaust Fan

If the building pressure is above the building pressure setpoint, the exhaust fan analog output increases. If the building pressure is below the building pressure setpoint, the exhaust fan analog output decreases.

The exhaust fan binary output energizes whenever the analog output is greater than 10% of the full voltage range (the percentage is adjustable). The exhaust fan binary output de-energizes whenever the exhaust fan analog output is less than or equal to 5% of the full range (the percentage is adjustable).

The system enforces a minimum deadband of 2% between the on and off setpoint. The deviation from the setpoint and length of time away from the setpoint determine the analog output's rate of change.

Modulating Exhaust Damper With Fan

To maintain the building pressure setpoint, the controller modulates the opening of the exhaust damper. If the building pressure is above the building pressure setpoint, the exhaust damper output increases to open the exhaust damper. If the building pressure is below the building pressure setpoint, the exhaust damper output decreases to close the exhaust damper.

If the exhaust damper is more than 10% open, the controller energizes the exhaust fan. If the exhaust damper is less than 5% open, the controller de-energizes the exhaust fan (these percentages are adjustable). To reduce fan cycling, a minimum deadband of 5% is enforced between the fan start and fan stop setpoints.

External Control

A third party (such as BAS communication or a voltage signal) directly controls the exhaust fan speed (also known as exhaust damper position). The exhaust fan's maximum speed is limited to be less than or equal to the supply fan's speed.

Return Air (RA) System

To control fans, users can select two options:

- Return fan discharge pressure
- Supply fan airflow tracking

Return Fan Discharge Pressure

To maintain the return fan discharge pressure to a setpoint, the controller modulates the speed of the return fan. As the supply fan speed increases or the return/exhaust dampers open, the return fan discharge pressure drops.

To maintain the pressure setpoint, the fan speed increases. As the supply fan speed decreases or the return/exhaust dampers close, the return fan discharge pressure rises. Again, to maintain the pressure setpoint, the fan speed decreases.

To maintain the building pressure setpoint, the controller modulates the opening of the exhaust damper. If the building pressure is above the building pressure setpoint, the exhaust damper output increases to open the exhaust damper.

If the building pressure is below the building pressure setpoint, the exhaust damper output decreases to close the exhaust damper. Alternatively, in this mode a third party (such as BAS communication or a voltage signal) directly controls the exhaust damper position.

Supply Fan Airflow Tracking

When equipped with optional supply and return fan airflow measuring stations, building pressure can be controlled through airflow tracking. The controller monitors the supply fan airflow and modulates the speed of the return fan to meet an return airflow setpoint. The return airflow setpoint is continuously updated to meet the supply fan airflow (minus an return airflow differential).

Economizer Control

If the economizer is installed, economizer suitability is true, and there is a call for cooling, then the unit controller modulates the OA damper between the minimum position and 100%. It does this to maintain the mixed air temperature to the discharge air temperature setpoint.

The system determines the economizer suitability based on the selected changeover option and the available sensors. If the economizer cannot meet cooling demand after the OA damper is open 100% for 3 minutes, the controller allows mechanical cooling.

Changeover Method

There are four free cooling changeover options:

- Dry bulb
- Single enthalpy
- Dual enthalpy
- Auto

When the free cooling selection variable is set to auto (the default setting), the economizer selects a free cooling method based on which temperature and humidity sensors are present and reliable. The user can manually select the free cooling type. The economizer selects the free cooling options in the following order:

1. Dual enthalpy
2. Single enthalpy
3. Dry bulb

Table 3 summarizes the free cooling sensory requirements.

Dry Bulb

When the OAT is less than the economizer's OAT enable setpoint of -1.0°F, local suitability is true. When the OAT is greater than the economizer OAT enable setpoint, local suitability is false. Free cooling is not available in the following situations:

- The OAT value is unreliable
- The OAT rises above the dry bulb setpoint

Single Enthalpy

When OA enthalpy is less than the OA enthalpy setpoint of 1 Btu/lb and the OAT is less than high limit shutoff of 1.0°F, the local suitability is true. When OA enthalpy is greater than the OA enthalpy setpoint or OAT is greater than the high limit shutoff temperature, the local suitability is false.

If either the OAT or OA humidity values are unreliable and the changeover method is manually set to single enthalpy, then free cooling is not available.

Table 3: Free Cooling Sensor Requirements

	OAT	OAH	RAT	RAH
Dry Bulb	x			
Single Enthalpy	x	x		
Dual Enthalpy	x	x	x	x

Dual Enthalpy

When OA enthalpy is less than the RA enthalpy of 1 Btu/lb and OAT is less than high limit shutoff, the local suitability is true. When OA enthalpy is greater than RA enthalpy or the OAT is greater than high limit shutoff, the local suitability is false.

If the changeover method is manually set to dual enthalpy and the OA or RA temperature or humidity values are unreliable, free cooling is not available.

Auto

Auto selects the changeover method based on which sensors are present and reliable. If the OA or RA temperature and humidity values are reliable, auto uses the dual enthalpy changeover method. If either of the RA temperature or humidity sensors is unreliable, auto uses the single enthalpy changeover method. If the OA humidity sensor is unreliable, auto uses the dry bulb changeover method.

Independent Twinning

Twinning is when two or more separate RTUs are tied into one common main duct trunk line. Multiple units can be twinned based on the BAS communication protocol. The twinning process allows redundancy with the cooling/heating load. All twinned units operate concurrently. Their sequence is not based on the cooling or heating load.

This operation allows the user to design the system in an arrangement of two or more units (N+1). For example, with four units (N+1), three units (N) are sized to handle the load and the fourth (+1) is available as a backup. The user can then shut down one of the running units and enable the backup via the BAS.

To ensure that the damper is fully open before the supply fan can start, each RTU requires an isolation damper with an end switch. In a twinning application, each RTU also requires its own duct SP sensor.

This allows the RTU to run independently if one or more units are shut down for maintenance or BAS communication is lost. To prevent over pressurization of the duct, each unit has a manual reset high duct pressure switch installed.

In a twinned system, several features are synchronized while others can operate independently. This process is known as independent twinning. Each twinned RTU in the group broadcasts the key values to the other RTUs in the group so that they can use it. If each unit has the exact same values, each RTU can execute the exact same control algorithm.

The advantages of this process are that it is more robust against communication errors and sensor failures, and less complex than a master and dependent unit arrangement. If one independent unit fails or manually shuts down, the remaining units continue running without interruption. If the master in a master and dependent unit arrangement fails or manually shuts down, the failover logic to a different master is more complicated and can lead to system disruption.

The synchronized features include supply fan control, economizer suitability, occupancy, demand control ventilation (DCV), exhaust fan control, and smoke control.

Supply fan control must be synchronized to maintain discharge air SP between all units serving the same duct. Synchronization requires all reliable discharge air SP values from the RTUs be averaged before passing to the proportional-integral-derivative (PID). An RTU's SP setpoint must also synchronize and any change to one setpoint must be shared with the other RTUs. This allows the PID in each RTU to calculate the same output value and run all the supply fans at the same speed.

During start-up of a previously shutdown RTU, the supply fan speed slowly ramps up until it matches the fan speeds of the RTUs that are currently operating. When the additional RTU begins ramping, the SP increases. This increase causes the other RTU fans speeds to slow down to reach the setpoint. Once the RTU fan speed matches the existing RTUs, it is released into control.

Economizer suitability, when available, allows all RTUs to use the OA damper for free cooling. This synchronization avoids a situation where one unit is operating in economizer mode while another unit operates in mechanical cooling mode.

Occupancy, for twinned units, allows all units to switch between occupied and unoccupied modes simultaneously. This includes the occupancy schedule and the warm-up/cool down (if enabled).

DCV allows the OA damper minimum position to be reset equally between the RTUs. To ensure sufficient OA is brought in for proper ventilation, this requires the indoor CO_2 values from the RTUs to be shared and the maximum to be passed to the reset logic. For the reset calculation to be the same across all units, the indoor CO_2 setpoint also synchronizes.

Exhaust fan system allows for proper building pressurization. This requires the building SP values from each unit to be averaged before being passed to the PID. The building SP setpoint also synchronizes and any change to one setpoint is shared with the other RTUs.

Smoke control ensures that all units properly switch between the purge, pressurization, or depressurization modes. This requires the three smoke-purge binary inputs and their priorities on each unit be shared. As a result, if any binary input is on, all twinned units respond the same way.

On each unit, the temperature control, return fan systems, and safety shutdowns can operate individually. The temperature setpoints synchronize to allow for the same operation between the units. Temperature loops are not required to be as tightly coupled as supply and exhaust fan systems.

Twinning has the following additional requirements:

- It only works for VAV systems.
- All units are the same size.
- Fan systems have tuning disabled.
- External control is not supported.
- The same SP probe can be connected to the following transducers:
    - All discharge pressure transducers
    - All building pressure transducers

Components and Operation

RTUs include the following main components and operations:

- Refrigeration
- Fans and airflow
- Ventilation

RTUs also have other important components and accessories, such as humidifiers, coil coating, and shaft grounding.

Refrigeration

The principles of refrigeration are based on the relationship between a liquid's boiling point and pressure. Raising a liquid's pressure raises the liquid's temperature and lowering its pressure lowers its boiling temperature. Under this principle, direct expansion (DX) refrigeration uses a refrigerant with a very low boiling point. The common refrigerant is R-410A. At standard atmospheric pressure, R-410A boils at -55.0°F. Refrigeration has a number of stages:

1. When there is a call for cooling, the compressor begins to run, which raises the pressure of the refrigerant vapor. The raised pressure on the vapor raises its temperature, resulting in a hot refrigerant gas.
2. The hot gas flows through the condenser coils.
3. Ambient air blows across the outside of the condenser coils. This removes heat from the hot gas and, at this high pressure, lowers its temperature below its condensing (boiling) point.
4. The hot refrigerant gas changes into a high pressure refrigerant liquid.
5. The high pressure liquid exits the condensing unit and flows to the expansion valve at the evaporator coil. There are two functions of the expansion valve:
    - Lower the pressure of the liquid refrigerant to the equivalent of a 40.0°F boiling point.
    - Match the flow of refrigerant to the heat load on the evaporator.
6. The supply air (SA) fan pulls warm air (usually 75.0°F) from the occupied space over the evaporator coil. The refrigerant absorbs the heat from the air, which causes the refrigerant to boil back into a gas and lowers the air temperature to 55.0°F.
7. The SA fan delivers the cool air to the occupied space. The now-gaseous refrigerant returns to the compressor to start the cycle again.

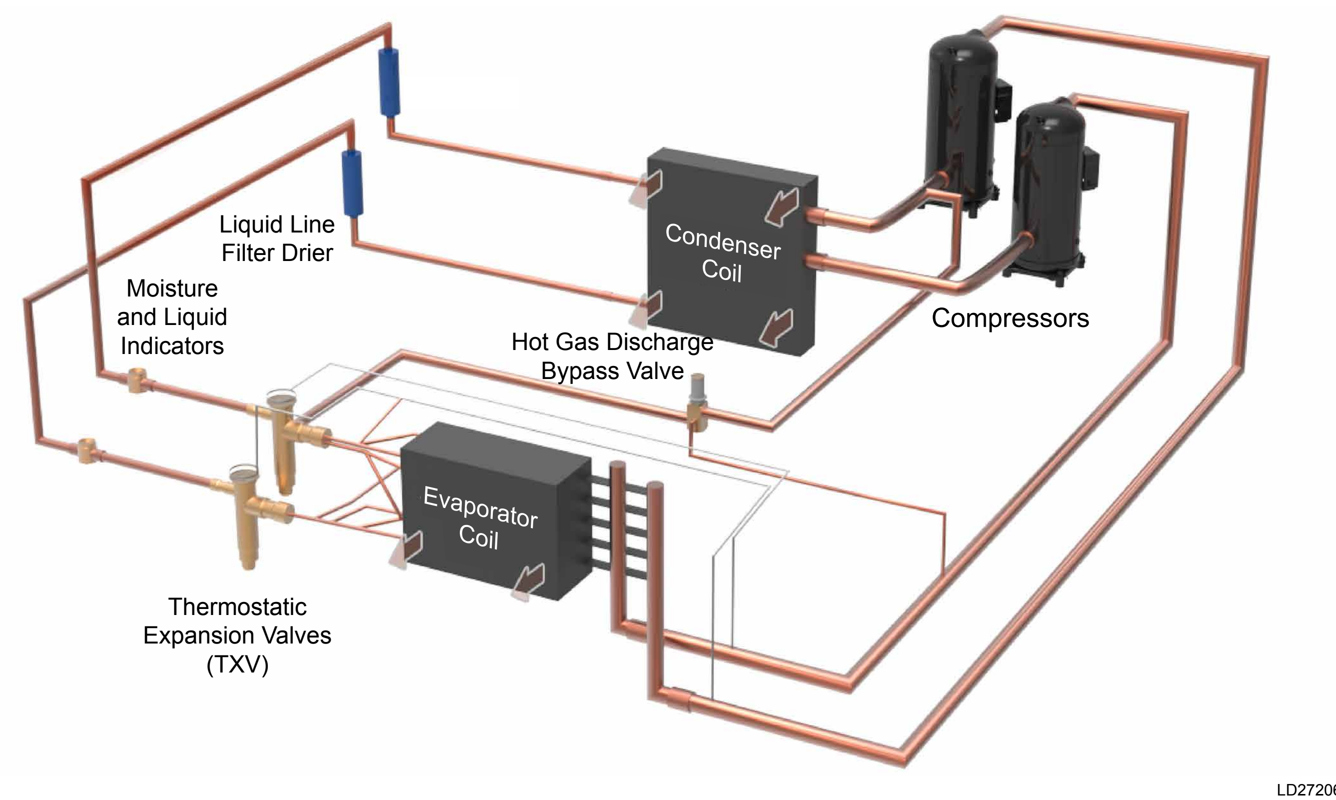


Figure 24: Refrigeration Circuit

Pressure Transducers

Refrigerant pressure transducers monitor system pressures for diagnostic purposes and to maintain operation of the refrigeration system within its operational envelope. These transducers ensure that the compressors always operate within their manufactured specifications and protect them in the event of a component failure.

Pressure transducers are usually installed on the refrigerant piping lines to display the refrigerant pressure during the unit's operation. The system can then convert the pressure reading to temperature, providing additional information and feedback to the system controller.

A pressure transducer has two main parts:

- An elastic material that deforms when exposed to a pressurized medium
- An electrical device that detects the deformation and converts it into a usable electrical signal. This electrical signal is an input signal for the controller to display the pressure inside the refrigerant line

The measured pressure value can be compared with the setpoints of the controller for various modes of operation, for example, various modes of operation, such as compressor sequencing, low ambient operation, hot gas reheat (HGRH), hot gas bypass (HGBP), and heating modes.

Multiple Cooling Stages

Large RTUs have multiple cooling stages. These stages are independent. Each stage has its own compressors, condenser coil, and evaporator coil. The condenser coils and evaporator coil sections nest together with similar components of the other stages.

By using multiple cooling stages, the RTU control system can match the system capacity with the comfort cooling load. This maximizes unit efficiency. Another benefit of the independent stages is redundancy. If one stage leaks refrigerant and goes down, the RTU can continue to operate at a reduced overall capacity until the system leak is repaired.

For example, a typical 150 Ton RTU has three separate refrigeration systems, each with a tandem compressor. By utilizing compressors with different capacities that can energize independently, the 150 Ton RTU has 14 stages of cooling, with a minimum capacity of 17 tons. See *Table 4*.

Table 4: 150 Ton Staging

Step	System 1	Compressors	System 2	Compressors	System 3	Compressors	% of Total Capacity
	One	Both	One	Both	One	Both	
1	On	Off	Off	Off	Off	Off	11
2	Off	Off	On	Off	Off	Off	15
3	Off	Off	Off	Off	On	Off	24
4	Off	Off	On	On	Off	Off	30
5	On	On	On	Off	Off	Off	37
6	On	On	Off	Off	On	Off	46
7	On	Off	On	Off	On	Off	50
8	Off	Off	On	On	On	Off	54
9	On	Off	Off	Off	On	On	59
10	Off	Off	On	Off	On	On	63
11	On	On	Off	Off	On	On	70
12	Off	Off	On	On	On	On	78
13	On	Off	On	On	On	On	89
14	On	On	On	On	On	On	100

Thermostatic Expansion Valve (TXV)

A thermostatic expansion valve (TXV) is a refrigeration device used to regulate the flow of liquid refrigerant injected into a DX evaporator. Proper flow of refrigerant is important to avoid liquid floodback to the compressor and to achieve highest system efficiency.

The TXV does not regulate other variables, such as air temperature, humidity, head pressure, suction pressure, or capacity. According to Lanzer (2013), using the TXV to control any of these system variables leads to poor system performance and can lead to compressor failure (Lanzer 2013).

In a refrigerant circuit, the TXV separates the high pressure and low pressure sides. As the TXV restricts liquid refrigerant flow to the evaporator, the refrigerant pressure is reduced.

The TXV is able to maintain precise refrigerant flow by matching the flow of refrigerant into the evaporator with the refrigerant's evaporation rate.

The TXV responds to both the temperature of the refrigerant vapor leaving the evaporator and the pressure in the evaporator. In response to this temperature and pressure, a movable valve pin is used to precisely control the refrigerant flow. As the valve pin restricts the flow of the liquid refrigerant, the following occurs:

- The liquid refrigerant pressure decreases.
- As a result of decreased pressure, a small amount of the liquid refrigerant converts to gas.

  This converted gas, also known as “flash gas”, represents a high degree of energy transfer because the refrigerant's sensible heat has been converted to latent heat.
- The low pressure liquid/vapor combination moves into the evaporator. The remaining liquid refrigerant absorbs heat and evaporates into its gaseous state (Lanzer 2013).

Lanzer (2013) expresses the TXV pressure balance with the following equation:

$$P1 + P4 = P2 + P3$$

With the following values:

- P1 = Bulb Pressure (Opening Force)
- P2 = Evaporator Pressure (Closing Force)
- P3 = Superheat Spring Pressure (Closing Force)
- P4 = Liquid Refrigerant Pressure (Opening Force)

Electronic Expansion Valve (EXV)

Similar to the TXV, the electronic expansion valve (EXV) controls the flow of liquid refrigerant entering a DX evaporator.

An EXV’s operation is more advanced than a standard TXV. The EXV uses a step motor to open and close the valve port in response to signal sent from an electronic controller. Unlike a typical motor, step motors are driven by a gear train and rotate in steps rather than continuously. The motor rotates a fraction of a revolution (a step) for each signal received from its electronic controller (Tomcyzk 2004).

Capable of moving 200 steps per second, the motors allow quick adjustment and recovery to their exact position. The controller continuously monitors the number of step signals received from the controller, allowing the valve to return to any previous position at any time. Most EXVs have 1,596 steps of control and each step is 0.0000783 inch (Tomcyzk 2004).

For a cutaway of an EXV with step motor and drive assembly, refer to Figure 2 in Tomcyzk's (2004) article.

The electronic signals sent by the controller to the EXV are commonly through a resistor known as a thermistor. This thermistor is connected to discharge airflow in the outlet of the evaporator. The thermistor changes resistance as its temperature changes. Other sensors are commonly located at the evaporator inlet and outlet to sense evaporator superheat. Similar to the TXV, this sensing protects the compressor from any liquid floodback under low superheat conditions (Tomcyzk 2004).

It is possible for the controller to open the EXV too much and cause overcooling. In this case, some applications may include combination of refrigerant floodback protection sensors and a discharge air temperature thermistor. This combination of sensors, also known as a feedback loop, allows the system to sense overcooling and feed information to the electronic controller and the EXV. The step motor responds by closing the valve, allowing less liquid refrigerant into the evaporator (Tomcyzk 2004).

For an illustration of a feedback loop, refer to Figure 3 in Tomcyzk's (2004) article.

Hot Gas Reheat (HGRH)

In areas of high sensible load and hot weather, an RTU is required to operate for a higher number of hours. During a cooling cycle, a standard RTU removes high levels of moisture from the outside air (OA) that enters the building.

Modern buildings are constructed in tighter spaces with higher demand for ventilation air. Each additional amount of ventilation air draws additional moisture into the space. An RTU reduces this moisture in a standard cooling mode.

Humidity levels need to be maintained at all times. Proper humidity control saves cooling costs, improves human comfort, reduces the potential for mold, and improves IAQ.

However, the standard RTU only comes on when there is a call for cooling, which is infrequent on mild temperature days. During mild temperature days, the RTU uses an economizer to take advantage of the cooler OA. This reduces the indoor temperature but does not satisfactorily dehumidify the air. The unit keeps turning on and off. The unit tends to overcool the space when it runs for longer hours to meet indoor humidity.

ANSI/ASHRAE Standard 55 relates human comfort to temperature and humidity levels and establishes a range of temperatures and humidity levels that are considered comfortable by 80% or more of the test subjects (ASHRAE 2017a) .

The standard requires that systems designed to control humidity must be able to maintain a dew point temperature of 62.2°F (16.8°C). A more specific range can be determined from the standard but depends on relative humidity, season, clothing worn, activity levels, and other factors.

The standard notes that HVAC systems must be able to maintain a humidity ratio at or below 0.012. This ratio corresponds to an upper relative humidity level equal to or greater than 80% at low dry bulb temperatures. However, the level may be lower depending on factors such as temperature or other factors mentioned previously. The standard does not specify a lower humidity limit. It notes that non-thermal comfort factors (for example, dry skin, dry eyes, mucus membrane irritation, and static electricity) may place limits on the acceptability of very low humidity environments.

Furthermore, ANSI/ASHRAE Standard 62.1 recommends controlling relative humidity in occupied spaces to less than 65% for mechanical systems with dehumidification capability. This reduces the likelihood of conditions leading to microbial growth (ASHRAE 2016b).

Wherever moisture is a problem, the system uses HGRH. The following buildings are examples of common spaces using HGRH:

- Offices
- Schools
- Auditoriums
- Restaurants
- Places of worship
- Health clubs
- Museums
- Libraries
- Theaters
- Computer rooms
- Data centers
- Indoor swimming pools

Ideally, the indoor humidity of such spaces is maintained around 50% of relative humidity. However, this may vary by application.

HGRH Refrigeration Cycle

In an occupied space, the system monitors the temperature through a wall-mounted thermostat. It monitors the humidity levels through a humidistat. It prevents the space from overcooling by using the HGRH option to maintain space temperature and the dehumidification mode to maintain humidity.

When the SAT reaches the setpoint and the humidity is above the setpoint, the reheat valve energizes. The unit enters reheat mode, going through the following stages:

1. The system cools and dehumidifies.
2. The system reheats the air using hot refrigerant gas.
3. The system delivers the air to the space, usually 2.0–5.0°F below room temperature.

Reheating the air along a constant sensible heat line reduces the relative humidity of the leaving air. Traditional methods of reheating air with hot water coil, electric, or gas cannot match this method's energy savings.

The refrigeration system is comprised of a reheat coil and a HGRH bleed valve. In cooling and heating modes, it works as a standard air conditioner or RTU. During the reheat mode, the compressor discharge gas diverts through the HRGH bleed valve to the reheat coil. The reheat coil is located downstream of the cooling coil. The superheated refrigerant gas reheats the cooling coil's leaving air.

A HGRH bleed valve is a solenoid valve that connects the reheat coil to the suction line. When dehumidification mode is inactive, this valve's function is to bleed off any remaining or trapped liquid in the reheat coil. When the unit enters dehumidification mode, the HGRH bleed valve closes.

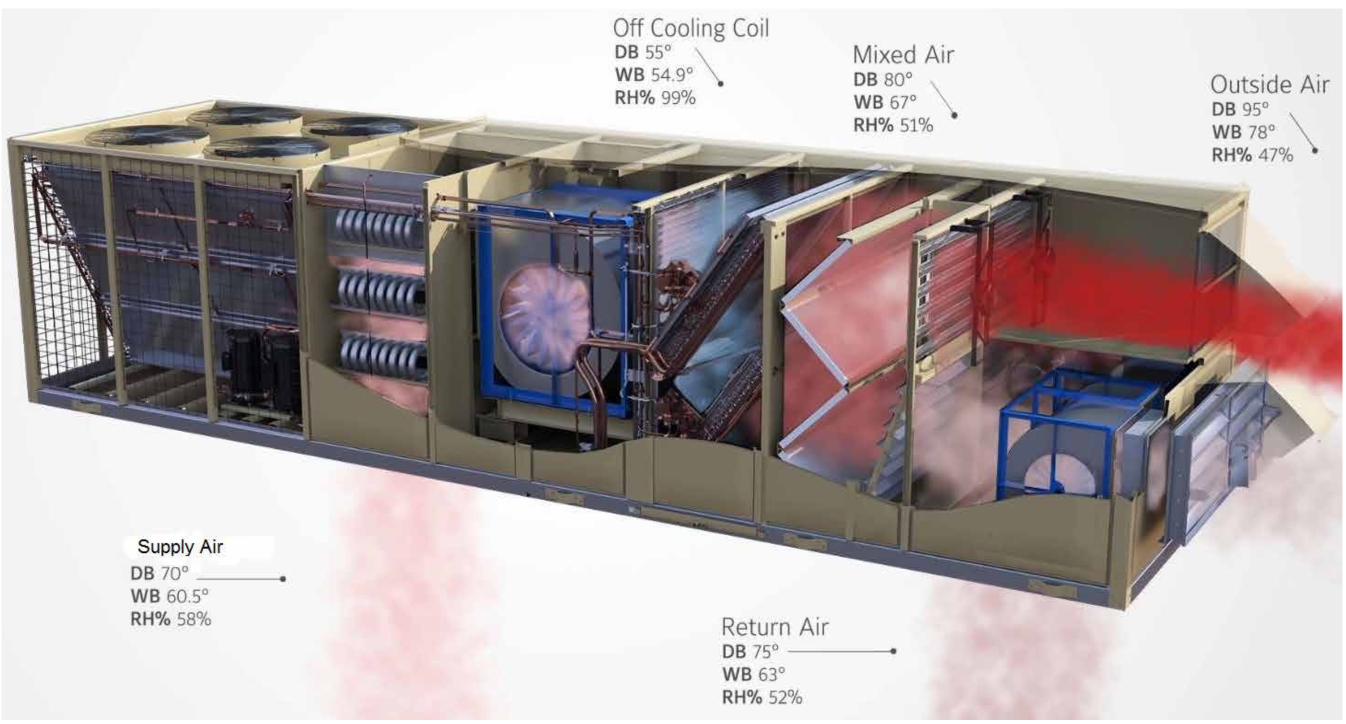


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Figure 25: Hot Gas Reheat (HGRH) – Cutaway

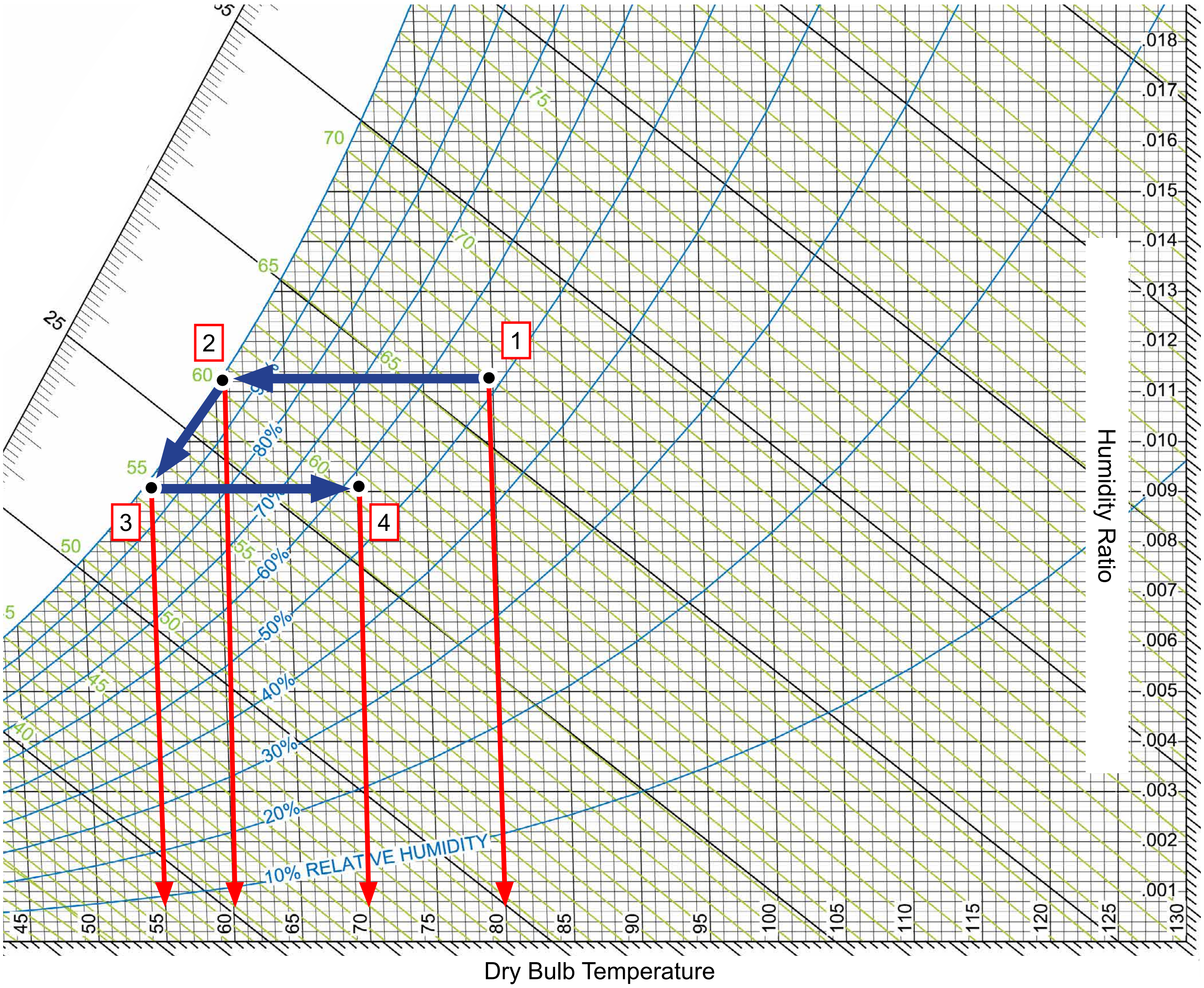

Figure 26: HGRH Psychometric Chart

Figure 26 shows an example of the HGRH process over four phases of the system.

Phase 1–2: The dry bulb temperature is 80.0°F and decreases to 60.0°F. The humidity ratio line remains the same throughout.

However, the relative humidity level rises because the cool air has less capacity to hold water. As the dry bulb temperature decreases, the corresponding absolute humidity at saturation decreases. The cooler air has less capacity to hold moisture than warmer air.

The relative humidity percentage is calculated with the following equation:

$$\frac{\text{Absolute humidity at temperature}}{\text{Absolute humidity at saturation}} \times 100$$

The result is a rise in humidity levels (from X% to Y%). This process follows the principle of saturation.

Phase 2–3: The dry bulb temperature decreases to 55.0°F. The unit enters dehumidification mode. Moisture content in the air is removed. There is a decrease in the humidity ratio.

Phase 3–4: The dry bulb temperature increases to 70.0°F. The removed moisture leads to a decrease in the humidity ratio while maintaining relative humidity. The unit reaches acceptable conditions of temperature and humidity.

Control Means for HGRH

Units with multiple compressors are controlled using a staging sequence. As the demand for cooling reduces, the compressor staging decreases. Alternatively, as demand increases, the compressor staging increases. When the desired cooling setpoint is met, compressors turn off in sequence. In the dehumidification mode, in the indoor space, the reheat coil along with the compressors maintain the temperature and humidity conditions.

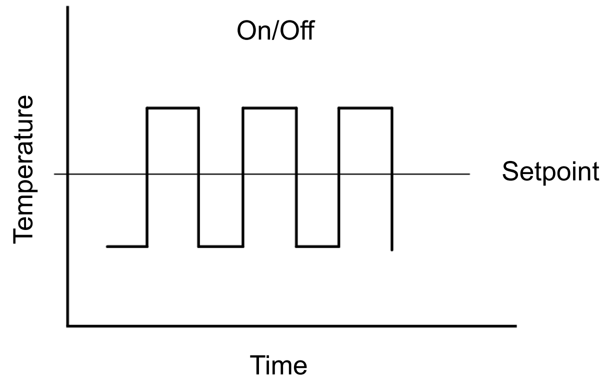

Figure 27: On/Off HGRH

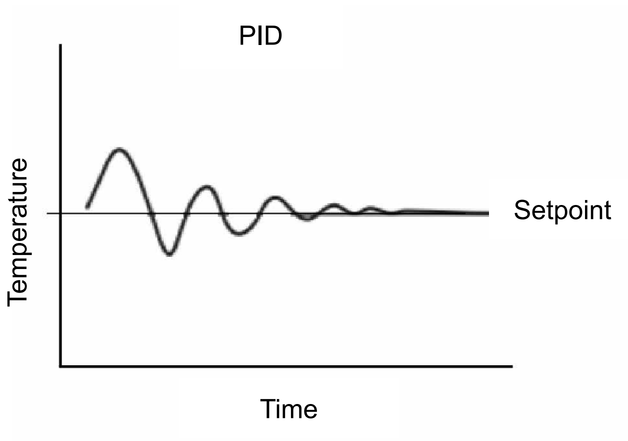

Figure 28: Modulating HGRH

Two combinations are currently available in the market:

1. **Staged compressors and on/off reheat:** During dehumidification, air reheating occurs with the reheat coil. To maintain the desired indoor temperature conditions, the leaving air is reheated as it leaves the unit. Over a period of time, as the system uses an on/off setting to control the reheat coil, the temperature fluctuates in the indoor space.
2. **Staged compressors and modulating reheat:** Units with modulating HGRH can vary the reheat coil temperature. These units are more capable of reacting to fluctuations in indoor temperature and can better maintain the desired conditions.

Unoccupied Setpoint for Reheat

In an unoccupied mode, an unoccupied setpoint helps save energy for dehumidification. During unoccupied hours, many system designs greatly reduce or even eliminate fresh air makeup. This design lessens the need for reheat. However, the *ASHRAE Handbook – HVAC Application* states: “In humid climates, seriously consider providing dehumidification during the summer, even if school is unoccupied, to prevent mold and mildew” (Owen 2015, 73).

The controller logic should contain an unoccupied setpoint that can be used for periods of desired unoccupancy. This setpoint enables the unit to satisfy the desired humidity levels even when there is less or no fresh air. The unoccupied command can be enabled either by a user interface (UI) or building automation system (BAS).

Hot Gas Bypass (HGBP)

HGBP is installed in DX equipment to reduce coil freeze-up and keep the system operating longer. HGBP works by allowing a percentage of hot discharge gas to bypass the condenser and go directly into the evaporator. A valve is used to regulate the amount of bypass refrigerant. The disadvantage of HGBP is that the operation creates an inherently inefficient operation.

HGBP diverts hot gas from the discharge of the compressor to the evaporator coil or suction to falsely load the system. This strategy can keep a compressor running longer. A longer running compressor keeps the evaporator coil active and keeps the space more comfortable in high humidity situations. This strategy also keeps the evaporator coil from freezing in low airflow situations, such as a VAV application with many closed boxes and the RTU supply fan running at low speed).

Generally in comfort-cooling RTU applications, HGBP valves deliver hot gas to the inlet of the evaporator. The valves measure and maintain a desired suction pressure. As suction pressure falls, the HGBP valve opens to admit more hot gas to the evaporator. When additional hot gas is delivered to the evaporator, the expansion valve senses the higher temperature at its sensing bulb, which is located on the suction line. The expansion valve opens to deliver more refrigerant.

The compressor must work harder to deliver the additional refrigerant. Therefore, the system runs longer. The additional hot gas in the evaporator also keeps the coil from freezing. Both mechanical and electronic HGBP valves are available.

The HGBP valve induces a parasitic load on the system. ANSI/ASHRAE/IES Standard 90.1 limits the use of HGBP to systems that have multiple steps of refrigeration system capacity control. HGBP is not allowed in systems with only one on/off compressor (ASHRAE 2016a, sec. 6.5.9).

ASHRAE (2016a) also limits the HGBP’s valve size to no larger than the following percentages of total system capacity:

- 15% for units up to 20 tons
- 10% for units over 20 tons

Variable Speed Compressors (VSC) versus Staged Compressors

The compressor is the heart of the refrigeration system. It is also the largest single electrical load in the refrigeration system. An air conditioning system is sized based upon worst case conditions of ambient temperature and cooling load. As a result, they are oversized for most operating conditions, resulting in an energy penalty at lower loads.

The most common type of compressor for comfort cooling is a single stage compressor. A single stage compressor operates at a single speed. There is no means to adjust the compressor capacity. At low cooling loads, the energy required to operate the compressor is much greater than a compressor selected at those low loads would require. At low loads, the compressor cycles on and off. This cycle can result in swings in room temperature and an inability to run long enough for dehumidification.

To help match compressor capacity to the load, manufacturers developed the staged compressor. Staged compressors offer two or more discreet capacities, enabling the compressor to more closely match the load and reduce energy consumption. The compressor can turn off a portion of its capacity based on suction and discharge pressures.

By more closely matching the load, the compressor can operate longer at low cooling loads. This increased operation time reduces temperature swings and enables better dehumidification. Staged compressors also reduce the overall compressor power draw, resulting in energy savings compared to a single stage compressor.

Staged compressors are a clear improvement over single stage compressors. However, they are still limited because of their discrete stages. The variable speed compressor (VSC) is an improved compressor design. An invertor motor drives the VSC, enabling a much larger variation in speed between the lower limit and full capacity. This variation enables a very precise match between compressor capacity and cooling load while simultaneously reducing the compressor's overall power consumption across the operating range.

The VSC enables the system to run longer to match cooling load, enables better dehumidification, and reduces the overall energy consumption. This raises the integrated energy efficiency ratio (IEER) rating. As a result, while there is a penalty in initial cost, in most cases it is offset by energy savings over the equipment's lifetime.

Fans and Airflow

Along with coil and refrigeration selection, fan selection is one of the most important aspects of good RTU design. Poor fan selection can lead to an inefficient operating system, unwanted noise, or premature damage to the fan and motor. Before the specifics of fan selection, it is important to understand the basics of a fan curve.

Figure 29 on page 41 shows the basics of a standard fan curve:

- **System Lines** – The green lines indicate the tested ratings of a given fan size and type independent of a specific system. They are noted in a percent relation to fully wide open (WO) CFM. A system with zero static pressure (SP) (y-axis) has a fully WO CFM.
- **Design System Curve –** Under the SP Curves equation, the orange line shows a system that remains unchanged.
- **Brake HP Lines –** The maroon lines assist the selection of the correct motor size. For a set system, BHP is a cubic relation to airflow:

  $BHP_1/BHP_2 = (CFM_1/CFM_2)^3$
- **Operating Point** – The design operating point is indicated by plotting the design CFM against the design SP on the curve.
- **Operating Point Section** – This shows design data for a given airflow and total SP.
- **Do Not Select in This Region** – This shows the surge or stall region for a given fan type and size.
- **Static Pressure Curves** – The blue lines are a plot of the various CFM and SPs for a given fan size and type at a set speed (RPM). For a set system, SP is a square relation to airflow:

  $SP_1/SP_2 = (CFM_1/CFM_2)^2$
- **RPM** – This represents the fan RPM, not the motor RPM. However, in direct drive applications, both values are equal. For a set system, the airflow is directly proportional to fan RPM:

  $RPM_1/RPM_2 = CFM_1/CFM_2$
- **Outlet (OV) –** This is the velocity (in FPM) of the air at the fan outlet
- **Fan Limits Section** – This shows the operating limits for the given fan type and size selected.
- **Max RPM** – This shows the maximum speed the fan can turn at the selected fan class.
- **Max TSP** – This shows the maximum total SP for the given fan type and size selected.

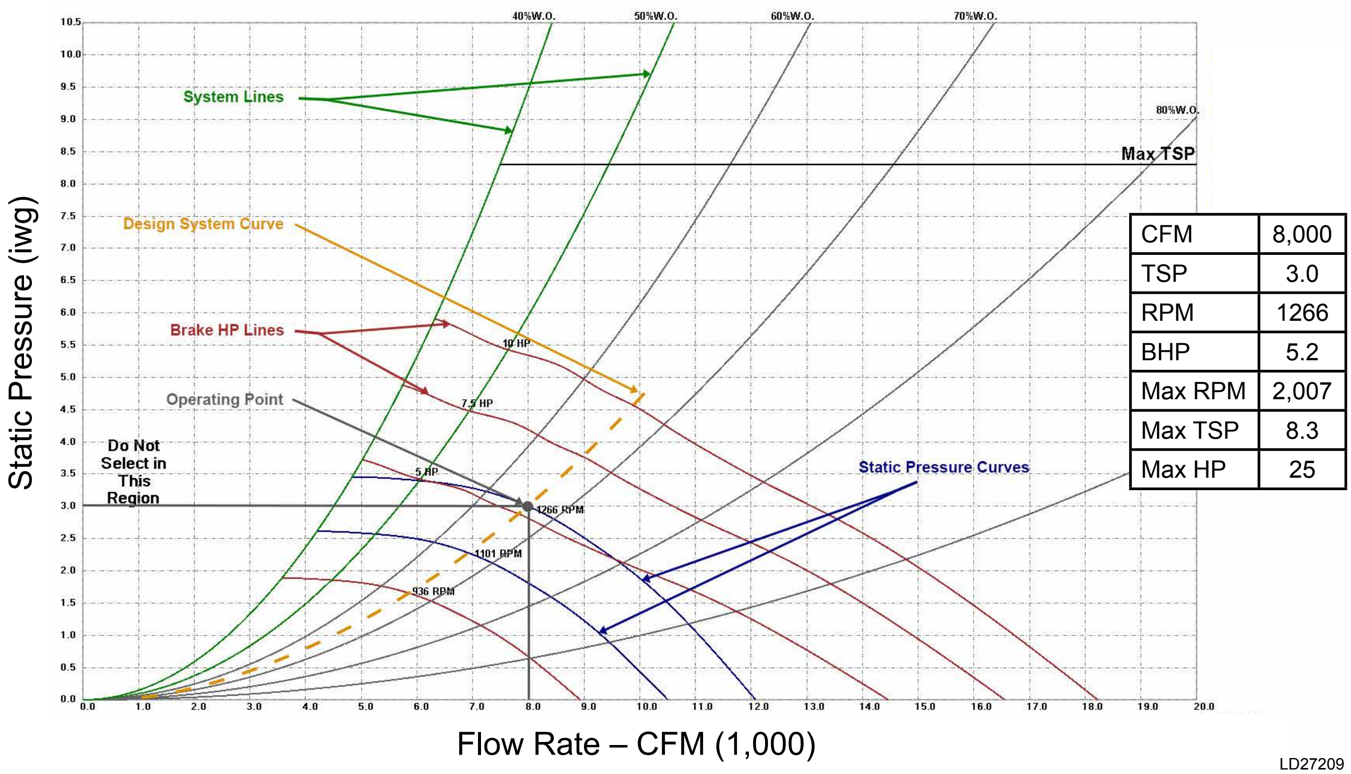

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Figure 29: Fan Curve Basics

- **Cabinet Limits Section** – This shows the limitations of the selected cabinet size.
- **Max HP** – This shows the maximum nameplate motor HP that is pre-engineered with the given fan type and size in the selected cabinet size.

Fan Laws

Fan laws are a series of equations for calculating the performance characteristics of a specific fan type and size. Fan laws in relation to measured data or a generated fan curve can be used to calculate fan speed (RPM), airflow (CFM), SP, and BHP at conditions other than those at which the data was collected.

Fan laws allow the user to calculate various points on a fixed system curve. The system curve expresses the pressure, or friction, losses associated with a specific system. The system generally includes the RTU internal losses, ductwork losses, and diffusers and grilles. With a fixed system, the effects of RPM change can be calculated and plotted on the system curve using fan laws:

CFM – Directly proportional to the RPM:

$$CFM_2 = CFM_1 \times (RPM_2/RPM_1)$$

SP – Varies as the square of the RPM:

$$SP_2 = SP_1 \times (RPM_2/RPM_1)^2$$

BHP – Varies as the cube of the RPM:

$$BHP_2 = BHP_1 \times (RPM_2/RPM_1)^3$$

Note that the fan laws calculate data on a fixed system curve. Before using the measured data in the fan law equations, it is important to verify that a system has not changed. Any resistance change in the system changes the system curve, such as with dirty filters, change in the ductwork, or a closed damper.

For the following examples of applied fan laws, see *Figure 30 on page 42*.

Consider a system designed with the following values:

CFM = 8,000

RPM = 1,186

Motor Nameplate HP = 7.5

SP = 2.5 iwg

BHP = 4.51

Given the design data, the system curve is drawn by using fan laws to calculate various points. Combining fan law equations yields the following equation:

$$SP_2 = SP_1 \times (CFM_2/CFM_1)^2$$

Using design data as the base, multiple points can be generated to define the design system curve (in red):

$CFM_1 = 5{,}000$

$CFM_2 = 6{,}000$

$CFM_3 = 9{,}000$

$SP_1 = 0.98$ iwg

$SP_2 = 1.41$ iwg

$SP_3 = 3.16$ iwg

Assume the unit is installed. However, during construction the contractor deviates from the designed ductwork and the external SP increases. The original design data cannot be used in the fan law equations to generate additional data. As a result, the system curve has changed and a new system curve must be generated.

The balancing report indicates the new operating data:

CFM = 8,000

RPM = 2,100

Motor Nameplate HP = 7.5

SP = 3.5 iwg

BHP = 6.24

Following the same process above, we can generate the new system curve (the purple line labelled 'New System Curve').

Consider the following scenario: after two years, a customer wants to increase airflow to 9,000. Assuming the system has not changed, the fan laws can be used to check if this increase is possible.

The fan RPM is verified to ensure that it does not exceed the maximum for the fan class (in this case, 1,500 RPM):

$RPM_2 = RPM_1 \times (CFM_2/CFM_1)$

$RPM_2 = 1{,}322 \times (9{,}000/8{,}000)$

$RPM_2 = 1{,}487$

Next, the BHP requirements are checked:

$BHP_2 = BHP_1 \times (RPM_2/RPM_1)^3$

$BHP_2 = 3.5 \times (1{,}487/1{,}322)^3$

$BHP_2 = 8.88$

In this case, the fan RPM is suitable. However, the original 7.5 HP motor is not large enough to accommodate the increase. If the customer wishes to increase CFM, the motor must be increased to a 10 HP motor.

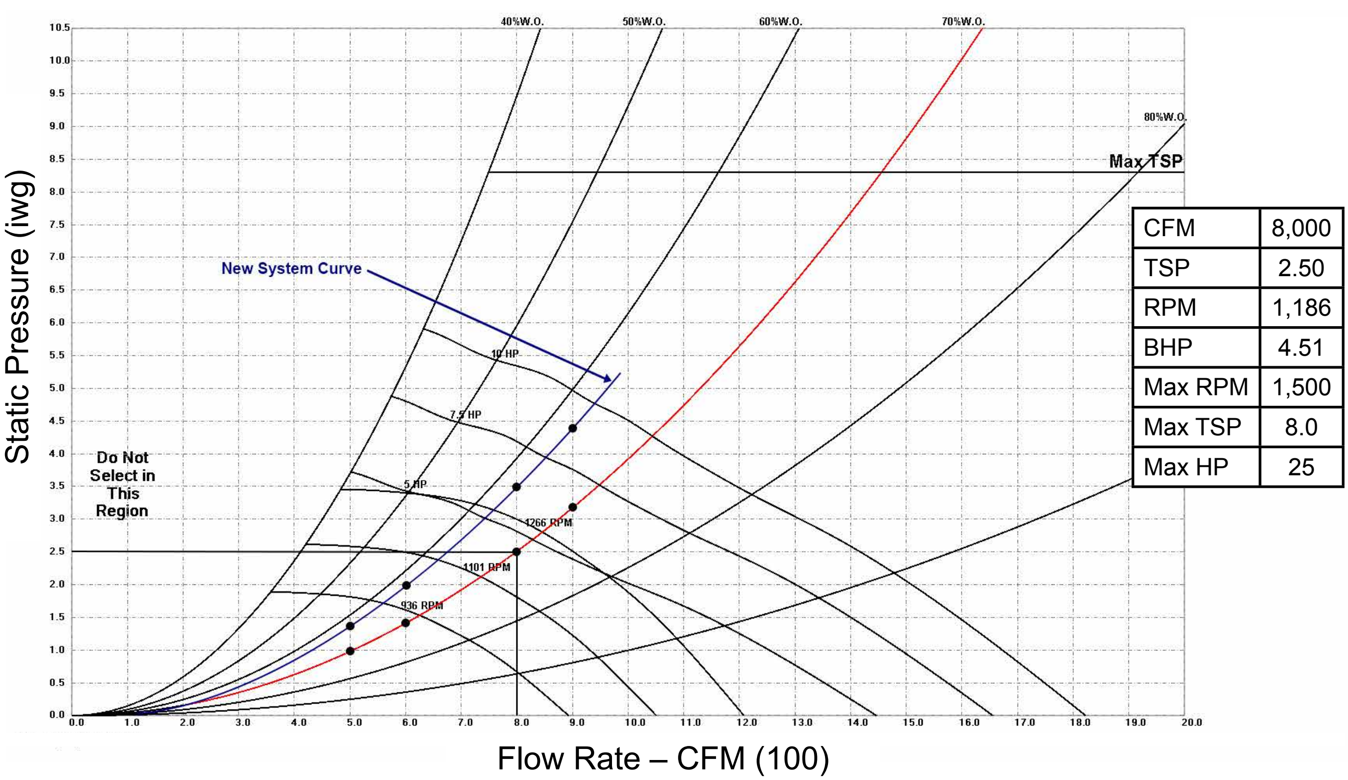

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Figure 30: Fan Curve With Two System Curves

Multiple Fans

Larger units can have dual fan options. These options give more freedom in air handling design. They increase a customized system's flexibility while still having the cost advantages of a factory-assembled package.

A dual fan segment is the key component that meets two primary needs:

- Airflow continuity
- Low-profile unit configuration

The dual fan segment is available with a variety of motor control options that can be tied directly into a BAS to guarantee the reliability of airflow application. This means that if one fan goes down, the remaining fan can operate to ensure there is still some airflow to the space until the fan is repaired. Premium efficiency motors are provided.

These new dual fan segments consist of two parallel placed fans, fan motors, a separation wall, and a means to control fan speed via variable frequency drive (VFD).

VAV Fan Turndown

A common mistake in selecting a fan for a VAV application is to assume that the fan follows a constant design system curve. VAV systems do not operate on a constant system curve. Instead, they operate in a range of operating points. The operation depends on what is required to satisfy the building requirements.

In VAV systems, the operating point continues to move from constant system changes in CFM and SP.

Find the minimum CFM using the following data:

- The design operating point
- The SP control point

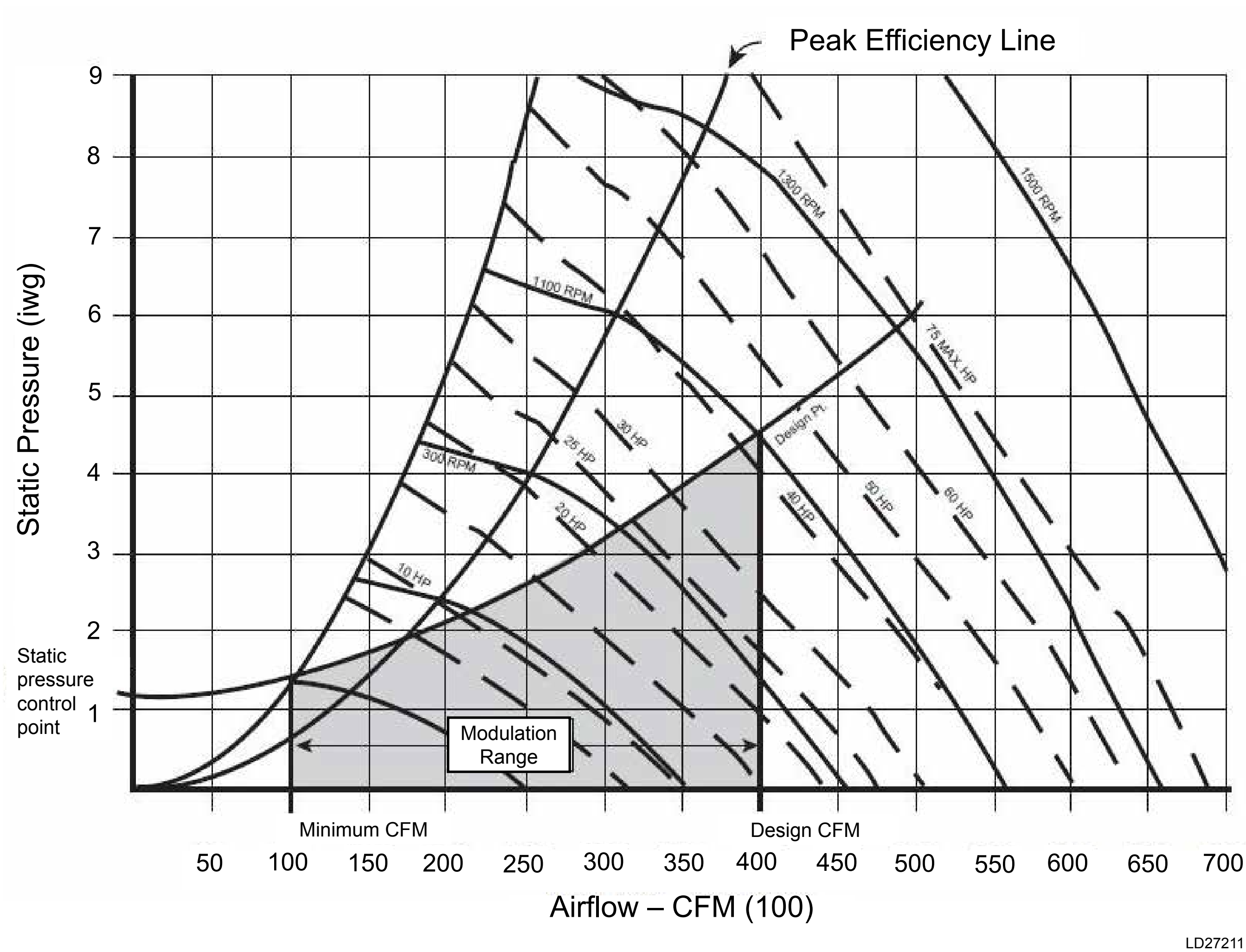

Figure 31: Plotting the Modulation Curve

The SP control point is the minimum SP required for the VAV system to operate properly. Usually, this point includes the following data:

- The minimum pressure to operate a VAV box
- Typical system losses including ductwork, diffusers and grilles, and RTU internal SP

The following example illustrates how this calculation is made, with the following values:

Design CFM (CFM_d) = 40,000

SP Control Point (SP_s) = 1.25 iwg

Design TSP (SP_d) = 4.5 iwg

The following equation expresses calculations for the minimum modulation CFM. In the process, the minimum CFM helps to generate a modulation curve and plot it on the fan curve.

$$\text{Min CFM} = CFM_d \times \sqrt{\left(\frac{SP_5}{SP_1 \times \left(\frac{CFM_d}{CFM_1}\right) + SP_s - SP_d}\right)}$$

$$\text{Min CFM} = 40{,}000 \times \sqrt{\left(\frac{4.5}{3 \times \left(\frac{40{,}000}{15{,}000}\right) + 1.25 - 4.5}\right)}$$

$$\text{Min CFM} = 10{,}517$$

CFM_1 and SP_1 are calculated from any arbitrary point on the surge line. In this example, the point is 15,000 CFM at 3.0 iwg.

After calculating the minimum CFM, the next step is to create the modulation curve that defines the range of stable operation. The following equations help calculate two arbitrary points to define the modulation curve. Solving for SP, we can select two airflows that fall below the design CFM.

$$SP_2 = \left(\frac{CFM_2}{CFM_d}\right)^2 \times SP_d - SP_s + SP_s$$

$$SP_3 = \left(\frac{CFM_3}{CFM_d}\right)^2 \times SP_d - SP_s + SP_s$$

Where CFM_2 = 30,000 and CFM_3 = 20,000

$$SP_2 = \left(\frac{30{,}000}{40{,}000}\right)^2 \times 4.5 - 1.25 + 1.25$$

$$SP_3 = \left(\frac{20{,}000}{40{,}000}\right)^2 \times 4.5 - 1.25 + 1.25$$

SP_2 = 3.1 iwg

SP_3 = 2.1 iwg

Direct Drive Plenum Fans

Plenum fans are unhoused centrifugal fans that draw air though a single opening and create a pressurized fan segment, or plenum. The plenum allows airflow to come from any direction. There is a direct drive if the motor shaft directly drives the fan with no belts or sheaves. The fan spins at the same speed as the motor. In this case, airflow control is always in the form of a VFD.

Ventilation

RTU ventilation can have a number of parts, including energy recovery wheels (ERWs) and dampers.

Energy Recovery Wheels (ERW)

Energy recovery involves energy transfer between an exhaust air (EA) stream and a supply air (SA) stream. An ERW can recover energy. For example, *Figure 32* illustrates an ERW's heat transfer process.

As the two airstreams pass through the ERW, the rotation of the wheel facilitates the transfer of energy from the higher energy airstream to the lower energy airstream. This means that the EA preheats the SA in the winter and precools the SA in the summer. Some systems use ERWs to reheat SA after it has been cooled. This is an effective means of humidity control.

Some ERWs transfer only sensible energy, while others transfer sensible and latent (in other words, total) energy.

Sensible Heat Transfer

When sensible heat transfers, the dry bulb temperature of the colder airstream increases and the dry bulb temperature of the warmer airstream decreases. Unless the warmer airstream's dry bulb temperature decreases below its dew point and condensation occurs, no moisture transfers. The humidity ratio of the two airstreams does not change.

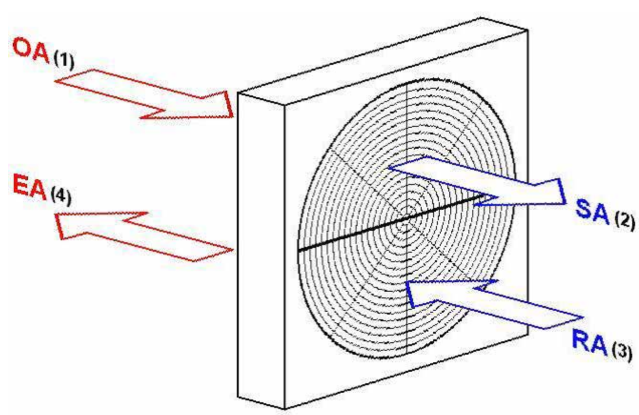

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Figure 32: Standard Airflow Conventions

Total Heat Transfer

Total heat transfer involves the transfer of sensible and latent heat energy. Latent heat energy depends on the amount of water vapor in the air. Therefore, total heat transfer can only occur when water vapor transfers from one airstream to the other.

ERWs accomplish this transfer through the use of a desiccant. The desiccant absorbs water vapor from the higher vapor pressure airstream and releases it to the lower vapor pressure airstream.

Effectiveness

An ERW's effectiveness is the ratio of the amount of energy the wheel transfers to the difference in energy levels of the two incoming airstreams. The total amount of energy transferred by the wheel is a function of the wheel's effectiveness, the airflow volumes of the two airstreams, and the difference in energy levels between the two airstreams.

Transfer of Air between Airstreams

Inherent in an ERW's operation is a direct transfer of air between the return and supply airstreams. See *Figure 33*. The air transfer is because of the following factors:

- Leakage through the seals separating the airstream
- The small amount of air carried over in the wheel matrix as it rotates from one airstream to the other

The air passing from the return to the supply airstream is defined as the EA transfer ratio (EATR). EATR is the percentage of SA that originates as RA. The measures of a tracer gas concentration in the return, supply, and outside airstreams determine EATR.

The air passing from the outside to the exhaust airstream is defined as the OA correction factor (OACF). OACF is the OA volume divided by the SA volume. Testing in accordance with AHRI Standard 1060 determines OACF and EATR for a given condition (AHRI 2014).

Figure 34 illustrates the leakage's effect on the airflow rates of an ERW with a balanced, nominal flow rate of 12,000 CFM.

Proper fan arrangement and control of the SP minimizes the air transfer caused by leakage through the gaskets and seals of the wheel.

Figure 35, *Figure 36*, *Figure 37*, and *Figure 38 on page 46* illustrate the four possible fan arrangements, including the advantages and precautions of each arrangement.

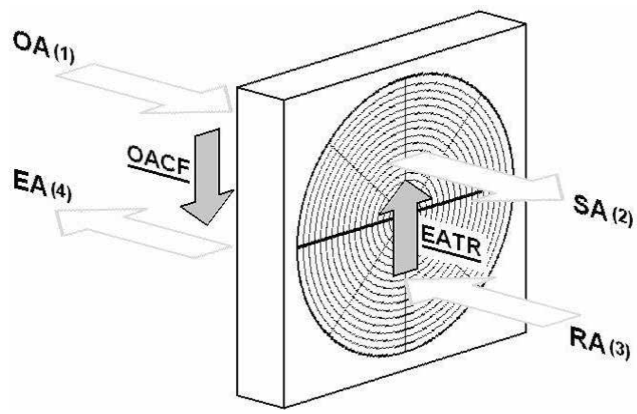

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Figure 33: Air Transfer Paths

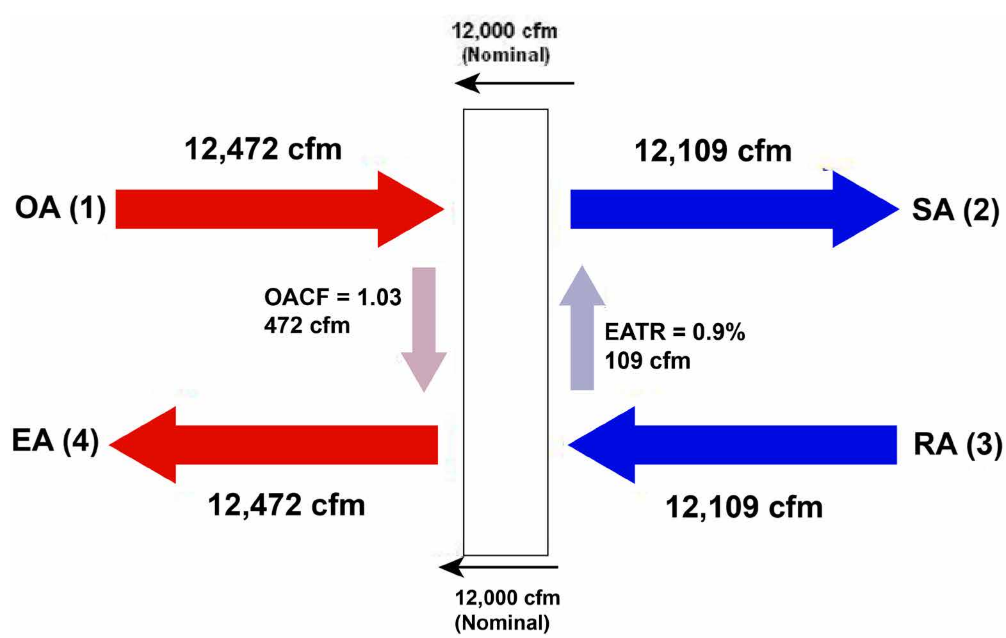

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Figure 34: Effect of EATR and OACF

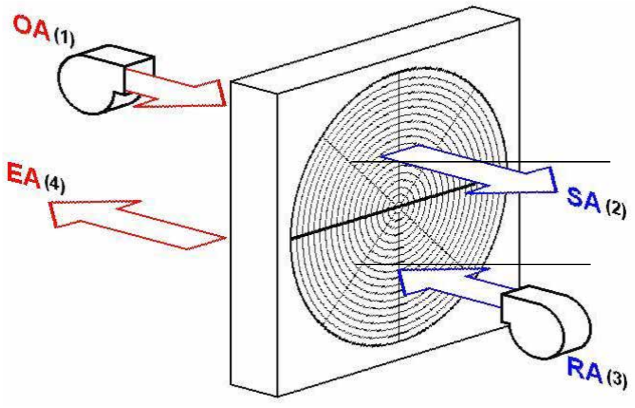

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Figure 35: Blow-Thru Supply or Exhaust

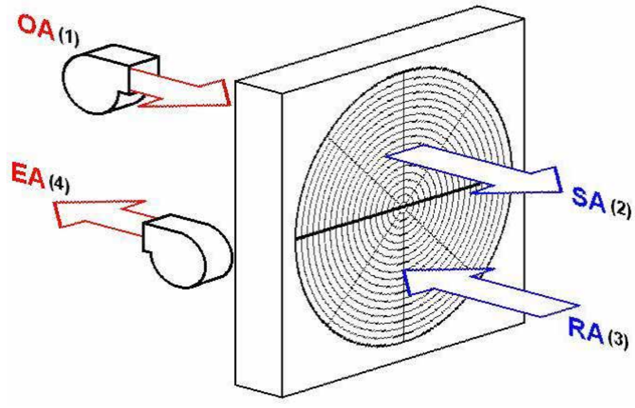

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Figure 36: Blow-Thru Supply or Draw-Thru Exhaust

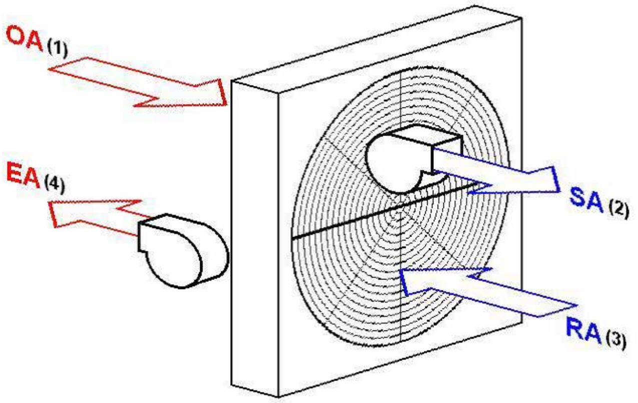

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Figure 37: Draw-Thru Supply or Exhaust

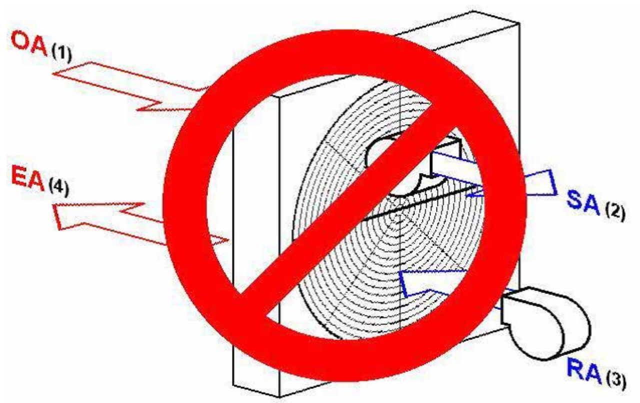

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Figure 38: Draw-Thru Supply or Blow-Thru Exhaust

Mechanical Purge

Some air from the return/exhaust path remains in the wheel's matrix as it rotates to the outside/supply airstream. This is known as carryover air. The carryover air mixes with incoming OA and enters the SA path. Carryover is not a significant problem for applications such as comfort cooling. However, if the air has high concentrations of hazardous substances (for example, volatile organic compounds (VOCs) or carcinogens), these substances can enter the supply airstream through the carryover air, which presents a health hazard to occupants.

A mechanical purge section can reduce the volume of carryover air (see *Figure 39 on page 47*). A mechanical purge isolates a section of the wheel at the RA/EA and OA/SA path boundary, where the wheel rotates from the RA/EA path into the OA/SA path.

Blocking off a section of the wheel on the RA/SA side forces OA that has traveled through the wheel to flow back into it in the opposite direction. This prevents air from the RA/EA path entering the last few degrees of the wheel before it rotates into the OA/SA path. For the RA that has entered the wheel before the purge section, there is time to exit the wheel on the exhaust side.

The angle of the purge section (how large a "slice" of the wheel the section covers) determines the purge's effectiveness. The larger the angle, the greater the carryover reduction. The purge cannot be relied on to completely eliminate carryover. When high concentrations of hazardous substances are likely in the exhaust airstream, avoid ERWs, even wheels equipped with purge sections.

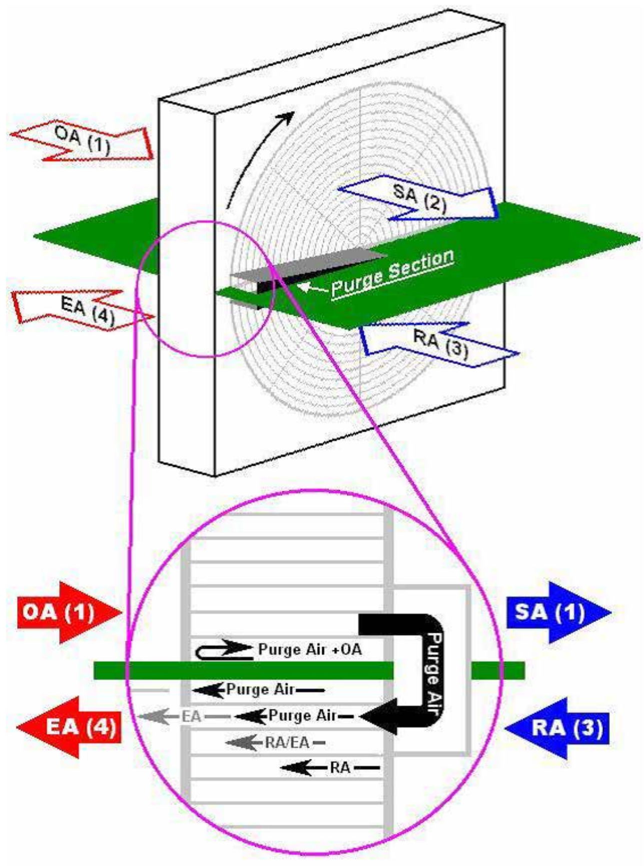

Figure 39: Energy Recovery Wheel (ERW)– Purge Section

Frost Control

HVAC units operating in sub-freezing weather are always challenged by frost formation. The likelihood of frost formation is greater on sensible-only heat transfer devices such as ERWs than on total energy transfer devices.

Figure 40 compares typical frost threshold temperatures of total ERWs and sensible-only heat exchangers. To the left of the respective boundary lines, condensation and frost begin to form. When the indoor air relative humidity is low, the dew point depression from total energy devices can lower the frost formation threshold far below 0.0°F.

Total ERWs in climates with extreme winter conditions or in systems with high indoor air relative humidity can still require frost prevention.

There are four common methods of frost prevention:

On/Off Control – The least expensive and least complicated method of preventing frost. When the wheel is not rotating, no energy transfer takes place and the moisture in the exhaust airstream is in no danger of condensing and freezing.

The drawbacks to this method are that when the ERW is not operating, there are no energy savings and it requires sizing of heating elements for design winter conditions. On/off frost prevention control is best suited for the following environments:

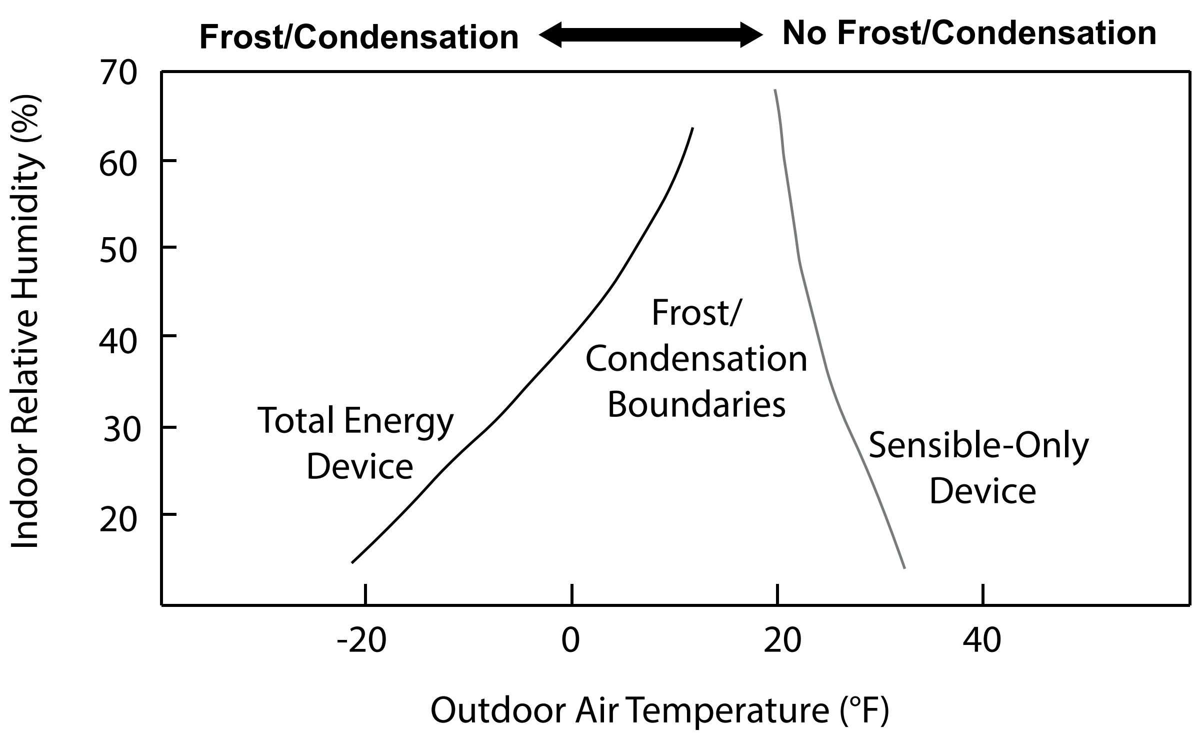

Figure 40: Frost Threshold Temperatures

- Mixed air systems with a low minimum OA requirement
- Climates where the OAT drops below the frost threshold mainly during unoccupied periods when ventilation is not required

Bypassing the OA or a Portion of OA – Another common tactic for frost formation prevention. When OA conditions are below the frost threshold, a bypass damper is used in the OA/SA path to divert some of the OA around the ERW. This reduces the heat transfer capacity of the wheel and prevents the leaving EA from reaching saturation. Proper mixing is crucial to prevent stratification, which can freeze downstream coils in the RTU.

The heating elements are larger in this type of system than in systems in which the wheel operates at 100% during winter design conditions. However, they do not necessarily have to be sized for design winter conditions.

Bypass systems are best suited for climates where there are very few hours below the frost threshold within a year and for systems that do not include humidifiers. To take full advantage of economizer operation, systems with airside economizers must have OA bypass.

Entering Air Preheat – A widely accepted method of frost control. This method has an advantage of allowing the wheel to operate at maximum air volume during design heating conditions. This method involves mounting a heating device on either the wheel's OA entering side (position one) or its RA entering side (position three). See *Figure 41*.

When the heating device is mounted in position one, the OAT is raised above the frost threshold temperature of the wheel, preventing the EA from getting cold enough to form frost. When the heating element is in position three, enough energy is added to the RA path to prevent heat transferring to the OA/SA path from lowering the EA temperature below the saturation point.

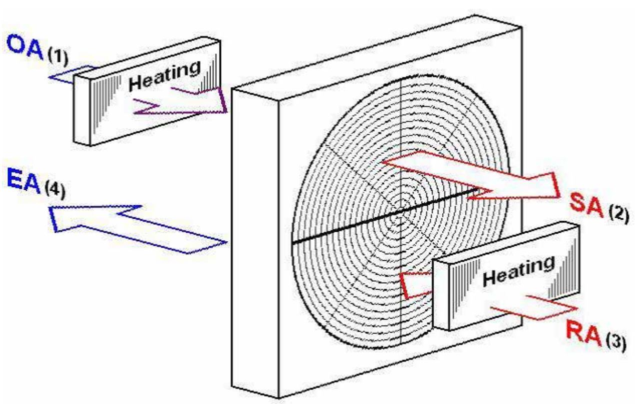

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Figure 41: Preheat Frost Control

This method of frost control suits climates with extreme winter conditions and systems that include some form of mechanical humidification. If the preheat device is a steam or hydronic coil mounted in the outside airstream, the coil must have some form of freeze protection as a precaution. No such precaution is necessary if a steam or hydronic coil preheats the RA.

A heating device mounted in position three typically requires a greater capacity than a device mounted in position one to provide the same amount of frost prevention. However, adding heat to the RA path increases the sensible heat transfer to the SA path. A preheat element in position three requires caution so that the entering RA temperature does not exceed the ERW's recommended operating temperature.

Variable Speed Control – Can be used to reduce the wheel's rotational speed, reducing the wheel's ability to transfer energy. There are many variables involved in this method. It requires a sophisticated control strategy that if not followed correctly can actually increase the likelihood of frost formation. Reducing the wheel speed can also lead to significant performance reductions and the need for supplementary freeze protection for downstream coils. As a result, the other three methods are generally preferable over variable speed control for frost prevention.

Capacity Control

Maximum heat transfer is not always necessary or desirable. The two most common methods of controlling a wheel's energy transfer rate are variable speed control and bypass of EA or OA.

Variable Speed Control – Uses a speed controller, such as a VFD attached to the wheel's drive motor. As the wheel's rotational speed decreases, the heat transfer capacity also decreases. However, there are limits to this type of capacity control. The capacity reduction is not proportional to the wheel's speed reduction. As a result, reducing the wheel's speed by 50% may only result in reducing the energy transfer by 10%.

EA or OA Bypass – Requires a bypass damper mounted in the OA/SA path or the RA/EA path. It has several advantages over variable speed capacity control.

Air bypass results in more linear control of the ERW capacity, making the control strategy more reliable. A greater overall reduction in capacity can also be achieved using bypass over variable speed control.

As the volume of air passing through the wheel reduces, the pressure drop of the system reduces. Placing the bypass damper in a position to divert EA instead of OA minimizes the possibility of stratification in 100% of OA systems.

This damper can also be placed in the EA to control the capacity of the wheel. It reduces the volume of EA available for energy transfer.

However, an RTU with an airside economizer must include some means of diverting the OA around the wheel when the system modulates from minimum OA operation to 100% economizer mode. This arrangement prevents excessive pressure drop and unwanted preheating. The economizer bypass damper can be used as a capacity control damper in the minimum OA mode. However, to prevent freezing of hydronic coils downstream of the ERW, proper mixing of treated and bypassed air is essential.

When and Where To Recover Energy

The two main uses for energy recovery are in OA preconditioning and SA tempering. Using an ERW can assist in the layout of the RTU.

Dampers

A damper is a series of valves, plates, or blades that stop or regulate the flow of air inside a duct, VAV box, RTU, or other air handling equipment.

Generally, dampers are part of the intake or discharge side of a piece of equipment. Here are some examples of their uses:

- Cutting off central air conditioning (heating or cooling) to unused rooms
- Regulating room-by-room temperature and climate control

Damper control can be manual or automatic. Manual dampers are turned by a handle on the outside of a duct. Automatic dampers can regulate airflow constantly and are operated by electric or pneumatic motors. A thermostat or BAS controls the motors.

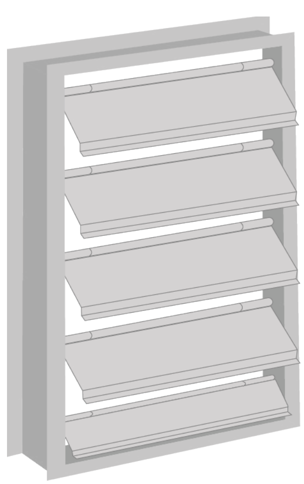

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Figure 42: Damper

Automatic or motorized dampers can also be controlled by a solenoid and the degree of calibrated airflow, possibly through signals from the thermostat going to the damper's actuator. This method modulates the flow of air conditioned air for climate control.

Opposed Versus Parallel Blades

There are two kinds of damper blades, opposed blades and parallel blades. See *Figure 43 on page 50*.

Opposed Blades – Rotate opposite each other in adjacent pairs. Air discharge through this type of damper is straighter and a bit quieter under partial-flow conditions. Opposed blade dampers are often specified where air direction control is important relative to other factors (for example, within final volume control devices).

The flow characteristics of opposed dampers are different to parallel dampers. To provide the same percentage of total air volume as a parallel damper that creates a lower modulating pressure drop, an opposed damper must be opened further. This opening creates a higher modulating pressure drop. For both types, when they are wide open, the pressure drop is the same.

Parallel Blades – Rotate so they are always parallel to each other. At any partially open position, they tend to redirect airflow and increase turbulence and mixing within the downstream duct work or plenum. This characteristic makes them good candidates for RA and OA intake into a mixing chamber. To coordinate control, the two dampers are often linked together (one opens and one closes).

Types of Damper

Controlled Dampers – A controlled damper has an actuator. When commanded by the temperature control system or unit controller, the actuator can open or close the damper.

Barometric Relief Damper – When barometric dampers reach a certain barometric pressure, the dampers open and allow air to escape.

Airflow Monitoring Damper – In one assembly, air monitoring dampers combine the functions of a control damper and flow measurement station.

Smoke and Fire Dampers – Fire dampers and fire/smoke dampers are part of the building’s life safety system. Usually, they require an auxiliary power source that is not interrupted if the building catches fire.

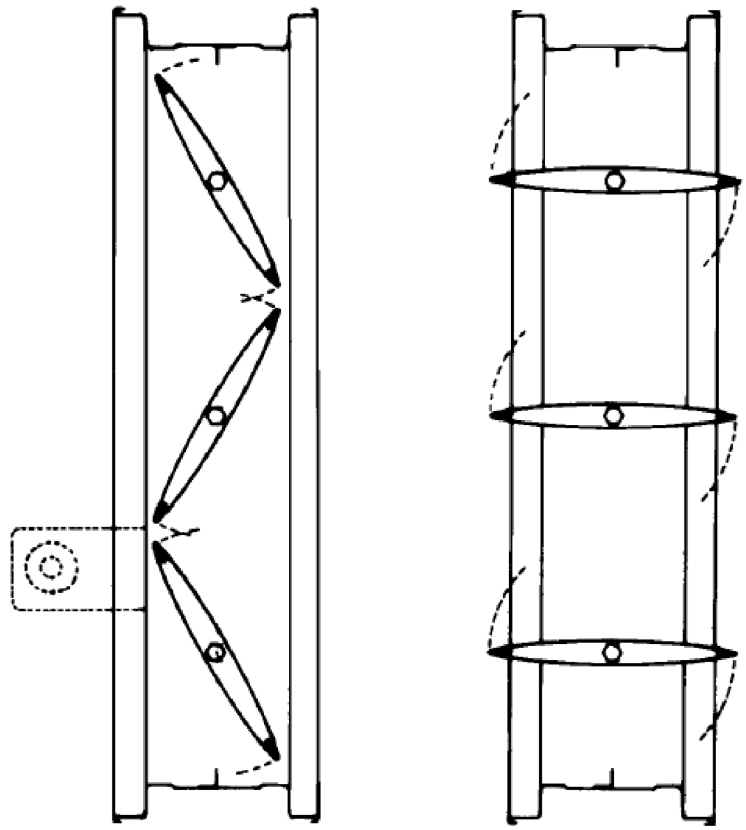


Figure 43: Opposed and Parallel Blades

Smoke dampers have the following general applications:

- Part of passive smoke control: Smoke dampers restrict the spread of smoke in HVAC systems that shut down during a fire. They close upon detection of smoke, preventing the circulation of air and smoke through a duct or a ventilation opening.
- Part of engineered smoke control: Smoke dampers control the movement of smoke in buildings with HVAC systems that include engineered smoke control. Engineered smoke control uses walls and floors as barriers and fans to create pressure differences. Pressurizing the areas surrounding a fire prevents the spread of smoke into other areas.

Smoke dampers are motorized with either an electric or pneumatic actuator. They are controlled by a smoke or heat detector signal, fire alarm, or some other building control system.

Smoke dampers are qualified under UL Standard 555S. They are always supplied with the appropriate factory mounted actuator and UL label.

It is important to determine which smoke damper is required, using the following criteria:

- Leakage Rating Class 1 (lowest leakage), 2, or 3 (highest leakage)
- Elevated temperature 250.0ºF or 350.0ºF
- Velocity and pressure

UL 555S requires all smoke dampers be rated for operation with an approved actuator. An approved actuation has a minimum airflow velocity of 2,000 FPM when open and against a minimum pressure of 4.0 iwg during closure. Smoke dampers are specified by the conditions they are exposed to in their applications (UL 2014).

Smoke dampers resist the passage of smoke and air. Controlled by smoke detectors and fire alarms, they are actuated to both open and close.

Fire dampers are different. They maintain the fire rating and integrity of walls by closing when a rise in temperature is detected and remaining closed. Installed where ducts penetrate floors or walls, most fire dampers are controlled by a mechanical fusible link that melts under sufficient heat and releases a closing mechanism.

Building codes differ on when fire or smoke dampers are required. However, it is usually when the duct penetrates a partition with a certain fire rating. The fire damper automatically closes ventilation when it gets the signal that there is a fire. This closes the air duct so that the fire is not fed by the air conditioning system and slows the fire spreading to other areas. Fire/smoke dampers are practically the same thing, except that they have a smoke sensor in addition to the fire sensor.

Damper Leakage Rates

According to ASHRAE Standard 90.1, dampers integral to the building envelope require “a maximum leakage rate of 4 CFM per square foot at 1 iwg when tested in accordance with AMCA 500-D.” (ASHRAE 2016a, 80).

According to Knapp (2007), dampers leak in two places:

- Between the blade edges
- Between the blade ends and the frame (the 'jamb')

As a result, dampers are normally supplied with blade and jamb seals to reduce air leakage. Damper leakage performance is expressed as CFM/sq. ft through a damper's face area vs. measured pressure differential across the damper. The type of seal used is directly related to the leakage rate of the damper. (Knapp 2007).

Damper manufacturers test and classify their products to AMCA Standard 500-D. This standard includes recognized and accepted standard test procedures for performance testing, including sealing performance (AMCA 2012).

See *Table 5* for leakage classifications.

Table 5: Leakage Classification per AMCA Standard 511

Pressure/ Class	Leakage, CFM/sq. ft			
	Required Rating		Extended Ranges[1]	
	1 iwg	4 iwg	8 iwg	12 iwg
1A	3	8	11	14
1	4	8	11	14
2	10	20	28	35
3	40	80	112	140

Notes
1. Optional
2. Adapted from Chart 1 (Knapp 2007)

Unit Construction and Accessories

The RTU has additional components and accessories as part of its construction, including humidifiers, coil coating, shaft grounding, vibration isolation, phase monitor, air blenders, gas-fired heating modules, and cabinet construction double wall.

Humidifiers

Humidity control is always a focus in any HVAC system design. Adding humidity to a system can add benefits such as increased comfort, control of airborne infection, improved manufacturing environments, or maintaining and preserving materials and furnishings.

A humidifier is a component or set of components designed to introduce moisture into a HVAC system’s airstream. The three main types of humidifiers are steam distributing, atomizing, and evaporative cooling. Each type is used for specific applications.

Steam distributing humidifiers maintain space humidity using either pressure or atmospheric steam during winter months.

Atomizing humidifiers maintain space humidity. Another possible application is to provide sensible cooling using compressed air and nozzles to distribute deionized/reverse osmosis (DI/RO) water as a fine mist into an airstream.

Evaporative cooling humidifiers provide sensible cooling using wetted media to lower the dry bulb temperature of the airstream.

In RTUs, the steam distributing humidifier is the most common humidifier type. A steam distributing humidifier consists of a distributor piped to an external source.

In an RTU, the distributor is usually installed in the discharge plenum downstream from the primary heating coil. The external source of humidity can be pressurized or atmospheric steam, using an existing source or generated by a system dedicated to the distribution device.

If a humidification system is going to be used every hour of every day, then the number and type of humidifiers need to reflect this. A critical system that needs to be constantly delivering a certain level of humidity must include run and standby humidifiers. Every humidifier needs to be shut down occasionally for maintenance.

Specifying the Correct Humidifier

A humidifier is considered in any application where humidity control is important, such as hospitals, healthcare facilities, museums, manufacturing environments, and comfort humidification applications.

Determining the correct humidification system largely depends on extracting the right information from the end user on how the system is to be employed.

However, it is difficult to acquire enough experience to know the right questions to ask when projects involving humidifiers only infrequently occur in a HVAC consultant's career. This basic guide helps avoid the most common specification errors.

Required Humidity Level and Acceptable Fluctuation Level

Different applications require different levels of humidity control. The most common requirement for an RTU application is an office environment's requirement of between 40–60% relative humidity. At this level, people are comfortable and static build-up is reduced.

Many manufacturing industries require a more specific level of humidity control. For instance, printers need to control humidity to a tighter 50–60% relative humidity. Textile manufacturers need a higher 65–75% relative humidity. An ideal museum environment is between 45–55% relative humidity with daily fluctuations limited to ±3% relative humidity to safeguard valuable exhibits. To prevent product waste, some pharmaceutical applications need even tighter ranges (for example, ±2% relative humidity).

If an application requires tight control of humidity, the humidifier selection is restricted to systems that quickly responds to a drop or increase in humidity, like resistive steam or spray units. Water treatment may also be required to improve the consistency of performance. These types of systems are usually only specified in precision or custom types of RTU.

Running Cost Importance and System Environmental Impact

Running costs vary widely by humidifier type. Some steam systems can use 150 times the energy of an efficient evaporative humidifier and cost 6 times as much in servicing and spare parts. The initial purchase cost is a lot less for the steam system. However, over the unit's life, an error in the initial product selection can cost the client and the environment dearly.

There are also advantages to using some evaporative humidifiers to reduce the running costs associated with the building's cooling system. Some in-duct evaporative humidifiers can provide up to 53.6°F (12.0°C) of adiabatic cooling to an RTU system. This can reduce the running costs associated with DX chillers and reduce the building's overall carbon footprint.

Type and Amount of Available Energy

There are cases where contractors arrive onsite to install equipment only to find out that there is not enough electricity to run a humidification system. For very large tasks, the energy requirements of an electrical system can become prohibitive. For the end user, evaporative, spray, or gas humidifiers may be a more viable option.

In facilities with central boiler systems, pressurized building steam is generally preferable over a steam exchange generator because of the low first cost and ease of maintenance. However, if the application requires "clean" steam (for example, a manufacturing process), a steam exchange generator may be preferable.

Usually, in facilities with no central boiler plant, electric generators offer the lowest first cost and simplest installation requirements. There are two main types of electric generators: electrode and resistive element.

Electrode Generator – Electronic power excites the electrodes in water, causing the water to boil. It is a cheaper first cost than a resistive element generator but is more expensive to maintain. It requires the use of potable water.

Resistive Element Generator – This generator boils water with a submerged heating element. This option costs more upfront, but it is less expensive to maintain. It can be used with potable or DI/RO water.

Generally, gas fired generators are the highest first cost. Installation can also be more involved, with additional piping and venting. If the humidification requirements are in the 400–600 lb/hr range, a gas fired generator can be first cost effective. In this scenario, the less expensive electric alternatives need multiple generators to match performance. This drives up equipment cost and complicates installation.

Coil Coating

Certain locations, such as coastal locations or industrial environments, can have airborne corrosive salts, acids, bases, or other manufactured chemicals. These can shorten the life of an RTU's air-cooled heat exchangers. If condenser and evaporator coils are properly coated with an effective corrosion protection system, they can withstand these specific harsh environments, providing long-term and cost-effective service. RTUs use these coil coating design features to improve unit longevity and maintain unit performance and efficiency.

The two general categories of protective coatings are pre-coats and post-coats.

Pre-Coat

When compared to the other coating options, pre-coated fin stock typically provides the lowest first cost or longevity benefit. It offers protection in low to moderately corrosive environments. However, when corrosion is a major concern, pre-coated is the least desirable option. Coating is missing on the fins' edges, the condenser framework, and the coil headers. In this situation, pre-coating results in rapid deterioration of the coil.

In addition, the application of pre-coat before manufacture can prevent the effective application of any other type of coating to the coil afterwards.

Post-Coat

There are numerous types of post-coat materials available: polyurethane, epoxy poly-urethane, phenolic, epoxy phenolic, silica-based coatings, and many others. Each of these materials has its own strengths and weaknesses, and each is also impacted by the coating or application process.

A traditional post-coating process includes dip and spray. Dip process coatings must be applied at the factory. The coils are immersed into product baths a defined number of times and for a specific time period. After this process, the coating can either be baked or allowed to air cure depending on the type of coating. The goal of this process is to provide as much surface area coverage of the coating as possible.

Generally, spray application processes can happen at the manufacturing facility or in the field:

- Factory-applied processes are typically more reliable. Coils in field installations can present significant access problems. In addition, to attempt complete penetration of the coil fin pack, a factory-applied spray process uses specific methodology and equipment for applying the coating.
- Field-spray applications range from aerosol cans to high-pressure spray guns. These applications are limited by access and space restrictions and have varying degrees of success in terms of coverage and fin pack penetration.

As with any manual application method, it is possible to perform a coating process incorrectly. The success of any coating application depends heavily on good adhesion to the base coil material. If the condenser coil is not properly cleaned and prepared before coating, failure is more likely to occur.

Ultimately, even with careful preparation and application, coating processes involving manual dipping or spraying can create surface imperfections or improper thicknesses on the coil. Improper thickness or fin bridging by these coatings leaves behind microscopic holes in the coating.

These holes allow salt air or other corrosives to reach the base metal, compromising system performance and service life. Aluminum microchannel coils with a dense fin construction only exacerbate this problem.

The highest level of coil coating (including microchannel design) can be specified and obtained with the modern e-coat process. ElectroFin® E-Coat is an electro-deposition coating process that involves epoxy paint particles suspended in deionized water. When electrically charged, these paint particles migrate to and bond with aluminum, copper, and other conductive metal surfaces that form the condenser and evaporator coils in RTUs.

The process begins with a thorough coil cleaning and de-greasing. The coil assembly to be coated is grounded electrically and suspended in the E-Coat tank. The paint particles carry a positive electrical charge so they are attracted to the metal surfaces and build up until they have achieved a uniform film. The coated surfaces are then baked until the coating cures. The cure creates a smooth surface finish designed to resist corrosion, pitting, and flaking.

Finally, a UV-resistant topcoat is applied to shield the final finish against ultraviolet degradation and to ensure maximum coating durability and service life.

This automated, computer-controlled process guarantees complete and uniform coil encapsulation and coverage. In addition, ElectroFin® E-Coat demonstrates exceptional technical properties in ASTM, DIN, and MIL-STD testing for resistance to thermal loss, UV degradation, and moisture intrusion.

Shaft Grounding

Part 31.4.4.3 of NEMA MG 1 recommends shaft grounding for an effective bearing protection of motors operated from inverter power (NEMA 2016).

Shaft voltage occurs in motors powered by VFD. VFDs induce such voltages on the motor shaft. The extremely high speed switching of the insulated gate bipolar transistors (IGBTs) can produce the pulse width modulation used to control air conditioner motors.

High frequency ground currents can cause sparks, arcing, electrical shocks, and can damage bearings. One grounding device is adequate to bleed down inverter-sourced shaft voltages, thereby protecting both motor bearings as large as 6085 frame.

The following techniques are examples of solutions to minimize this voltage problem:

- Alternate discharge paths
- A Faraday shield
- Insulated bearings
- Ceramic bearings
- A grounding brush and shaft grounding ring (SGR)

For practical reasons and cost, the simplest solution is for the motor manufacturer to install SGRs. An SGR is similar to a grounding brush. However, an SGR's brush uses conductive microfibers. These fibers create a low impedance path from the motor shaft to the ground.

Vibration Isolation

A full range of vibration isolation features include 1-inch springs, 2-inch springs, and 2-inch springs with seismic restraint. These springs are intended to isolate the vibration caused by high speed motors and fan systems. The seismic restraints assist in protecting the fan and springs from axial forces due to a seismic event.

Phase Monitor

A phase monitor is a small device that monitors the incoming voltage and protects the motor from damage caused by a number of voltage events, such as voltage unbalance, low voltage, high voltage, and lost phase.

Phase monitors improve motor. They are useful in many applications but particularly in areas where the power supplies have some variability or an unusual likelihood of losing a leg. Phase monitors with surge protection assist protect ion against damage from lightning strikes. However, for replacement cost and ease of service, separate devices may be preferable.

Air Blenders

In cold climates, insufficient mixing of OA and return air (RA) can lead to problems within a unit and throughout the building. If there is a large temperature difference between OA and RA, stratification is likely to occur at the inlet side of an RTU. This is where the increase in plenum size reduces the possibility of maintaining the turbulent conditions that promote mixing.

An air blender can increase the mixing effectiveness of the OA and RA airstreams. It reduces the likelihood of stratification (which can lead to freezestat trips), coil ruptures, or cold spots and poor indoor air quality (IAQ).

By also mixing two airstreams more effectively, an air blender can provide the following services:

- Provide better direct digital control (DDC) control in the building
- Increase the economizer cycle hours of operation

When and Where To Use Air Blenders

Air blenders are used to increase mixing effectiveness whenever there is a large percentage of OA in a system. Air blenders can offer benefits in:

- Cold climate zones where freeze protection is a concern, for example, mid to northern US and Canada
- Warm climate zones in high percentage OA systems where improper mixing can produce excessive condensate on a coil and lead to carryover

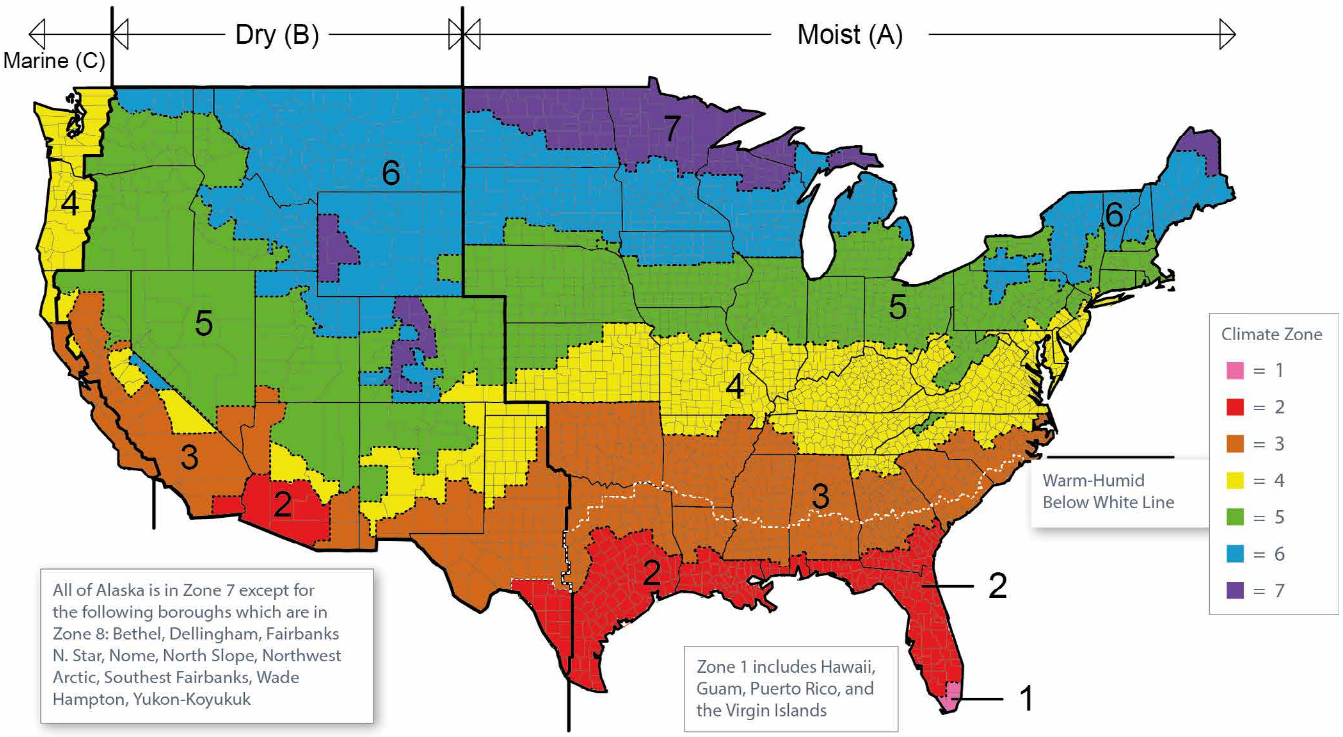
Extending Economizer Operation With Air Blenders

Historically, engineers have prematurely abandoned economizer cycles in the control sequence because of the tendency for OA and RA airstreams to stratify at low temperatures. This stratification calls for freeze protection. However, raising the mixed air's temperature above the freezestat setpoint allows engineers to operate the economizer cycle for much longer. An air blender allows increased mixing effectiveness of the two airstreams, thereby extending economizer operation.

The benefit of using an air blender for this application is dependent upon low winter OATs. As a result, not all climate zones in the US offer benefits for this application. *Figure 44 on page 55* illustrates the typical climate zones in the US and the approximate payback time of an air blender in a unit.

Table 6: Air Blender Application Summary

	Function			Result
Air Blender	Prevents Stratification	STOP	Freezestat trips	Saves maintenance cost to reset freezestat, replace coils, or troubleshoot
			Coil rupture or freeze up	Saves potential energy costs from preventative measures (for example, adding glycol, adding OA preheat)
			Cold spots	Eliminates cold spots in the building that could occur from stratified
			Poor IAQ	Eliminates poor IAQ that could occur from stratified air (in other words, ventilation air not properly mixed and dispersed)
				Eliminates poor IAQ that could occur because ventilation air is reduced to prevent freezestat trips
	Increases Mixing Effectiveness	GIVE	Better DDC control	Increased mixing effectiveness prevents temperature sensor errors that could hinder DDC control
			Extended economizer operation	Increased mixing effectiveness allows extended economizer operation to temperatures less than 30.0°F
				Extended economizer operation reduces payback and cuts energy



Note
"Reprinted from the US Department of Energy's *Guide to Determining Climate Regions by County* in August 2015 (DOE 2015, 2)

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Figure 44: Climate Zones and Typical Paybacks With Air Blenders

Gas-Fired Heating Modules

Currently, gas heat is the lowest cost heating energy. Gas heat modules are designed to be factory assembled and installed and to provide a wide range of heating options. The modules are design certified, tested to national safety standard ANSI Z83.8-2016/ CSA 2.6-2016. and installed in accordance with NFPA 54/ANSI Z223.1 (NFPA 2018).

The gas heat controls allow multiple modules to sequence either in a two-stage or modulating operation based on the selected ignition system.

The control system operates the units to meet the heating requirements under part load conditions. The heating modules are designed with a combustion draft inducer to create negative pressure in the heat exchanger. This negative pressure is required to maintain a clean, quiet, and efficient combustion process.

For each inducer, an outlet connects to a common vent located on the outside of the cabinet. The cabinet is covered by a vent cap that minimizes the effect of wind and weather.

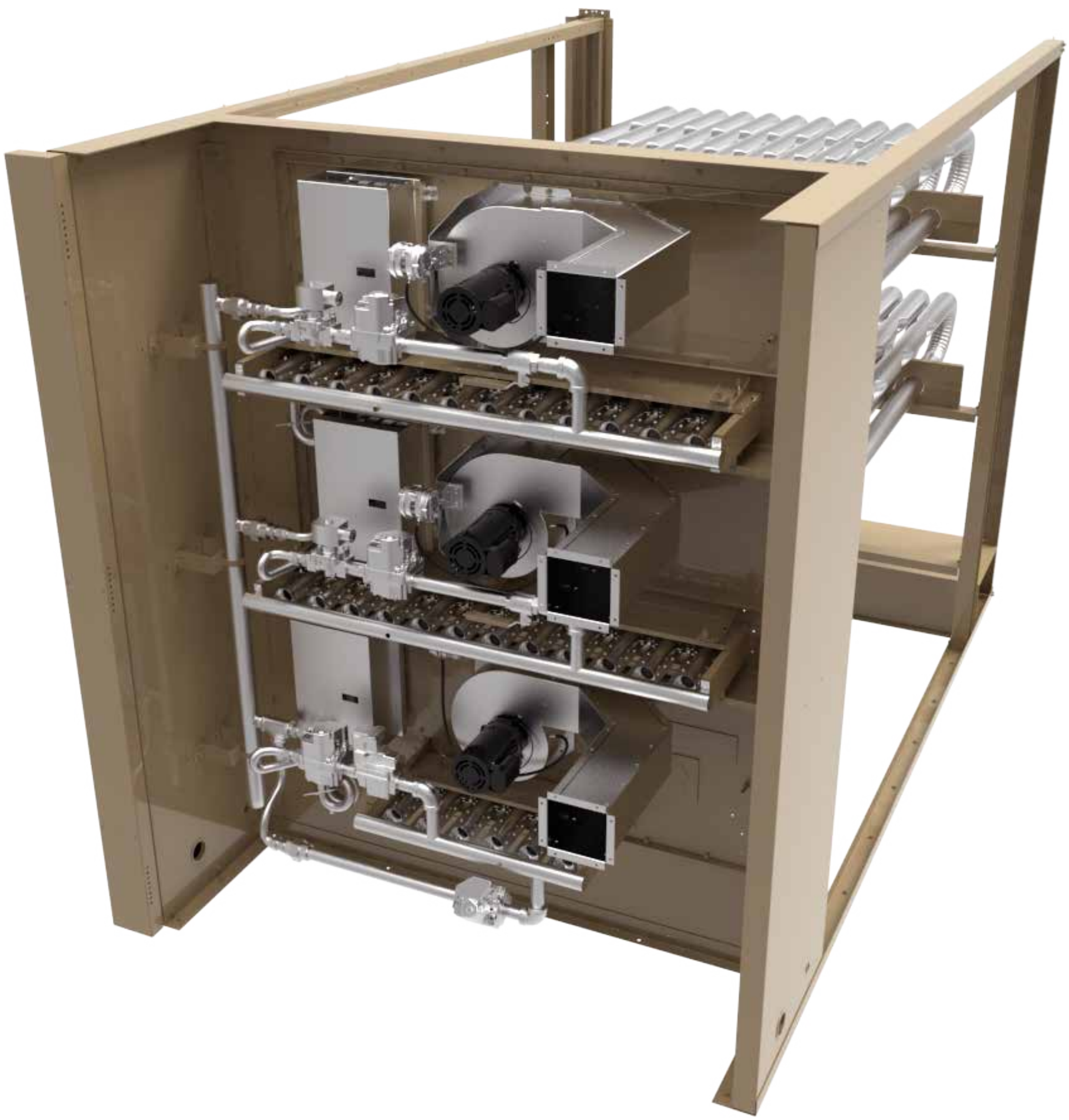

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Figure 45: Gas-Fired Heating Modules

Gas-Fired Burner

Gas-fired burners are popular choices for HGRH for the following reasons:

- They can be factory-installed inside the RTU. This makes installation easier and maximizes space.
- They do not require freeze protection and are often cheaper to operate than electric heat.
- Their outside locations reduces the danger from combustion gases.

Indirect-fired burners use a heat exchanger to separate the combustion process and airstream. They are located downstream from the supply fan. This ensures the RTU pressure is greater than the outside pressure, reducing the chance of combustion gases being drawn into the building through the supply airstream.

Cabinet Construction Double Wall

Sheet metal surfaces encapsulate insulation to form double wall construction. This type of construction protects the insulation from moisture and reduces the potential for microbe growth. It also keeps the insulation from eroding and contaminating the conditioned air.

Solid sheet metal surfaces with foam-injected insulated material are ideal in RTUs. The solid sheet metal protects the insulation. Foam is not as susceptible as other materials to erosion or airstream contamination (such as fiberglass).

Double wall construction with solid sheet metal liners and foam-injected insulation complies with of ASHRAE Standard 62.1, requiring that the airstream surface resists mold growth (ASHRAE 2016b, sec. 5.4.1). By resisting erosion, this type of double wall construction also complies with Section 5.4.2 of the same standard.

An additional benefit to such solid sheet metal liners is they are easy to clean. The interior surfaces exposed to the airstream can be cleaned periodically to remove debris accumulated from the system's mechanical components and treated air.

Hot Water and Steam Coils

For buildings that have an existing source of hot water or low pressure steam, a water/steam heating coil can be supplied as part of the RTU. A control valve with a 0–10V modulating actuator is normally a customer-supplied device, wired to the RTU controller. In heating mode, when the space sensor calls for heat, the controller begins opening the valve and modulates the flow through the heating coil. This is in response to the sensor demand.

Freeze Protection: The controller monitors the supply air temperature. If the SA fan is on, the unit is not in heating mode, and the SA temperature drops below 38.0°F, the controller begins modulating the heating valve to keep the air temperature above 38.0°F.

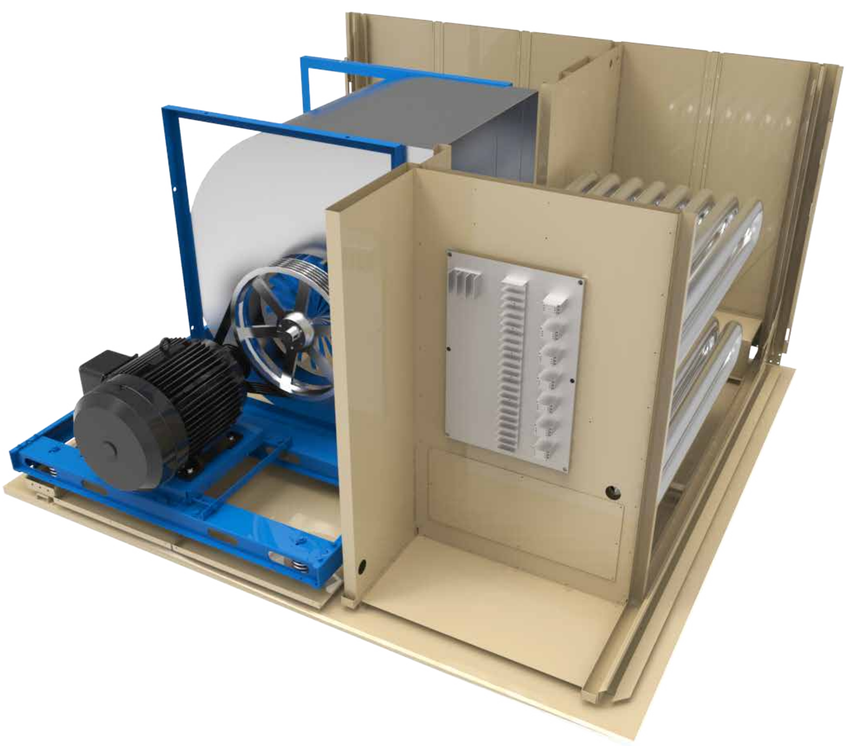

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Figure 46: Indirect-Fired Burner

When the supply fan's status is off and the OAT is less than 40.0°F, the heating valve opens to its maximum. Maximum opening ensures that water continues to flow through the coil when cold temperature could cause freezing.

Freezestat: RTUs with a steam or hot water coil include a freezestat that monitors the temperature entering the coil. If the temperature drops below the setpoint of 35.0°F, the freezestat opens and the controller opens the heating valve 100% and start a 5 minute timer. If the temperature does not rise above the freezestat setpoint within 5 minutes, the controller shuts the RTU down and displays an alarm message identifying the shutdown for freeze protection.

Codes and Standards

An important part of variable air volume (VAV) design for roofttop units (RTUs) is that it meets industrial, country, and state codes. This section discusses the most common and important codes covering RTU VAV:

- Energy: Department of Energy (DOE)
- Energy efficiency: ANSI/ASHRAE/IESNA Standard 90.1, Title 24
- Green buildings: ANSI/ASHRAE/USGBC/IES Standard 189.1, Title 24
- Performance: AHRI Standard 340/360
- Safety: IEC 60335-1, IEC 60335-2-40
- Ventilation: ANSI/ASHRAE Standard 62.1
- Wind and seismic: ASCE/SEI 7-16
- Other standards: CEE and CSA

Energy – Department of Energy (DOE)

The DOE is chartered by the Energy Policy and Conservation Act of 1975 (EPCA) and the Energy Efficiency Improvement Act of 2015. It establishes an energy conservation program for (amongst other products) commercial unitary air conditioners (CUACs) and commercial unitary heat pumps (CUHPs). The Energy Conservation Program encompasses testing, labeling, federal conservation standards, and certification and enforcement procedures.

These requirements are documented in Title 10 of the Code of Federal Regulations (CFR). Part 429 of Title 10 addresses the general guidelines for DOE product certification and compliance enforcement (10 CFR §429 (2018)). Subpart F of Part 431 of Title 10 addresses testing procedures and minimum efficiency standards for rooftops up to 760,000 Btuh or 53.3 tons (10 CFR §431.F (2018)).

Note that since January of 2010, DOE mandates minimum efficiencies at full load conditions or energy efficiency ratio (EER). However, from January 2018, DOE has revised to part load minimum efficiencies or integrated energy efficiency ratio (IEER) up to 53 tons (10 CFR §431.97 (2018)).

Along with DOE setting minimum cooling efficiency standards for rooftops, it also sets energy efficiency standards for gas-fired commercial warm air furnaces commonly used in RTUs. Part 431.77 establishes a minimum efficiency of 80% through January 1, 2023, and 81% after that date (10 CFR §431.77 (2018)).

DOE reviews the need for an energy conservation standard revision no later than 6 years after the issuance of any final rule establishing or amending a standard. If it is determined that an amendment of the rule is not required, the rule is reviewed subsequently every three years for potential revision.

Energy Efficiency – ANSI/ASHRAE/IESNA Standard 90.1

The purpose of ANSI/ASHRAE/IESNA Standard 90.1 is to save energy on all buildings. ASHRAE (2018) states that the DOE estimates that ANSI/ASHRAE/IESNA Standard 90.1-2016 will provide the following national savings in commercial buildings:

- 8.3% energy cost savings
- 7.9% source energy savings
- 6.8% site energy savings

ANSI/ASHRAE/IESNA Standard 90.1 was first published in 1975 and has been updated numerous times since then. The standard is currently on a cycle to update every three years. ANSI/ASHRAE/IESNA Standard 90.1-2016 focuses on commercial buildings, written in a way that allows code officials to adopt it. The code does not apply to residential buildings three stories or less above grade (ASHRAE 2016a).

Almost all states in the US adopt some level of ANSI/ASHRAE/IESNA Standard 90.1. State and local jurisdictions are free to adopt some or all requirements as they see fit. The DOE maintains a database that shows the minimuHim adoption level by state (DOE 2015).

When identifying the version of ANSI/ASHRAE/IESNA Standard 90.1 applicable to a jurisdiction, it helps to know that the jurisdiction has adopted the The International Energy Conservation Code (IECC). The IECC, as published by International Code Council (ICC), references ANSI/ASHRAE/IESNA Standard 90.1 as an option for compliance (ICC 2014).

Similar to ANSI/ASHRAE/IESNA Standard 90.1, IECC versions update every few years. The IECC versions typically refer to the latest ANSI/ASHRAE/IESNA Standard 90.1 versions. For example, the 2015 IECC references ANSI/ASHRAE/IESNA Standard 90.1-2013 (ICC 2014).

This application guide reviews ANSI/ASHRAE/IESNA Standard 90.1. It focuses on RTUs (without heat pumps) with cooling capacities greater than 65,000 Btuh that are air-cooled. This section is not intended to stand alone from ANSI/ASHRAE/IESNA Standard 90.1. It is recommended that the engineer of record read through the standard, then use this guide to highlight and further enhance the standard's knowledge and application.

Within ANSI/ASHRAE/IESNA Standard 90.1-2016, the Heating, Ventilating, and Air Conditioning section (part 6) defines compliance with the standard through a few paths. The first path is through the simplified approach. The simplified approach is geared towards single zone VAV (SZVAV) applications. The second and third paths focus on meeting the mandatory provisions and prescriptive path, or mandatory provisions and alternative compliance path. The alternative compliance path focuses on computer room HVAC applications (ASHRAE 2016a, sec. 6).

Because this document is focused around RTU VAV systems, the highlights discussed in this section are for ANSI/ASHRAE/IESNA Standard 90.1-2016 Mandatory Provisions and Prescriptive Path.

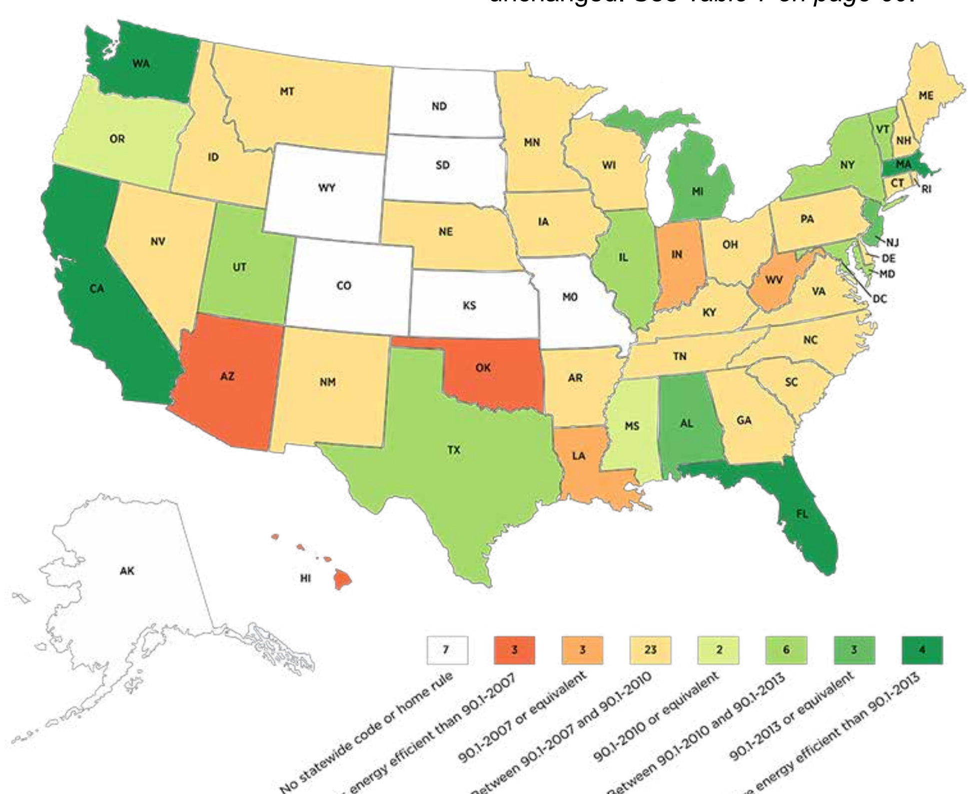
Note that compliance with the standard is met by the entire system. For example, a whole system is constituted by RTUs, VAV boxes, building automation systems (BAS), ductwork configurations, and ancillary fans.

Energy Efficiency

The minimum energy efficiencies for RTUs within ANSI/ASHRAE/IESNA Standard 90.1-2016 are defined in section 6.8. These minimum efficiencies must be met for all new equipment in a job site. If the project is the addition of an existing building, the systems and equipment are exempt from meeting section 6.8 requirements only if they already exist (ASHRAE 2016a, sec. 6.8).

RTUs discussed in Section 6.8 focus around the minimum efficiency requirements of electrically operated unitary air conditioners and condensing units (ASHRAE 2016a, sec. 6.8).

The efficiencies in the 2016 version took effect on January 1, 2016. The part load efficiencies (IEER) have increased from the original ANSI/ASHRAE/IESNA 90.1-2013 version. The full load efficiency (EER) values are unchanged. See *Table 7 on page 60*.



Note
"Reprinted from the US Department of Energy's *Status of State Energy Codes* in June 2018 (DOE 2018)

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Figure 47: Status of State Energy Code Adoption

Table 7: Electrically Operated Unitary Air Conditioners and Condensing Units – Minimum Efficiency Requirements

Equipment Type	Size Category	Heating Section Type	Subcategory or Rating Condition	Minimum Efficiency	Test Procedure
Air Conditioners, Air-Cooled	>=65,000 Btuh and <135,000 Btuh	Electric Resistance (or none)	Split System and Single Package	11.2 EER 12.9 IEER	AHRI 340/360
		All other		11.0 EER 12.7 IEER	
	>= 135,000 Btuh and <240,000 Btuh	Electric Resistance (or none)		11.0 EER 12.4 IEER	
		All other		10.8 EER 12.2 IEER	
	>= 240,000 Btuh and <760,000 Btuh	Electric Resistance (or none)		10.0 EER 11.6 IEER	
		All other		9.8 EER 11.4 IEER	
	>=760,000 Btuh	Electric Resistance (or none)		9.7 EER 11.2 IEER	
		All other		9.5 EER 11.0 IEER	

Note
Adapted from Table 6.8.1-1 (ASHRAE 2016a, 110).

Table 8: Maximum Damper Leakage (CFM/sq. ft at 1 iwg)

Climate Zone	Ventilation Air Intake		Exhaust/Relief	
	Nonmotorized[1]	Motorized	Nonmotorized[1]	Motorized
0, 1, 2				
Any Height	20	4	20	4
3				
Any Height	20	10	20	10
4, 5B, 5C				
Fewer Than Three Stories	NA	10	20	10
Three or More Stories	NA	10	NA	10
5A, 6, 7, 8				
Fewer Than Three Stories	NA	4	20	4
Three or More Stories	NA	4	NA	4

Notes
1. Dampers smaller than 24 inches in either dimension may have leakage of 40 CFM/sq. ft
2. NA = Not Allowed
3. Adapted from Table 6.4.3.4.3 (ASHRAE 2016a, 80)

Mandatory Requirements

Mandatory requirements must be met by all buildings where it is not possible to meet the simplified approach option. Depending on the project, the mandatory requirements must be paired with the prescriptive path, alternative compliance path, or the energy cost budget method. Here are some of the areas with mandatory requirements.

Economizer Shutoff Damper Controls

The shutoff damper control for outside air (OA) and exhaust air (EA) is critical to ensure the ventilation system is energy efficient. The standard requires that an actuator control the dampers so that when there are unused systems or spaces, the actuator closes the dampers (ASHRAE 2016a).

The standard also requires that these dampers close during periods where occupancy is not expected, such as during warm-up, cool down, and setback. The exception to these non-occupied periods is when a supply of ventilation air helps reduce energy consumption or where a code requires there be ventilated air (ASHRAE 2016a).

ASHRAE (2016a) standard states a few exceptions in which it is acceptable for non-motorized dampers (for example, barometric relief, back-draft) to apply to RTUs:

- All buildings less than three stories
- All buildings in climate zones 0, 1, 2, and 3
- Depending on the amount of ventilated air, systems where OA or EA is less than or equal to 300 CFM

Dampers

As part of the damper shutoff requirement, the standard details the damper's leakage rate requirement (ASHRAE 2016a). *Table 8 on page 60* is adapted from Table 6.4.3.4.3 of ANSI/ASHRAE/IES Standard 90.1. The table shows that these leakage rates vary depending on the damper being motorized or non-motorized or on the building's climate zone.

The standard requires that the leakage rating of the damper is met when tested in accordance with AMCA 500-D (ASHRAE 2016a). Certain damper manufacturers not only test to AMCA Standard 500-D but also certify to AMCA Standard 511. AMCA 511 certification ensures that the data is properly presented (AMCA 2016).

Economizer Fault Diagnostics Detection (FDD)

The section on economizer FDD only applies if an economizer is required according to Section 6.5.1. FDD is geared towards alerting a facility's personnel if there is a fault in the economizer. Faults in the economizer can result in too much or too little OA being introduced into the space, causing the RTU to use more energy or even lose control of the space setpoint. FDD is also geared to help facilities' personnel identify a bad sensor (ASHRAE 2016a).

Controls – Setback

The setback control is required to save energy and is often a setpoint when in unoccupied modes. The code defines it as “reduction of heating (by reducing the setpoint) or cooling (by increasing the setpoint) during hours when a building is unoccupied or during periods when lesser demand is acceptable” (ASHRAE 2016a, 31).

In an RTU VAV system, a BAS most commonly controls the setback. There are cases in which the end user may choose to program the setback controls into the RTU controller. However, this is not typical of a VAV system.

Controls – Optimum Start

Optimum start is used to quickly bring a system from a setback temperature to a point with greater demand or occupancy. Optimum start controls are required on systems with a supply of setback controls and direct digital control (DDC).

The sequence requires the system to “be a function of the difference between space temperature and occupied setpoint, the outdoor temperature, and the amount of time prior to scheduled occupancy” (ASHRAE 2016a, 78).

The measurement of OA was introduced in the 2013 standard. Importantly, it allows the system to respond to rapid changes in the weather. Mass radiant floor slabs also require a floor temperature (ASHRAE 2016a).

Controls – Demand Control Ventilation (DCV)

According to ANSI/ASHRAE/IESNA Standard 90.1-2016, demand control ventilation (DCV) is “a ventilation system capability that provides for the automatic reduction of outdoor air intake below design rates when the actual occupancy of spaces served by the system is less than design occupancy” (ASHRAE 2016a, 13).

DCV allows the system to save energy and ensure proper ventilation. DCV is required for all spaces with an occupancy of 25 or more people per 1,000 sq. ft (ASHRAE 2016a).

Per ASHRAE (2016a), the HVAC system must have at least one of the following features:

- An air economizer
- A motorized modulating OA damper
- OA greater than 3,000 CFM

DCV can be met through a few different methods, such as detecting carbon dioxide (CO_2) or occupancy. Many RTUs offer the ability to connect into a CO_2 sensor that allows a greater amount of OA into the space. A number of exceptions do not require DCV. These exceptions are for relatively discrete applications. For a list, refer to the standard (ASHRAE 2016a).

Controls - Humidity/Dehumidification

Humidity control is a critical function of RTUs. High humidity levels can serve as a growth environment for molds and microorganisms that are harmful to humans.

Before standards such as ANSI/ASHRAE/IESNA Standard 90.1, it was common practice to overcool OA, dehumidifying the air. However, the resulting supply air temperature (SAT) leaving the coil was too low to supply the occupied space. To meet comfort requirements, the air required reheating to the zone level. While meeting the building occupant's comfort needs, the practice requires large amounts of energy in order to also meet system needs.

The energy to properly maintain humidity levels must be controlled. For example, a high relative humidity can increase the chance of mold growth and low relative humidity can increase the chance of bacterial growth.

ANSI/ASHRAE/IESNA Standard 90.1-2016 restricts the use of fossil fuel or electricity based on the zones' humidity levels. The warmest zone in the system cannot use fossil fuel or electricity to achieve a humidity level greater than 30%. The coldest zone in the system cannot use fossil fuel or electricity to achieve a humidity level less than 60%. Systems that can both humidify and dehumidify are not allowed to run both operations at the same time (ASHRAE 2016a).

Table 9: IEER Efficiency Improvement Values Required To Eliminate Airside Economizer

Climate Zone	Efficiency Improvement	≥ 240,000 Btuh and < 760,000 Btuh		≥ 760,000 Btuh	
		Electric Heat or Cooling Only	All Other	Electric Heat or Cooling Only	All Other
		IEER Improvement	IEER Improvement	IEER Improvement	IEER Improvement
2A	17%	13.6	13.3	13.1	12.9
2B	21%	14.0	13.8	13.6	13.3
3A	27%	14.7	14.5	14.2	14.0
3B	32%	15.3	15.0	14.8	14.5
3C	65%	19.1	18.8	18.5	18.2
4A	42%	16.5	16.2	15.9	15.6
4B	49%	17.3	17.0	16.7	16.4
4C	64%	19.0	18.7	18.4	18.0
5A	49%	17.3	17.0	16.7	16.4
5B	59%	18.4	18.1	17.8	17.5
5C	74%	20.2	19.8	19.5	19.1
6A	56%	18.1	17.8	17.5	17.2
6B	65%	19.1	18.8	18.5	18.2
7	72%	20.0	19.6	19.3	18.9
8	77%	20.5	20.2	19.8	19.5

Note
Climate zone and efficiency improvement data based on Table 6.5.2.1-2 (ASHRAE 2016a)

Table 10: IEER Values Based On Air-Cooled Conditioners

≥ 760,000 Btuh		≥ 240,000 Btuh and < 760,000 Btuh	
All Other (IEER)	Electric Heat or Cooling Only (IEER)	All Other (IEER)	Electric Heat or Cooling Only (IEER)
11.0	11.2	11.4	11.6

Note
Adapted from Table 6.8.1-1 (ASHRAE 2016a)

ASHRAE (2016a) statements on humidification and dehumidification have a few exceptions, for example:

- Zones served by desiccant systems used with direct evaporative cooling in series
- Systems requiring tighter control, such as those that must be approved by the authority having jurisdiction, are required by accreditation standards, or comply with applicable codes

Before applying one of these exceptions when designing a project, specific tolerances around the relative humidity and dead bands require further review in the standard (ASHRAE 2016a).

Prescriptive Requirements

Prescriptive requirements are a method of compliance that also include mandatory requirements. For compliance purposes, if the user has elected to use the simplified approach, energy cost budget method, or alternative compliance paths, only some of the prescriptive requirements may need to be met.

The following sections highlight some areas of the prescriptive requirements.

Economizer

Economizers enable RTUs to supply free cooling to the space. The economizer operation maximizes free cooling by bringing in 100% OA. During economizer operation, to further reduce the SAT, mechanical cooling may be required.

The RTUs must be able to allow both economizer operation and mechanical cooling together. Economizer operation is best during the shoulder months when mild temperatures are neither too cool nor too hot.

According to ANSI/ASHRAE/IESNA Standard 90.1-2016, in relation to air-cooled rooftop units (RTUs), each cooling system requires airside economizers. The standard contains a review of numerous uncommon standard rooftop applications that do not need airside economizers. A notable exception to this economizer requirement is when certain efficiency improvements are met (ASHRAE 2016a). See *Table 9* and *Table 10 on page 62*.

High Limit Shutoff

High limit shutoff prevents the airside economizer from bringing in high levels of OA when it no longer reduces energy usage. The control scheme uses predefined setpoints to determine the moment to drive the unit from economizer operation back to a minimum OA operation mode. See *Table 11 on page 63*.

Economizer Controls

According to the standard, the "use of mixed-air temperature limit control shall be permitted for systems controlled from space temperature (such as single-zone systems)" (ASHRAE 2016a, 88).

The standard's definition of high limit shutoff for airside economizers dictates when the economizer no longer operates and the OA dampers go to their minimum ventilation position.

Table 11: High-Limit Shutoff Control Settings for Air Economizers

Control Type	Allowed Only in Climate Zone at Listed Set Point	Required High-Limit Set Points (Economizer Off When)	
		Equation	Description
Fixed dry bulb temperature	0B, 1B, 2B, 3B, 3C, 4B, 4C, 5B, 5C,6B, 7, 8	T_{OA} > 75.0°F	Outdoor air temperature (T_{OA}) exceeds 75.0°F
	5A, 6A	T_{OA} > 70.0°F	T_{OA} exceeds 70.0°F
	0A, 1A, 2A, 3A, 4A	T_{OA} > 65.0°F	T_{OA} exceeds 65.0°F
Differential dry bulb temperature	0B, 1B, 2B, 3B, 3C, 4B, 4C, 5A, 5B, 5C, 6A, 6B, 7, 8	$T_{OA} > T_{RA}$	T_{OA} exceeds RA temperature (T_{RA})
Fixed enthalpy with fixed dry bulb temperature	All	h_{OA} > 28 Btu/lb[1] or T_{OA} > 75.0°F	Outdoor air enthalpy (h_{OA}) exceeds 28 Btu/lb[1] of dry air or T_{OA} exceeds 75.0°F
Differential enthalpy with fixed dry bulb temperature	All	$h_{OA} > h_{RA}$ or T_{OA} > 75.0°F	h_{OA} exceeds h_{RA} or T_{OA} exceeds 75.0°F

Notes

1. At altitudes substantially different than sea level, the fixed enthalpy limit shall be set to the enthalpy value of 75.0°F and 50% relative humidity. As an example, at an elevation of approximately 6,000 feet, the fixed enthalpy limit is approximately 30.7 Btu/lb.
2. Devices with selectable rather than adjustable setpoints shall be capable of being set to within 2.0°F and 2 Btu/lb of the listed setpoint.
3. Adapted from Table 6.5.1.1.3 (ASHRAE 2016a).

The climate zone for the building determines the type of control (for example, dry bulb or single enthalpy) (ASHRAE 2016a).

ASHRAE (2016a) defines sensor calibration accuracies for OA, return air (RA), and supply air (SA). This is one of the few areas of the standard that refers to the accuracy of the devices. It is unclear why the committee chose to call out the accuracies in this specific location, though it can be assumed that the impact on energy savings is critical within the airside economizer operation.

The standard for integrated economizer control defines the unit economizer operation and mechanical cooling operation as being linked together (ASHRAE 2016a).

The standard also defines the specifics of when to prevent the economizers from closing while the unit is mechanically cooling. Linking of the economizer and mechanical cooling is based on preventing coil freeze up while balancing the moment the unit starts false loading the system (by reducing OA content) and minimum compressor run times (ASHRAE 2016a).

When there are airside economizers, RTUs greater than or equal to 240,000 Btuh must have at least four cooling stages with a minimum compressor displacement of less than or equal to 25% of full load (ASHRAE 2016a).

Simultaneous Heating and Cooling Limitation

The standard defines simultaneous heating and cooling limitations. These limitations are for mechanical cooling or heating specific to zone control, hydronic system control, dehumidification, humidification, and ventilation air heating control (ASHRAE 2016a).

Reheat to help dehumidification can be used when “at least 90% of the annual energy for reheating for providing warm air in mixing systems is provided from site-recovered energy” (ASHRAE 2016a, 93).

A common method of dehumidification within RTUs is through hot gas reheat (HGRH). According to the *ASHRAE Design Guide for Dedicated Outdoor Air Systems* (DOAS), HGRH meets the intent of site-recovered energy. The DOAS design guide specifies that HGRH is a common source of site-recovered heat (ASHRAE 2017a, 61).

Fan System Power and Efficiency

For all systems in which the nameplate horsepower exceeds 5 HP, there are two options to meet requirements for fan system power and efficiency.

The first option relies on motor nameplate HP and a simple formula that applies to all systems. The second option accounts for brake horsepower (BHP) and various components that may be installed within the system. The calculations are based on all fans that operate during the design fan condition (ASHRAE 2016a).

ASHRAE's (2016a) section on fans refers to a number of control and efficiency requirements for the air system, including but not limited to the following requirements:

- Minimum supply fan speed requirements
- Static pressure (SP) sensor locations
- VAV setpoint reset, return, and relief fan control methods
- Minimum relief
- Return fan speed requirements
- SAT reset control
- Ventilation optimization and design

Energy Recovery

The adoption rate of EA energy recovery has increased over the revisions in ANSI/ASHRAE/IESNA Standard 90.1. In the 2016 version, the advice when and where EA energy recovery needs to be applied has slightly changed to include additional climate zones. Where an energy device is required with an RTU, the most common approach may be to install the device in the rooftop. However, where the wheel is installed, there may be alternative approaches (ASHRAE 2016a).

Table 12 on page 64 shows common applications that can impact RTUs. The standard refers to other less common applications that can also apply to RTUs, and that require an energy recovery device in the system (ASHRAE 2016a).

Table 12: EA Energy Recovery Requirements

Climate Zone	% Outdoor Air at Full Design Airflow Rate	
	≥ 10% and <20%	≥ 20% and <30%
	Design Supply Fan Airflow Rate, CFM	
3B, 3C, 4B, 4C, 5B	NR	NR
0B, 1B, 2B, 5C	NR	NR
6B	≥ 28,000	≥ 26,500
0A, 1A, 2A, 3A, 4A 5A, 6A	≥ 26,000	≥ 16,000
7, 8	≥ 4,500	≥ 4,000

Notes

1. Data for ventilation systems operating less than 8,000 hours/year
2. NR = Not Required
3. Adapted from Table 6.5.6.1-1 (ASHRAE 2016a)

Energy Efficiency – Title 24

The California Energy Commission's *2016 Building Energy Efficiency Standards for Residential and Non-Residential Buildings* is codified in the California Building Standards Commission's *2016 California Energy Code*, which is Title 24, Part 6 of the California Code of Regulations. Title 24 identifies certain requirements intended to reduce electrical energy consumption of HVAC equipment.

This Application Guide focuses on the requirements of Title 24 that affect rooftop units (RTUs), how the Premier RTUs meet those requirements, and how to select the appropriate options in the Selection Navigator tool to configure a Premier RTU to meet the 2016 Title 24 requirements. Specific requirements are referenced throughout this document using the Title 24 paragraph or section number.

One of the most significant impacts of Title 24 on RTUs is the requirement for ventilation and an associated Fault Detection and Diagnostic (FDD) system (CBSC 2016, sec. 120.1–120.2). Occupied buildings are required to provide either natural ventilation or mechanical ventilation. For a RTU, this means an economizer.

Ventilation Basics

One common way to reduce energy cost is to use outside air that is below the required supply air temperature (SAT), called *free cooling*. Years ago, that meant opening the windows on a nice day. Today's large, sealed buildings require a more sophisticated approach, typically using an economizer to simulate the open window on a much larger scale.

An economizer is an adjustable opening on the return air side of the coil that allows outside air to enter the return air stream under certain conditions. Motorized dampers are modulated between full open and full closed by the control system based on comparative measurements of inside and outside air conditions (temperature or enthalpy) and also the need for cooling or ventilation of the occupied space. The economizer is used not only for the free cooling aspect, but also to provide fresh air to meet ventilation requirements based on occupancy or carbon dioxide (CO_2) levels. The exhaust damper (or fan) provides an outlet for the excess air to prevent over-pressurization of the building.When the ambient air conditions are outside of the established parameters or cooling is not required, the damper closes to maintain minimum ventilation air.

Table 13: EER/IEER Ratings

Capacity	Efficiency	Heat Source	EER	IEER
25 Tons	Standard Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.8	14.9
		Gas/Steam/Hot Water	10.6	14.8
	Standard Capacity/High Efficiency	Cooling Only/Electric Heat	11.4	16.7
		Gas/Steam/Hot Water	11.2	16.6
	High Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.8	14.9
		Gas/Steam/Hot Water	10.6	14.8
30 Tons	Standard Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.5	14.3
		Gas/Steam/Hot Water	10.3	14.2
	Standard Capacity/High Efficiency	Cooling Only/Electric Heat	10.9	15.2
		Gas/Steam/Hot Water	10.7	15.1
	High Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.5	14.3
		Gas/Steam/Hot Water	10.3	14.2
40 Tons	Standard Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.9	14.6
		Gas/Steam/Hot Water	10.7	14.5
	Standard Capacity/High Efficiency	Cooling Only/Electric Heat	11.1	16.1
		Gas/Steam/Hot Water	10.9	16.0
	High Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.7	14.6
		Gas/Steam/Hot Water	10.5	14.5
50 Tons	Standard Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.8	15.0
		Gas/Steam/Hot Water	10.6	14.9
	Standard Capacity/High Efficiency	Cooling Only/Electric Heat	10.9	15.9
		Gas/Steam/Hot Water	10.7	15.8
	High Capacity/Standard Efficiency	Cooling Only/Electric Heat	10.5	14.8
		Gas/Steam/Hot Water	10.3	14.7

Title 24 requires use of a low leakage damper to prevent infiltration of outside air when the damper is fully closed. Damper leakage shall be less than 10 CFM/sq. ft. at 1.0 iwg when tested per AMCA Standard 500 (CBSC 2016, sec. 140.4(e)4C).

Regulatory Requirements

Title 24, Part 6 is broken down into nine subchapters. Subchapter 2 defines the minimum efficiency requirements for RTUs in Table 110.2-A (CBSC 2016, sec. 110). Premier RTUs are available in 25–50 ton cooling capacities.

See *Table 13 on page 65* for Energy Efficiency Ratio/Integrated Energy Efficiency Ratio (EER/IEER) ratings.

Subchapter 3 defines mandatory requirements for non-residential, high-rise residential, and hotel/motel occupancies. Pertinent requirements include:

1. If the design occupancy or maximum occupancy is 25 people/1,000 sq. ft. or more, the RTU must incorporate demand control ventilation. Exceptions exist for certain high density purposes, such as classrooms, call centers, etc. (CBSC 2016, sec. 120.1(c)3).
2. Demand control ventilation requires CO_2 sensors in the occupied space and an outside air CO_2 sensor (CBSC 2016, sec. 120.1(c)4).
3. Variable air volume (VAV) units shall include dynamic controls to maintain measured outside air ventilation rates within 10% of required outside air rates at all airflow rates (CBSC 2016, sec. 120.1(e)2). This requirement necessitates an airflow monitoring station.

Subchapter 5 identifies prescriptive requirements for space conditioning systems. Requirements relative to RTUs include:

1. Damper Reliability Testing: Economizers shall be certified for operational life in excess of 60,000 cycles (CBSC 2016, sec. 140.4(e)4B). Premier's ultra low leak dampers meet this requirement.
2. Damper Leakage:Title 24 requires the maximum damper leakage rate to be 10 CFM/sq. ft. at 1.0 iwg (CBSC 2016, sec. 140.4(e)4C). Our Premier economizer outside air ultra low leak dampers are certified to have a maximum leakage rate of 4 CFM/sq-ft at 1.0 iwg when tested in accordance with AMCA Standard 500.
3. Electric resistance heat shall not be used for space heating (CBSC 2016, sec. 140.4(g)).
4. Hot gas bypass is not allowed except on the lowest stage of cooling (CBSC 2016, sec. 140.4(e)5C).
5. Warranty: Economizer must have a 5-year manufacturer's warranty (CBSC 2016, sec. 140.4(e)4A).
6. Fan Control:
   a. Direct expansion (DX) systems greater than 65,000 Btuh that control cooling capacity directly based on space temperature shall have a minimum of two fan speeds (CBSC 2016, sec. 140.4(m)1).
   b. VAV units shall have proportional fan control so that at 50% airflow, the fan shall draw no more than 30% of its full speed power (CBSC 2016, sec. 140.4(m)2).
   c. Systems with an economizer shall have a minimum of two fan speeds (CBSC 2016, sec. 140.4(m)3).

Economizer Fault Detection and Diagnostics (FDD)

When properly maintained and working correctly, economizers are an excellent way to reduce energy costs by utilizing ambient air cooling. Failure of the economizer—such as inoperable actuators, sticking damper blades, or bad sensors—can waste a large amount of energy by introducing cold air in the winter or hot, humid air in the summer. Title 24 addresses this issue by requiring a means to monitor the proper operation of the economizer.

Title 24 requires that all RTUs 4.5 tons and larger with an economizer shall have a means to monitor the system's operation and report any failures of temperature sensors, dampers, or improper operation (CBSC 2016, sec. 120.1(i)).

The FDD must provide a system status showing:

- Free cooling is available
- Economizer is enabled
- Compressor is enabled
- Heating is enabled (CBSC 2016, sec. 120.1(i)4)

Should any of the following faults occur, the FDD is designed to detect the fault and notify the operator using local annunciation and provide a fault signal via a building communication system (BACnet®):

- Air temperature sensor fault/failure
- Not economizing when required

- Economizing when not required
- Damper not modulating
- Excess outdoor air (CBSC 2016, sec. 120.1(i)7)

FDD functionality is built into the standard software. It can be enabled in the field via the unit keypad and display.

Selecting a Unit in Selection Navigator to Meet Title 24

The California Energy Commission requires submission of equipment for listing by model number. Visit the California Energy Commission website to download a spreadsheet that lists all certified RTU models submitted to date (California Energy Commission 2018a).

There is also a spreadsheet that lists all of the controllers that are certified to meet the Title 24 FDD requirements (California Energy Commission 2018b).

The following items must be part of the RTU to meet Title 24 requirements:

1. Cooling:
   a. Return/Exhaust Option: Exhaust fan or gravity damper
2. Economizer:
   a. Economizer: Full indoor air quality (IAQ) with full airflow measurement
   b. CO_2 sensor (check box)
   c. Economizer Control: Dry bulb sensor or single enthalpy sensor
   d. Type of Damper: Ultra low leakage
3. Heating
   a. NO electric heat

Ventilation – ANSI/ASHRAE Standard 62.1

ANSI/ASHRAE Standard 62.1 provides requirements for ventilation for acceptable indoor air quality (IAQ). ANSI/ASHRAE Standard 62.1's requirements significantly impact an RTU's design. The following overview covers the standard's purpose and how it can impact overall RTU design. The specific requirements vary depending on the location and building design. A definition of the methods meeting those specific requirements is beyond this document's scope.

ANSI/ASHRAE Standard 62.1 focuses on ventilation for occupied spaces to control contaminants. It does not focus on the occupant's thermal comfort. The stated purpose of the standard is to specify measures (for example, minimum ventilation rates) that provide indoor air quality acceptable to human occupants and minimize adverse health effects (ASHRAE 2016c).

Because air quality is inherently subjective, the standard defines acceptable IAQ as "air in which there are no known contaminants at harmful concentrations as determined by cognizant authorities and with which a substantial majority (80% or more) of the people exposed do not express dissatisfaction" (ASHRAE 2016c).

Ventilation Challenges

There are some challenges to IAQ ventilation:

- When the quality of outdoor air is poor, ventilation might not effectively improve IAQ. Bringing in contaminated outdoor air can decrease levels of one pollutant yet increase levels of another. Before completing the ventilation system, the standard requires that an investigation of outdoor air quality meet the conditions (ASHRAE 2016c, sec. 4.1-4.2).
- There is a necessary trade off with energy efficiency. When bringing in OA for ventilation, to maintain a neutral building pressure, an equivalent amount of treated air in the building must be exhausted to the outside. The incoming OA must be cooled and dehumidified, or heated (depending on the time of the year). This activity negatively impacts the building's overall energy efficiency.

There must be a delicate balancing act between ventilation and energy efficiency. Otherwise, buildings can become unhealthy for occupants. For example, the "sick building" phenomena of the late 1970s and 1980s came from buildings sealed for maximum energy efficiency and minimal ventilation.

To minimize the impact of ventilation, the standard encourages the use of energy recovery methods. Using energy recovery wheels (ERWs) or counterflow heat exchangers allows the outgoing EA to bring the incoming ventilation air closer to the desired temperature and humidity conditions. This reduces the load on the building system.

Building occupancy typically varies by day and time. As a result, the actual required ventilation rate varies from design conditions. Matching the ventilation to the actual occupancy of each zone can reduce the overall ventilation requirements and save energy. This is known as DCV. DCV requires one of the following methods:

- Occupancy tables: to predict zone occupancy at different times of the day, for example, classroom occupancy based on class schedules
- Occupancy sensors: to determine zone occupancy, for example, motion sensors

Ventilation Air Distribution

The standard requires that the ventilation distribution system be designed for 'air balancing', for example, the design provides a means of controlling the distribution of ventilation air quantities to the appropriate zones.

Ventilation Rates

Calculations of the required ventilation rates use the following procedures:

- The ventilation rate procedure
- The IAQ procedure
- The natural ventilation procedure
- A combination of any of the above procedures

The building's intended use determines the minimum required outdoor air ventilation, in CFM per person or CFM per square foot of occupied space. The standard includes a user's manual in the addenda that explains the standard more thoroughly and provides examples.

For more information and the actual procedures, refer to the standard (ASHRAE 2016c, sec. 6) or the user's manual (ASHRAE 2016b). To assist with ventilation calculations, there are also data sheets available through the user's manual (ASHRAE 2016b).

Air Filters

ANSI/ASHRAE Standard 62.1 requires that air filters and air cleaners rated MERV 8 and higher be mounted upstream of all cooling coils with a wetted surface supplying air to an occupied space. The standard does not require filters upstream of coils that provide sensible-cooling only (non-wetted coils). If no filters are used in a sensible-only application, operating controls must be provided that keep the coil surface temperature above the dew point of the airstream (ASHRAE 2016b, sec. 5.8).

Wet coil surfaces can accumulate dirt. Dirt is a potential source of microbial growth, which makes the ventilation system a source of contaminants. Air filters also reduce airborne contaminants that could be harmful to humans, such as airborne microorganisms.

Drain Pans

Standing water or surfaces that are wet for an extended period of time increase the likelihood of microbial growth. To limit these conditions, the standard has specific design requirements for drain pans underneath cooling or dehumidification coils. The requirements include requirements for pan size, slope, and drain outlet size (ASHRAE 2016c, sec. 5.10).

The requirements apply to RTU drain pans and all other drain pans in the ventilation system, for example, pans handling snowmelt from outdoor ducts or pans downstream of humidifiers. For RTUs, an important consideration is the P-trap. The evaporator section of a rooftop is under negative pressure relative to the ambient.

As a result, a P-trap is required to seal against ambient air drawn in through the drain opening and to allow condensate drainage. When designing the drain's P-trap, refer to the installation manual for the RTU carefully.

Wind and Seismic – ASCE/SEI 7-16

RTUs placed on rooftops are exposed to the outside elements. These include wind, sun, precipitation, and even seismic activity. In this section, we discuss the codes that apply to RTU VAV and wind.

Wind Standards

Wind load provisions are contained in ASCE/SEI 7-16. The latest version is ASCE/SEI 7-16 and has been adopted into 2018's International Building Codes (IBC) (ICC 2017b). Wind load provisions in ASCE/SEI 7-16 are very different to previous versions of the standard.

The following chapters of ASCE/SEI 7-16 are the most relevant to wind standards for RTU VAV:

- Wind Loads: General Requirements (ASCE 2016, sec. 26)
- Wind Loads on Buildings: Main Wind Force Resisting System (Directional Procedure) (ASCE 2016, sec. 27)
- Wind Loads on Buildings: Main Wind Force Resisting System (Envelope Procedure) (ASCE 2016, sec. 28)
- Wind Loads on Building Appurtenances and Other Structures: Main Wind Force Resisting System (Directional Procedure) (ASCE 2016, sec. 29)
- Wind Loads: Components and Cladding (ASCE 2016, sec. 30)
- Wind Tunnel Procedure (ASCE 2016, sec. 31)

In addition, ASCE (2016) includes the following figures most relevant to wind standards for RTU VAV:

- Figures 26.5-1A and 26.5-2A, that show risk greater than Risk Category I and 15%
- Figures 26.5-1B and 26.5-2B, that show risk greater than Risk Category II and 7%
- Figures 26.5-1C and 26.5-2C, that show risk greater than Risk Category III and 3%
- Figures 26.5-1D and 26.5-2D, that show risk greater than Risk Category IV and 1.6%

For these percentages, the probability is for exceedance in a 50 year period (ASCE 2016).

Velocity Pressure

To calculate the velocity pressure at a height above ground, ASCE (2016, sec. 26) provides equation 26.10-1:

$$qz = 0.00256 KzKztKdKeV^2 (lb/ft^2)$$

The terms in the equation have the following values per ASCE (2016):

- qz is velocity pressure at height z
- Kz is velocity pressure exposure coefficient (Table 26.10-1)
- Kzt is the topographic factor (Section 26.8.2)
- Kd is the wind directionality factor (for roof-mounted equipment) (see *Table 14 on page 69*)
- Ke is the ground elevation factor (see *Table 15 on page 69*)

The velocity pressure at mean roof height (qh) is qz (as calculated in equation 26.10-1) with Kz equal to the mean roof height h (ASCE 2016).

V is the basic wind speed in miles per hour, shown in the speed maps. V is used to determine design wind loads on rooftop structures, rooftop equipment, and other building appurtenances, where the risk category equals the greater one of the following categories:

- Risk category for the building on which the equipment or appurtenance is located
- Risk category for any facility to which the equipment or appurtenance provides a necessary service (ASCE 2016)

Table 14: Wind Directionality Factor

Structure Type	Directionality Factor (Kd)
Buildings	
Main wind force resisting system	0.85
Components and cladding	0.85
Arched roofs	0.85
Circular domes	1.0[2]
Chimneys, tanks, and similar structures	
Square	0.90
Hexagonal	0.95
Octagonal	1.0[2]
Round	1.0[2]
Sold freestanding, walls, rooftop equipment, and solid freestanding and attached signs	0.85
Open signs and single-plane open frames trussed towers	0.85
Triangular, square, or rectangular	0.85
All other cross sections	0.95

Notes

1. Directionality factor Kd = 0.95 shall be permitted for round or octagonal structures with non-axisymmetric structural systems.
2. Adapted from Table 26.6-1 (ASCE 2016)

Table 15: Ground Elevation Factor

Feet[2]	Metre[2]	Ground Elevation Factor (K_e)
<0	<0	See note 4
0	0	1.00
1,000	305	0.96
2,000	610	0.93
3,000	914	0.90
4,000	1,219	0.86
5,000	1,524	0.83
6,000	1,829	0.80
> 6,000	>1,829	See note 4

Notes

1. Adapted from Table 26.9-1 (ASCE 2016)
2. Values are for ground elevation above sea level.
3. The conservative approximation Ke = 1.00 permitted in all cases.
4. The factor Ke shall be determined from *Table 15* using interpolation or from the following formula for all elevations:
   - $Ke = e^{-0.0000362\,zg}$ (zg = ground elevation above sea level in feet)
   - $Ke = e^{-0.0000362\,zg}$ (zg = ground elevation above sea level in feet)
5. K_e is permitted to be taken as 1.00 in all cases.

Table 16: Velocity Pressure Exposure Coefficients

Height Above Ground Level (z)		Exposure		
Feet	Metre	B	C	D
0–15	0–4.6	0.57 (0.70)[1]	0.85	1.03
20	6.1	0.62 (0.70)[1]	0.90	1.08
25	7.6	0.66 (0.70)[1]	0.94	1.12
30	9.1	0.70	0.98	1.16
40	12.2	0.76	1.04	1.22
50	15.2	0.81	1.09	1.27
60	18	0.85	1.13	1.31
70	21.3	0.89	1.17	1.34
80	24.4	0.93	1.21	1.38
90	27.4	0.96	1.24	1.40
100	30.5	0.99	1.26	1.43
120	36.6	1.04	1.31	1.48
140	42.7	1.09	1.36	1.52
160	48.8	1.13	1.39	1.55
180	54.9	1.17	1.43	1.58
200	61.0	1.20	1.46	1.61
250	76.2	1.28	1.53	1.68
300	91.4	1.35	1.59	1.73
350	106.7	1.41	1.64	1.78
400	121.9	1.47	1.69	1.82
450	137.2	1.52	1.73	1.86
500	152.4	1.56	1.77	1.89

Notes

1. Exposure B, when z is less than 30 feet, use 0.70 (ASCE 2016, sec. 28).
2. The velocity pressure exposure coefficient K_z may be determined from the following formula:
    - For 15 feet (4.6 meter) <= z <= zg, $Kz = 2.01 (z/zg)^{2/\alpha}$
    - For z < 15 feet (4.6 meter), $Kz = 2.01 (15/zg)^{2/\alpha}$
3. Alpha and zg are tabulated in Table 26.11-1 (ASCE 2016).
4. Linear interpolation for intermediate values of height z is acceptable.
5. Exposure categories are defined in Section 26.7 (ASCE 2016, sec. 26.7)
6. Adapted from Table 26.10-1 (ASCE 2016)

Wind Loads

Lateral wind load calculations must take into account the vertical projected area of rooftop equipment on a plane normal to the direction of wind. The calculations use the following equations from ASCE (2016):

Ground mounted: Fh = qzGCfAf

- Roof mounted – all heights:

    Fh = qh(GCr)Af

- GCr is 1.9 for most applications
- qh is evaluated at mean roof height
- Af is the vertical projected area of rooftop equipment on a plane normal to the direction of wind

Vertical uplift wind load calculations must take into account the horizontal projected area of rooftop equipment on a plane normal to the direction of wind. The calculations use the following equation from ASCE (2016):

Fv = qh(GCr)Ar

- GCr is 1.5 for most applications
- qh is evaluated at mean roof height
- Ar is the horizontal projected area of rooftop equipment on a plane normal to the direction of wind

Wind and Hurricane Qualification

For wind qualification, especially for hurricanes, products require the following approvals:

- Miami-Dade County
- Department of Building and Neighborhood Compliance (BNC)
- Testing Application Standard (TAS): Testing requirements set up for exterior walls, windows, doors sheds, etc.
- State of Florida/Florida Building Code
- Florida Department of Community Affairs (FDCA)
- Texas
- Texas Department of Insurance (TDI)

In the past century, about 25 hurricanes have hit the US, causing billions of dollars' worth of damage. In 2017, hurricane Harvey and Irma alone caused between $150–200 billion worth of damage to Texas and Florida.

Hurricances are uncontrollable. However, design requirements can be set that make structures such as office buildings and homes less vulnerable to harsh and destructive weather.

Preparation is central to protecting a property from the costly damage of a hurricane. In March 2002, a new building code came into effect in Florida. This code ensures that buildings in high-intensity hurricane-prone areas can withstand the high impact from a hurricane or any wind-borne debris (ICC 2017a).

This is known as impact resistance. Building products such as doors, windows shutters, and RTUs must pass a series of tests that prove the impact resistance of the product. These tests must be carried out on any building product that is used in hurricane-prone areas (in particular, Miami-Dade County (ICC 2017a)). Products that satisfy and pass these tests receive a Notice of Acceptance (NOA) number. The NOA number certifies that the product can be used in hurricane-prone areas. This guide discusses the following NOA testing requirements for Miami-Dade County:

1. TAS 202: Uniform Static Air Pressure Test
2. ASTM E72: Racking Load Test
3. TAS 201: Impact Test (Missile Test)
4. TAS 203: Cyclic Pressure Loading Test

This guide also discusses the process for acquiring an NOA number that allows a specific product to be used in Miami-Dade County.

TAS-202: Uniform Static Air Pressure Test

TAS-202 tests if a particular product used as an external protection enveloping the building provides sufficient resistance to wind forces specified by Section 1619 of the Florida Building Code. Examples include exterior wall cladding, doors, and windows. For RTUs, examples include RTU walls, doors, and roofs (ICC 2017a).

Per ICC (2017a), the tests for an NOA application consist of applying both positive and negative uniform static air pressure loads from the inside of the test unit:

- The initial loading is 50% of the full design load. The load is held for 30 seconds and then reversed and held for 30 seconds.
- The second loading is the full design load. The load is held for 30 seconds and then reversed and held for 30 seconds.
- The final loading is 150% of the full design load. The load is held for 30 seconds and then reversed and held for 30 seconds (ICC 2017a).

ASTM E72: Racking Load Test

ASTM E72 measures a panel's resistance when subjected to a racking load as expected by wind loads blowing at 90° to the panel. The tests for an NOA application consist of applying a load to the test unit in increasing stages. The load is applied to the top unit corner with the unit secured at the base (ICC 2017a).

TAS-201: The Impact Missile Resistant Products

TAS-201 tests if a particular product used as an external protection enveloping the building provides sufficient resistance to windborne debris specified by Section 1619 of the Florida Building Code (ICC 2017a).

There are two types of impact resistance products:

- Large Missile Resistant
- Small Missile Resistant

The types have the following differences:

Large Missile Resistant – A product exposed to various impacts with a piece of lumber weighing approximately 9 lbs. and 2 inches x 4 inches x 6 feet in size, traveling at a speed of 50 feet per second (34 mph) (ICC 2017a).

The product is then subjected to hurricane loading ranging from 1,342 to 9,000 wind cycles. This is used on building components 30 feet or less above ground level (ICC 2017a).

Small Missile Resistant – A product exposed to various impacts with 30 pieces of roof gravel traveling at a speed of 80 feet per second (50 mph). The product is then subjected to hurricane loading ranging from 1,342 to 9000 wind cycles (ICC 2017a).

This is used on building components 30 feet above the ground level.Building components located 30 feet above ground level can use either the small missile testing or the large missile testing (ICC 2017a).

TAS-203: Cyclic Loading Pressure Test

TAS-203 is a test to measure, observe and record the deflection, deformation, and nature of any distress or failure of the specimen under a specified differential pressure across the specimen (ICC 2017a).

Following the successful completion of TAS-201 (Impact Missile testing), this test consists of applying cyclic air pressure loads from the inside of the test unit. The air pressure is cycled at the following percentages of design load and number (ICC 2017a):

- 50% design load (0–32.5 psf positive and negative) for 600 cycles
- 60% of design load (0–39 psf positive and negative) for 70 cycles
- Finally, 130% design load (0–84.5 psf positive and negative) for 1 cycle

Seismic Standards

This section provides an outline of IBC seismic and Office of State Wide Health Planning and Development (OSHPD) certification requirements (ICC 2011).

IBC Seismic Requirements

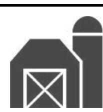
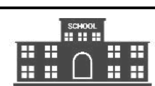
Table 17 on page 72 details when a Certificate of Compliance is required (ICC 2011).

Table 17: IBC Seismic Analysis Requirements

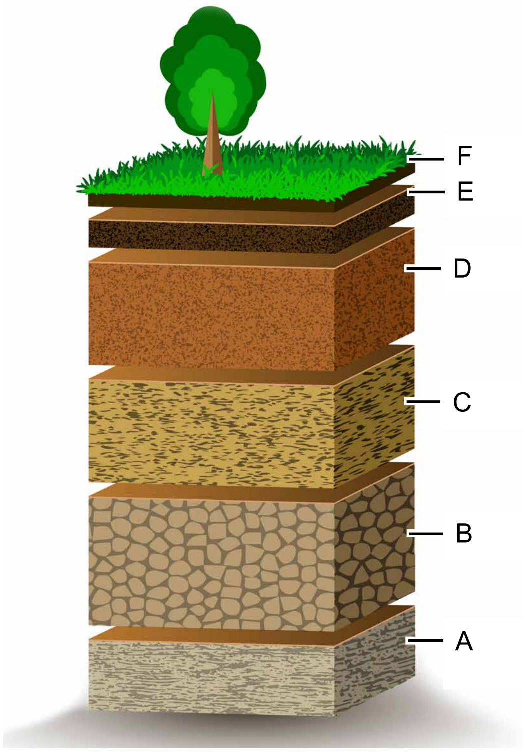
IBC Seismic Analysis Requirements							
Occupancy Category IBC	**Component Operation Required for Life Safety**	**Building Seismic Design Category**	**Required Analysis Type**			**Certificate of Compliance**
			Anchorage	**Equipment Structural Capacity**	**Equipment Operational Capacity**	
I, II, III, IV	NO	A	Not Required	Not Required	Not Required	Not Required
I, II, III	NO	B, C	Not Required	Not Required	Not Required	Not Required
I, II, III	NO	D	Static	Dynamic or Test	Not Required	For Mounting Only
I, II, III	YES	C, D	Static	Dynamic or Test	Dynamic or Test	For Continued Operation
I, II, III	NO	E	Static	Dynamic or Test	Dynamic or Test	For Continued Operation
IV	NO	C, D	Static	Not Required	Not Required	Not Required
IV	YES	C, D	Static	Dynamic or Test	Dynamic or Test	For Continued Operation
IV	NO	F	Static	Dynamic or Test	Not Required	For Mounting Only
IV	YES	F	Static	Dynamic or Test	Dynamic or Test	For Continued Operation

Notes
1. The type of building classifications is given by the Occupancy Category.
2. The structural engineer of record determines the seismic design category.

Table 18: Occupancy and Risk Category

OCC/Risk Category	Type	Description	Building Importance Factor
I	 Barn	Buildings and other structures that represent a low hazard to human life in the event of failure.	1.0
II	 Commercial/Residential	All buildings and other structures except those listed in Occupancy Categories I, III, and IV.	1.0
III	 School	Buildings and other structures that represent a substantial hazard to human life in the event of failure.	1.25
IV	 Hospital	Buildings and other structures designated as essential facilities.	1.5

Site Class



A	Hard rock
B	Rock
C	Very dense soil
D	Stiff soil (default)
E	Soft clay soil
F	Fill, special analysis required F allows an assumption of Site Class D when: • Class is unknown • Class E or F is not expected

LD27213

Figure 48: Site Class

Table 19: Component Importance Factor

I_P = 1.5	I_P = 1.0
Used for life-safety purposes (sprinklers, emergency power systems, smoke exhaust, stairwell pressurization)	Everything else
Conveys or contains toxic, explosive, or hazardous materials (natural gas, biohazard fume hood exhaust, high pressure steam, medical gas such as O_2)	
Required for continued operation of Risk Category IV (all systems in or connected to hospitals, fire stations, water treatment plants, storm shelters)	

Force Equation

ASCE (2016, sec.13) defines the amount of lateral seismic design force on non-structural components with the following equation (equation 13.3-1 in the standard):

$$F_p = \frac{0.4 a_p S_{DS}}{\left(\frac{R_p}{I_p}\right)} \left(1+2\frac{z}{h}\right) W_p$$

In practice, this equation can be simplified:

$F_p = g \times W_p$

- g is a force factor
- Apply F_p at the center of gravity of components

Table 20: Seismic Restraint Requirements by Seismic Design Category

RISK CATEGORY	+ SITE CLASS	+ SEISMIC ACCEL.	+ SEISMIC DESIGN CATEGORY

A – Exempt (Non-structural)
B – Exempt (Non-structural)
C – Only consider components assigned with I_P = 1.5 (Hazardous, Life Safety, or Continued Operation Requirement)
D, E, or F – Consider all non-structural components (I_P = 1.0 & 1.5) (Some components are exempt based on size, location, and weight)

Anchorage

If the component importance factor (I_P) is equal to 1.0, equipment requires both attachment/anchorage and over-turning certification. The certification is required even if the equipment does not function. After a seismic event, all external parts must remain attached or anchored (ICC 2011).

Rooftop equipment must be secured to the building structure because failing to do so can cause significant damage. For example, FEMA spent almost $100 million in aid for repairs to the damage caused by earthquakes in 1989 (Santa Cruz, CA; 7.0) and 1994 (Los Angeles, CA; 6.6). An estimated 40–80% of the damage was non-structural from the lack of a proper restraint system (ICC 2011).

Generally, the following anchorage guidelines apply:

- Life safety is the primary purpose of seismic or wind design for HVAC equipment and systems.
- Anchorage cannot depend on gravity. The equipment must be positively attached to the building structure.
- Restraint systems and attachments to the building structure must be detailed by calculations.
- Compliance with building code requirements is necessary for securing rooftop equipment to building structure.

All 50 states have adopted IBC in effect at the local or state level. IBC provides a direct reference to ASCE/SEI 7-16 for load calculations, both wind and seismic. Note that the building code establishes minimum design criteria. Frequently, the code requirements are increased by local amendments (ICC 2011).

Here are some examples of such amendments:

- Miami-Dade County for wind
- NFPA 13 for sprinklers
- Data centers – owner's request
- Government, Military applications, Bomb Blast and Anti-Terrorism (UFC Code)

Mandates

HVAC anchorage, including RTU VAV anchorage, has a number of general mandates:

- Design the systems (HVAC Equipment) so that they meet the applicable building code.
- Avoid property damage or loss of human life. Restrain non-structural components from overturning or moving.
- Create a proper load path to transfer wind or seismic forces to the building structure. Supports and attachment of the support to the component must be designed for force displacement.

Seismic and Wind Calculations

Rooftop equipment requires both seismic and wind calculations. For both selection and calculation of the restraints, the greater load must be considered.

For wind load, the equipment size and shape directly affects its magnitude. The building's location also significantly impacts the magnitude, for example, there is a difference in wind load magnitude between buildings located on the coast and buildings located inland.

For seismic load, the operating weight relates directly to the magnitude. The equipment's size and shape also impact the seismic load. Other significant impacts come from project location, soil type, and building occupancy.

Seismic Loads

Seismic load calculations must take into account the equipment operating weight. These calculations use the equation used for lateral seismic load (see equation 13.3-1 (ICC 2011)).

RTU Configuration

RTU VAVs come in a number of different configurations for anchorage. The anchorage configurations can depend on the RTUs' materials and components. Steel structures require the calculation of the following configuration (ICC 2011):

- The equipment's attachment to supporting structure
- The attachment of the supporting structure to the restrained isolator
- The seismically-rated restrained isolator attachment to the building structure

Concrete piers, including air-cooled chillers and cooling towers, require details of the concrete anchors' type and size be calculated. The calculation requires the anchor's proper edge distance, spacing, and embedment (ICC 2011).

Curbs, including sheet metal curbs, non-isolated curbs and isolation curbs, require the following configuration (ICC 2011):

- The curb must be rated for wind and seismic applications.
- Details of the curb attachment and equipment attachment to the curb to be calculated.

Concrete curbs and pads must be designed by structural design professionals and must be part of the building structure. Seismically rated concrete anchors must be used for attachments to the concrete pad. If required, they use seismically rated brackets. They require details of the equipment attachment to the bracket be calculated (ICC 2011).

The rooftop equipment must be analyzed for both seismic and wind loads. The greater load must be considered for restraint calculations. The design must include a proper load path to transfer seismic or wind load to the building structure (ICC 2011).

The supports and attachment of the support to the component must be designed for forces and displacements. The following calculations are required (ICC 2011):

- Seismic and wind calculations to select the load that governs the equipment
- Overturning calculations to determine the reaction forces on equipment connection to building structure. Select the proper restraint system based on these calculations.
- If mounted on vibration isolators or supported by a sheet metal curb, details of the connection of isolators or sheet metal curb to building structure
- Rating of vibration isolators for seismic or wind applications
- Rating of the curb for seismic or wind applications

Green Buildings – ANSI/ASHRAE/USGBC/IES Standard 189.1

The following information has been compiled from ANSI/ASHRAE/USGBC/IES Standard 189.1 with a focus on packaged RTUs. For the complete text, reference the standard. This document is intended to help compliment the standard and does not replace it.

ANSI/ASHRAE/IESNA Standard 90.1 and ANSI/ASHRAE/USGBC/IES Standard 189.1

ANSI/ASHRAE/IESNA Standard 90.1 is the energy standard for buildings other than low-rise residential buildings. This standard provides minimum requirements for the energy-efficient design of new, renovated, or retrofitted buildings (ASHRAE & USGBC 2014).

Building design and construction throughout the US follow ANSI/ASHRAE/IESNA Standard 90.1. This standard describes in detail the minimum requirements for design and construction. However, it does not detail state of the art design. The intent of ANSI/ASHRAE/USGBC/IES Standard 189.1 is to describe in detail the design and construction of high-performance green buildings (except for low-rise residential buildings). This standard offers guidance beyond the minimum requirements of ANSI/ASHRAE/IESNA Standard 90.1 (ASHRAE & USGBC 2014).

The requirements of ANSI/ASHRAE/USGBC/IES Standard 189.1 are more stringent than ANSI/ASHRAE/IESNA Standard 90.1's requirements. ANSI/ASHRAE/USGBC/IES Standard 189.1 provides a holistic approach to reduce energy and environmental impact through sustainability. Over time, the requirements in ANSI/ASHRAE/IESNA Standard 90.1 supersedes ANSI/ASHRAE/USGBC/IES Standard 189.1 (ASHRAE & USGBC 2014).

Purpose and Scope of ANSI/ASHRAE/USGBC/IES Standard 189.1

ASHRAE & USGBC (2014, 6) states that the purpose of ANSI/ASHRAE/USGBC/IES Standard 189.1 is to provide

> minimum requirements for the sitting, design, construction and plan for operation of high-performance green buildings to:
>
> > a. balance environmental responsibility, resource efficiency, occupant comfort and well-being, and community sensitivity; and

b. support the goal of development that meets the needs of the present without compromising the ability of future generations to meet their own needs.

The scope of this standard covers the following (ASHRAE & USGBC 2014):

- New buildings and their systems
- New portions of buildings and their systems
- New systems and equipment in existing buildings

The scope does not cover the following (ASHRAE & USGBC 2014):

- Single-family houses
- Multifamily structures of three stories or fewer above grade
- Manufactured houses, both mobile homes and modular
- Buildings that do not use electricity, fossil fuel, or water

The following categories are applicable to increasing efficiency (ASHRAE & USGBC 2014):

- Site sustainability; water use efficiency; energy efficiency; indoor environmental quality (IEQ); the building's impact on the atmosphere, materials, resources; construction and plans for operation.

Finally, in this standard, there are three provisions for compliance (ASHRAE & USGBC 2014):

- **Mandatory** – Provisions in this option must be met at all times. These provisions take priority over equal or more stringent provisions in other subsections.
- **Prescriptive** – An alternative to the performance option, provisions in this option must be met in addition to all mandatory provisions. This option involves minimal calculation and has a simple compliance approach.
- **Performance** – An alternative to the prescriptive option, provisions in this option are performance based. The provisions must be met in addition to all mandatory provisions. This option involves advanced calculations and offers a more complex alternative approach. Compared to the prescriptive option, the performance option is expected to give similar or better performance results.

Mandatory Requirements

Section 7 covers an RTU's energy efficiency requirements. The mandatory requirements are the mandatory requirements of ANSI/ASHRAE/IESNA Standard 90.1 (ASHRAE & USGBC 2014, sec. 7).

Section 8 covers an RTU's requirements for IAQ, outdoor air delivery monitoring, ventilation, and thermal comfort. It has the following mandatory requirements (ASHRAE & USGBC 2014, sec. 8).

For air quality, RTUs must comply with the ventilation rate procedure of Standard 62.1. The minimum outdoor airflow rates in health care facilities must comply with Standard 170 (ASHRAE & USGBC 2014, sec. 8).

For all mechanical ventilation systems, outdoor air intake measurement must be a permanently installed feature, for testing, balancing, recommissioning, and monitoring OA. The permanently installed outdoor air intake measurement device must comply with Standard 111 (Testing, Adjusting, and Balancing of Building HVAC Systems). It must meet the following requirements (ASHRAE & USGBC 2014, sec. 8):

- It is accurate to 10% of minimum outdoor airflow
- Provide a fault status to the building monitoring system. This fault status is manually reset

For filtration and air cleaner requirements, particulate matter has the filter requirements (ASHRAE & USGBC 2014, sec. 8):

- For wetted surfaces, provide MERV 8 filters upstream of all cooling coils
- For particulate matter smaller than ten micrometers (PM10), provide MERV 8 filters
- For particulate matter smaller than 2.5 micrometers (PM2.5), provide MERV 13 filters

Healthcare facilities are excluded from these requirements for particulate matter. ANSI/ASHRAE/ASHE Standard 170 applies to them (ASHRAE 2017c).

Sealing has the following requirements (ASHRAE & USGBC 2014, sec. 8).:

- In cases where multiple filters are used in a system, seal gaps between each filter with a gasket. Seal gaps between the filter and its track or support with gaskets that expand when the filter is removed.
- Seal filter tracks and filter supports to the HVAC equipment housing and ducts.
- Seal filter access doors appropriately. When replacing a filter with a smaller filter, do not install spacers.

For preoccupancy ventilation, automatic controls must provide outdoor air by enabling ventilation systems in spaces that have been unoccupied for 24 hours or longer. The system must provide ventilation continuously for an hour before expected occupancy at the following rates (ASHRAE & USGBC 2014, sec. 8):

- A minimum outdoor air flowrate
- An outdoor air flowrate for a period providing the same number of air changes as the design minimum outdoor air for for one hour

The start time of preoccupancy ventilation in Section 81 takes priority over Section 6.4.3 of ANSI/ASHRAE/IESNA Standard 90.1 (ASHRAE 2016a, sec. 6.4.3). This section excludes hotel and guest rooms in Section 7 and Section 8 (ASHRAE & USGBC 2014, sec. 7–8).

For thermal environmental conditions for human occupancy, the building design must comply with Section 6.1 and Section 6.2 of ANSI/ASHRAE Standard 55-2017. These sections cover compliance of design and design documentation, respectively (ASHRAE 2017b, sec. 6.1–6.2).

For humid spaces, any analysis must be conducted in spaces where process or occupancy requirements have unique dew point conditions with respect to the building's other spaces.

Section 9 covers the building's impact on the atmosphere, materials, and resources. The purpose of this section is to detail the requirements for reducing the impact on the atmosphere through regulating materials and resources (ASHRAE & USGBC 2014, sec. 9).

These materials and resources include those used by the equipment and also during the building's construction. In particular, a mandatory requirement is that the selected refrigerants are not based on Chlorofluorocarbons (CFCs) (ASHRAE & USGBC 2014, sec. 9).

Finally, Section 10 covers construction and operation plans. This includes construction, commissioning tests, performance tests, maintenance, and life-cycle plans (ASHRAE & USGBC 2014, sec. 10).

All requirements in this section are mandatory. For example, HVAC systems (including RTU VAV) need to meet the following requirements (ASHRAE & USGBC 2014, sec. 10):

- Acceptance testing
- Commissioning
- Maintenance (and monitoring) of air quality
- Outdoor airflow
- Moisture control
- Data monitoring
- Periodic maintenance
- Service life

For activities before building occupancy, systems required by the manufacturer must be installed. For each acceptance test, a test form must be completed. The form must include the following information (ASHRAE & USGBC 2014, sec. 10):

- Signature
- License number
- Operation and maintenance documentation
- Full warranty

HVAC systems (including RTU VAV) and their associated controls must undergo acceptance testing. They must also be commissioned (ASHRAE & USGBC 2014, sec. 10).

For the construction management of IAQ, HVAC systems (including RTU VAV) must not be operated during construction unless the systems are undergoing start-up, testing, balancing, or commissioning (ASHRAE & USGBC 2014, sec. 10).

Table 21: ASHRAE 90.1, ASHRAE 189.1, and Department of Energy CFR – Minimum Efficiency Requirements

Reference*		Table 6.8.1-1 Electrically operated Unitary Air conditioners & Condensing Units-Minimum Efficiency Requirements	Normative Appendix B Reduced Renewables & Increased Equipment Efficiency Approach in Section 7.4.1.1.2	Department of Energy; Minimum Efficiency Requirements	
		Minimum Efficiency Effective Date		Effective Date	
Size Category	Heating Selection Type	(ASHRAE 90.1-2016)	(ASHRAE 189.1-2014)	January 1, 2018	January 1, 2023
>=240,000 Btuh <760,000 Btuh	Electric Resistance (or none)	10.0 EER 11.6 IEER	10.5 EER 11.3 IEER	11.6 IEER	13.2 IEER
	All Other	9.8 EER 11.4 IEER	10.3 EER 11.1 IEER	11.4 IEER	13 IEER
>=760,000 Btuh	Electric Resistance (or none)	9.7 EER 11.2 IEER	9.9 EER 11.1 IEER		
	All Other	9.5 EER 11.0 IEER	9.7 EER 10.9 IEER		

Notes

1. *Table 21* shows data for a test procedure and AHRI 340/360.
2. The equipment type is air-cooled air conditioners.
3. The subcategory or rating conditions are single packaged split systems.
4. The Department of Energy 10 CFR §431.97 is effective from Jan 1, 2018.

Prescriptive Requirements

Along with complying with requirements in Section 6 of ANSI/ASHRAE/IESNA Standard 90.1, RTUs must comply with modifications and additions in Section 7.4.3 of ANSI/ASHRAE/USGBC/IES Standard 189.1. This section covers heating, ventilation and air conditioning. RTUs must especially comply with the following requirements (ASHRAE & USGBC 2014, sec. 7):

- Section 7.4.3.1 which details the minimum equipment efficiencies for the alternate renewables approach
- Normative Appendix B
- Relevant ENERGY STAR requirements in Section 7.4.7.3.2

Table 21 shows a comparison between the minimum efficiency requirements under ANSI/ASHRAE/IESNA Standard 90.1, ANSI/ASHRAE/USGBC/IES Standard 189.1, and the DOE 10 CFR §431.97 (effective Jan 1, 2018 and 2023). Note that DOE minimum efficiency requirements supersede the other efficiencies listed here.

Systems must meet requirements that supersede Section 6.4.3.8 of ANSI/ASHRAE/IESNA Standard 90.1, detailed in 7.4.3.2 (ASHRAE & USGBC 2014, sec. 7.4.3.2). The most relevant requirement is that densely occupied spaces must provide DCV.

DCV is provided through one of the following components (ASHRAE & USGBC 2014, sec. 7):

- An airside economizer
- Automatic modulating controls of the outdoor air dampers
- Design outdoor airflow that is greater than 1,000 CFM (500L/s).

There are some exceptional scenarios to this requirement (ASHRAE & USGBC 2014, sec. 7):

- EA energy recovery systems that comply with Section 7.4.3.6, design outdoor airflow less than 750 CFM (L/s)
- Spaces where more than 75% of the space design outdoor airflow is utilized as makeup or transfer air to provide makeup air for other spaces
- Spaces with one of the following occupancy categories as defined in ANSI/ASHRAE Standard 62.1: cells in correctional facilities; daycare sickrooms; science laboratories; barbers; beauty and nail salons; and bowling alleys (seating).

The DCV system must be designed in compliance with Section 6.2.7 of ANSI/ASHRAE Standard 62.1 (ASHRAE 2016b). CO_2 sensors must meet the following requirements (ASHRAE & USGBC 2014, sec. 7):

- One sensor or probe per 10,000 sq. ft of floor space
- Sensors installed 3–6 feet above the floor
- Sensors are accurate within 50 ppm of 1,000 ppm

The measurement of outdoor air CO_2 concentration is found through one of the following methods (ASHRAE & USGBC 2014, sec. 7):

- A CO_2 sensor
- A fixed typical value, from a review of available statistical data on CO_2 concentration

For economizers, systems meet the prescriptive requirements of of ANSI/ASHRAE/IESNA Standard 90.1, Section 6.5.1, except for certain modifications (see *Table 22*). In particular, there are the following requirements (ASHRAE & USGBC 2014, sec. 7):

- As outlined in Table 7.4.3.3, economizers are required for all units with a cooling capacity greater or equal to 33,000 Btuh. This supersedes Tables 6.5.1-1 and Table 6.5.1-2 of ANSI/ASHRAE/IESNA Standard 90.1 (ASHRAE 2016a).
- If they have a capacity equal to or greater than 54,000 Btuh (16 kW), RTUs must meet the staging requirements in ANSI/ASHRAE/IESNA Standard 90.1, Section 6.5.3.1 (ASHRAE 2016a).
- During the economizer operation, systems that control to a fixed leaving air temperature must be able to reset the SAT up to at least 5F (3C).

For zone controls, there are exceptions to Standard 90.1, Section 6.5.2.1. Standard 189.1 removes exception (1). Exception (2)(a)(2) refers to the zone's design outdoor airflow rate.

Table 22: Minimum System Size Requiring An Economizer

Climate Zones	Cooling Capacity For Which An Economizer Is Required
1A, 2B	No Economizer Requirement
2A, 2B, 3A, 3B, 3C, 4A, 4B, 4C, 5A, 5B, 5C, 6A, 6B, 7, 8	>= 33,000 Btuh (9.7 kW)[1]

Notes

1. Where economizers are required, the total capacity of all systems without economizers shall not exceed 480,000 Btuh (140 kW) per building or 20% of the buildings air economizer capacity, whichever is greater.

2. Adapted from Table 7.4.3.3 (ASHRAE & USGBC 2014).

In Section 7.4.3, several perspective requirements in ANSI/ASHRAE/USGBC/IES Standard 189.1 supersede or are in addition to prescriptive requirements of ANSI/ASHRAE/IESNA Standard 90.1. The following requirements are the most relevant to RTU VAV (ASHRAE & USGBC 2014, sec. 7.4.3):

- The fan power for all supply fans, return or relief fans, exhaust fans, or any fan-powered terminal units associated with heating or cooling systems, must be limited to 10% below the limit specified in ANSI/ASHRAE/IESNA Standard 90.1, Table 6.5.3.1-1 (ASHRAE 2016a). This requirement supersedes ANSI/ASHRAE/IESNA Standard 90.1, Section 6.5.3.1 and Table 6.5.3.1-1 (ASHRAE 2016a).
- The efficiency of fans at the design point of operation must be within 10%. This is less than the 15% specified in ANSI/ASHRAE/IESNA Standard 90.1, Section 6.5.3.1.3. Otherwise, all the exceptions for fans listed in ANSI/ASHRAE/IESNA Standard 90.1 apply (ASHRAE 2016a).
- The EA energy recovery effectiveness must be equal to or above 60%. This is greater than the 50% specified in ANSI/ASHRAE/IESNA Standard 90.1, Section 6.5.5.1 (ASHRAE 2016a).
- The automatic control of HVAC and lights in hotels or motels with over 50 guest rooms must meet the following requirements:
    - Within 30 minutes of all occupants leaving the room, the automatic controls must reset the setpoints by raising or lowering the temperature by at least 5.0°F (3.0°C) from the occupant's setpoint.
    - Within 30 minutes of all occupants leaving the room, all ventilation and exhaust systems must turn off. An automatic preoccupancy purge cycle provides outdoor air ventilation (ASHRAE & USGBC 2014, section 8.3.1.6).

Performance – AHRI Standard 340/360

The Air Conditioning, Heating, and Refrigeration Institute (AHRI) is the trade association representing manufacturers of air conditioning, heating, commercial refrigeration, and water heating equipment. AHRI develops standards for and certifies the performance of many products for sale in North America and internationally, for example, residential and commercial air conditioning, space heating, water heating, and commercial refrigeration equipment and components.

AHRI's globally recognized and industry respected certification program assists equipment and component manufacturers in certifying products and assuring that equipment performs accurately and consistently.

One standard for performance rating of commercial and industrial unitary air conditioning and heat pump equipment is AHRI Standard 340/360 (AHRI 2015).

The purpose of AHRI Standard 340/360 is, for commercial and industrial unitary air conditioning and heat pump equipment, to establish the following details (AHRI 2015):

- Definitions
- Classifications
- Test requirements
- Rating requirements
- Minimum data requirements for published ratings
- Operating requirements
- Marking and nameplate data
- Conformance conditions

The scope of the AHRI Standard 340/360 certification program covers 50 Hz and 60 Hz equipment, including the following equipment (AHRI 2015):

- Unitary air conditioners and heat pumps from 65,000 Btu/lb to less than 250,000 Btu/lb. This includes the following systems:
    - Single packaged and split systems
    - Air-cooled and water-cooled
- Air-cooled air conditioning condensing units from 135,000 Btu/lb to less than 250,000 Btu/lb (AHRI 2015)
- Air-cooled single packaged unitary air conditioners from 250,000 Btu/lb to less than 760,000 Btu/lb

The following certified ratings are verified during tests of AHRI Standard 340/360 (AHRI 2015).

For unitary air conditioners:

- Air-cooled, water-cooled, and evaporative-cooled from 65,000 Btu/lb to less than 250,000 Btu/lb:
    a. Cooling capacity, Btu/lb at standard rating conditions
    b. EER Btu/W h at standard rating conditions
    c. IEER Btu/W h at standard rating conditions
- Unitary air-cooled packaged air conditioners from 250,000 Btuh to less than 760,000 Btu/lb:
    a. Cooling capacity, Btu/lb at standard rating conditions
    b. EER Btu/W h at standard rating conditions
    c. IEER Btu/W h at standard rating conditions

For air source unitary heat pump equipment:

- Air-cooled from 65,000 Btu/lb to below 250,000 Btu/lb:
    a. Cooling capacity, Btu/lb at standard rating conditions
    b. EER Btu/W h at standard rating conditions
    c. IEER Btu/W h at standard rating conditions
    d. High temperature heating standard rating capacity, Btu/lb at 47.0°F
    e. High temperature coefficient of performance, COPH, W/W, at 47.0°F
    f. Low temperature heating standard rating capacity, Btu/lb, at 17.0°F
    g. Low temperature coefficient of performance, COPH, W/W, at 17.0°F

Conformance to the requirements of the maximum operating condition test, cooling low temperature operation test, insulation efficiency test (cooling), and condensate disposal test (cooling) are also initially verified by test for manufacturers applying to the AHRI ULE certification program (AHRI 2015).

Safety – IEC 60335-1 and IEC 60335-2-40

The IEC Standard 60335 is an international document created by the technical committee of the International Electrotechnical Commission (IEC). This standard will replace the UL Standard 1995 in 2022. The IEC is a non-profit, non-governmental organization that prepares and publishes standards for electrical and related technologies. It is a world-wide organization promoting standardization and international cooperation.

Part 1 (IEC 60335-1) of the standard is universal and contains all general requirements for electrical/electronic products covered by the standard. The fifth edition of this standard was adopted on October 31st, 2011 by a tri-national committee (Canada, US, Mexico). Part 1 works in conjunction with Part 2 to cover aspects that are very specific to the appliance in question (IEC 2010).

Part 2 (IEC 60335-2-40) of the standard regulates product markings, instructions, construction, materials, components, and testing, for products included in the following categories (IEC 2018):

- Electrical heat pumps
- Air conditioners
- Dehumidifiers

The second edition of this standard was released in October 2017 and will be the default standard when UL 1994 becomes obsolete. It builds upon requirements and testing from UL 1994 but expands requirements to ensure safety and standardization (IEC 2010 and IEC 2018).

Consortium for Energy Efficiency (CEE)

The Consortium for Energy Efficiency (CEE) is a non-profit, Boston-based group made up of US and Canadian program administrators. It is dedicated to developing cutting-edge strategies for accelerating commercialization of energy-efficient solutions. These solutions are primarily for electric and gas consumption (CEE 2016).

CEE's focus is to accelerate energy efficient products and services in targeted markets. Reducing energy use can avoid the need for building new power plants. CEE often develops specifications for products with different tiers of efficiency. Utility companies can offer incentives, such as rebates, for users of products with higher tiers of efficiency (CEE 2016).

CEE specifications exist for both residential and commercial HVAC equipment. CEE's standard covers variable refrigerant flow (VRF), heat pumps, water source heat pumps, RTUs and split systems. There are three tiers of efficiency for air-cooled RTUs up to 63.3 tons: Tier 1, Tier 2, and Advanced Tier. For units above 63.3 tons, there are only Tier 1 and Tier 2 efficiency levels (CEE 2016).

CEE's specification was established in 1993. Since 1993, there have been seven advances in performance levels. In 2018, a new advance to the specification is expected that reflects DOE efficiency 2018 levels (CEE 2016).

Canadian Standards Association (CSA)

The Canadian Standards Association (CSA) has a registered CSA Mark and is the certification agency for electrical equipment sold or installed in Canada.

In North America, for HVAC equipment, CSA is active in electrical and gas safety standards, testing, and certification. Beginning in 1996 through 2003, CSA and UL have worked to accept each other's certification. Coupled with Canada and US adoption of international ANSI and ISO standards, this makes CSA and UL certification practically identical (CSA 2003).

Installation

Variable air volume (VAV) for rooftop units (RTUs) installation involve a series of stages. Some stages apply to all RTUs, while others are particular to RTU VAVs.

RTU Installation

RTU installation includes the following stages:

- Rigging and lifting
- Erecting curbs
- Rating short circuit currents
- Multipiece construction

Rigging and Lifting the RTU

RTUs are large and complex pieces of equipment. It is important to safely and effectively install them. This section describes some general guidelines for installation, including rigging and lifting, and rooftop curbs.

For safety reasons, rigging and lifting is performed only by licenced, qualified, and experienced engineers, carried out within the bounds of local laws and codes.

There are also general equipment guidelines for rigging/lifting of HVAC units. Generally, rigging and lifting an RTU, engineers take a number of important steps.

⚠ WARNING

The following section is only a description of general installation. For actual installation, including rigging and lifting, consult the installation manual that ships with the RTU. Improper installation can create a condition where the operation of the product could cause personal injury or property damage.

Before rigging and lifting, determine the weight and center of gravity of the load. Confirm the load matches the type and model number of the unit being rigged or lifted. If uncertain of the weight, consult the manufacturer. After determining the load, compare it to the rated capacities of the rigging, lifting equipment, and other essential support equipment.

Having determined the weight, choose hitches and slings according to the type of rigged load. For hitches, use a spreader bar when installing taller units. Pay detailed attention to the rated capacities of the selected sling and hitch. Rated capacities for a sling type can change based on the type of hitch. Avoid shock loading and remove slings from service if there's any evidence of damage.

Check the sling angle and the tension applied to the hardware. The tension on the sling increases when the angle between the load and sling decreases. The equipment-rated capacity decreases as the unit deviates from a linear position. Prior to each lift, check the sling manufacturer tag for rated capacities of angles and weight.

Select the type of essential load equipment for rigging such as hooks, eyebolts, shackles, hoist rings, and turnbuckles. Review rated capacities for the hardware. Rated capacities can vary by manufacturer and by supporting hardware.

For each type of hardware, maintain and review an inspection checklist. Ensure that each piece of hardware or sling has its manufacturer tag, and do not use equipment without tags. Before using the lifting equipment, hardware inspections are critical. Any equipment wears out over time and becomes unable to support rated capacities.

When rigging and lifting, avoid tilting loads. A balanced load is important during each lift. A low centre of gravity helps keep loads stable. Test each load by slowly raising it a few inches above the ground, then check for sling position, balanced load, and potential failures.

Finally, re-rig the load and test again. Do not lift the load unless completely satisfied. If anything seems unsafe during the operation, give the stop signal immediately. For the centre of gravity of asymmetrical loads, consult the manufacturer.

Rooftop Curbs

A roof curb serves a multitude of purposes, for example:

- It supports the RTU
- It transfers the RTU load onto the roof's structure
- It connects the unit ductworks for vertical supply and return units

Unit manufacturers supply roof curbs to the general contractor or building manufacturer. An experienced erector installs them. The erector follows the installation instructions for assembling RTUs on the curbs. Curb manufacturers can provide curb adapters to allow a different model replacement RTU to use existing roof penetrations or convert a unit from vertical discharge to horizontal discharge.

Curbs come in many styles, depending on their purpose. They can be welded into a single assembly or shipped as a knockdown curb that must be assembled on site.

Typical curbs provide full perimeter support. However, some only provide perimeter support for the evaporator end with only a single end support for the condenser end. They can be fully enclosed, with weatherproof panels under the condenser end. They can also be open curbs with no panels under the condenser and relying on the roof surface to provide weather protection under the condenser.

Figure 49: Curb Structure – Example

There can also be specialty curbs that include seismic or wind curbs. There can also be isolation curbs that prevent vibration transmission into the building structure.

Before installation, the erector reviews curb drawings and specification for information on such features as unit dimensions, weight, cross braces, component details, and the dimensions of supply and return opening layouts.

Roof curbs match the exact footprint of each of the provided units. The curb is structurally capable of supporting intense loads and designed such that drains and power lines do not penetrate roof flashing.

The roof flashing is not directly attached to an RTU. The curb is furnished with a wood nailer to permit mechanical attachment of the flashing material.

The general recommended nominal curb height is at least 14 inches. Typically, curb manufacturers also offer custom curbs of greater height. The curb material is galvanized steel and may vary from 18 to 14 gauge based on the unit load and quantity of the cross bracing provided.

When installing, the best practice is to use elevated roof curbs or dunnage steel to support the unit and raise it to specific heights. When selecting roof curbs, it is also important to consider access and ensure that the selection meets local codes and regulations.

RTUs typically include an integral design base rail with marked lifting points and connections for condensate drainage. A recessed curb mounting surface provides a continuous surface for field application of curb gasketing. This gasketing creates a weather-tight seal between the RTU and the curb.

Prior to setting the units, the erector connects ductwork to the curb. Both unit and curb orientations are critical during installation. Before installation, the erector also reviews the airflow, marking detail drawings on the curb or RTU.

Before setting the equipment, the erector also ensures the roof curb is square and level and applies a sealing material for a watertight connection between the unit and the curb. Sealing material prevents moisture and air leakage in the duct system. Sealing material also prevents noise transmission and vibration in the building by using flexible collars when connecting ductwork.

In seismic zones, the erector installs a seismic isolation curb to support unit vibration and to offer seismic or wind restraint.

Finally, seismic, vibration isolation, and wind resistant curbs are not the same. Following are definitions of the various types from Stancato 2013's article on the website of MicroMetl, a roof curb manufacturer (Stancato 2013):

Vibration Isolation Curbs – Isolation curbs contain strong steel springs with a 2 inches deflection. They are designed and engineered to control noise and vibration of packaged, curb-mounted mechanical equipment from the roof structure.

Wind Curb – These are built to secure the RTU to the curb in high wind conditions. They gained recognition following the devastation of South Florida from Hurricane Andrew. The wind curb is PE stamped and requires field anchoring.

Specialty Curbs – Certain applications require specialty curbs to meet code regulations or structurally calculated curbs. Sometimes, the building code calls for a calculated or seismic-rated roof curb that is approved and stamped by a professional engineer. These requirements are often based upon the California Building Code and the International Building Code (IBC).

Short Circuit Current Rating

When installing the RTU into an electrical supply, it is important to consider the RTU's available fault current of the supply and the short circuit current rating (SCCR). In the event of a short circuit fault, the electrical supply momentarily allows a large amount of current to flow before an overcurrent protection device in the supply trip. This is the supply's available fault current.

The SCCR is the amount of fault current a device can safely handle. When installing an RTU into an electrical supply, the available fault current of the supply must be less than the RTU's SCCR.

Multipiece Construction

Generally, RTUs are mounted on the roof of a building or structure. In many cases, the building is multiple stories. It is very common for RTUs to be mounted on buildings with two to three stories. Although not as common, many RTUs are mounted on buildings with up to five or even seven stories.

In many cases, the RTU is mounted away from the edge of the building. When placed like this, the RTU's vertical supply and return ducts run as close as possible to elevator shafts, and can reach the various floors.

Generally, a crane lifts the RTU to the roof. In some cases, a helicopter lifts it. When using a crane, the crane size required to lift the rooftop depends on several factors, for example:

- The RTU's weight
- The building's height or lift
- The distance of the RTU's intended location from the edge of the building
- The crane's own location

Cranes range in size from 15 tons up to 400 tons. The larger the crane the higher the cost. In some areas, larger cranes can be harder to find.

For light commercial RTUs, helicopters are common. However, because of size and weight, they are not as commonly used for lifting large RTUs. The larger the lift, the larger the helicopter. As with cranes, a larger helicopter is a higher cost and harder to find.

To address these challenges, some RTUs can be shipped and lifted to the roof in multiple pieces. This facilitates shipment and minimizes the size of the required crane. This can also make it possible to use a helicopter lift. A multi-piece RTU provides greater flexibility on its placement on the roof, for example, it allows the use of commonly found cranes.

Multi-piece rooftops can be reconfigured into one piece on the curb. For the RTU's multiple pieces, engineers reconnect the following components:

- The electrical power and control wiring
- The refrigeration system piping
- The unit casing and baserail

Generally, the manpower cost of reconfiguring the unit into one piece is less than the cost of a large crane. Once reconfigured into a single piece on the curb, multi-piece RTUs are as reliable as RTUs shipped as a single unit.

Dual Point Power

RTUs can be installed on facilities that support critical needs, for example, medical office buildings, life safety facilities, data centers, and 911 call centers. To maintain site power supply during main power grid outages, these facilities use emergency generators. These generators are sized to handle only the facility's critical power needs. As a result, they may not have the capacity to service the RTU's entire electrical needs.

For these facilities, RTUs should be provided with a dual point power terminal block capability. One terminal block is circuited to the supply and exhaust fans (if applicable) and unit controller transformer. The transfer circuit of the facility emergency backup generator is wired to this terminal block. This wiring maintains the RTU fan operation during emergency generator operation and allows air circulation within the facility.

The second terminal block is wired to facility main power and is circuited to the compressors, condenser fans, and electric heat (if applicable) to allow normal RTU operation.

Post Shipment Documentation

Once an RTU ships, several pieces of information are required by various parties, for example, trades, facility operators, and owners. This information is often answered in documentation supplied to one individual. However, it can be misplaced or never received by another individual. Documentation can include the following information:

- Weights and rigging information
- Controls integration
- Sequence of operation
- Maintenance manuals
- Parts documents
- Service bulletins
- Warranty period
- The unit's performance ratings (for situations in which parties need to modify design parameters)

One solution is to specify that all documentation for unit maintenance, service, repair, and operation is available through a single online repository, a repository that contains information specific to the actual unit installed.

VAV Installation

For hanger rod diameters up to 3/8 inches, VAV terminal units can have hanger brackets. Hanger straps are an alternate method of suspending the units. To ensure they do not interfere with working components or access panels, hanger straps can be mounted directly to the sides and bottom of equipment casing using screws that penetrate the unit cabinet no deeper than 3/8 in. They are not secured to electric heaters, coils or control enclosures. Hanging equipment uses the support method in the job specifications prescribed for rectangular ducts.

Duct Connections

When fastening ductwork to an equipment cabinet, the fasteners must not penetrate it more than 3/8 inch. Fasteners that penetrate the cabinet deeper than 3/8 inch may come in contact with live electrical parts or pierce other components within the casing, causing damage.

All duct connections are configured and installed in accordance with Sheet Metal and Air-Conditioning Contractors National Association (SMACNA) guidelines and local code requirements. Engineers install the duct before there is any equipment inlet or discharge.

The diameter of the inlet duct for round valves must be equal to the listed size of the equipment. To allow the round ductwork to slip over the air valve inlet collar, the round air valve inlet collar of the equipment is 1/8 inch smaller then the listed size. Ductwork is not inserted into the air valve inlet collar.

When making ductwork connection to air valve inlet collar and insulating air valve inlet, engineers have to take caution not to damage or remove the flow sensor connections. These sensors are vital to unit control. They put insulation around the entire inlet collar (all the way to the equipment casing).

The following discharge duct connections are permitted:

- Straight flanged
- Slip and drive
- Drive and screw

If equipment is installed in a location with high humidity, external insulation around the heating coil is also installed.

Sound Critical Applications

We do not recommend flexible duct connectors on equipment discharge. These fitting's sagging membranes can generate noise because of turbulence and higher air velocities. Also, lightweight membrane material allows noise to break out. This increases sound levels in the space below.

Coil Connections

Hot water and steam coils are male sweat connections. Engineers use the appropriate brazing alloy for system temperature and pressure. For specific connection sizes, they refer to construction submittal drawings. The maximum hydronic system operating pressure must be below 300 psig. The maximum steam system pressure must be below 15 psig.

Electrical

All field wiring complies with NEC and all local codes. Electrical or control wiring diagrams are located on the control enclosure box. All electric heaters are staged according to specifications. To assist balancing the building electrical load, the installing electrician rotates the incoming electric service by phase. To size wire feeders, the minimum circuit ampacity (MCA) determine's the equipment's maximum operating load. The maximum overcurrent protection (MOP) is the largest breaker or fuse in the electrical service panel that is able to protect the equipment. Engineers only use copper conductors.

Operation

VAV terminal units have some general operational guidelines.

Start-Up

Only qualified individuals perform start-up and service, always taking thorough safety precautions. They check all electrical work is finished and properly terminated. They also check all electrical connections are tight and the proper voltage is connected.

Phase Balancing

The alternating current (AC) power imbalance cannot exceed 2%. Engineers ensure the AC power is within 10% of the rated voltage at the rated frequency and within 5% of the rated frequency at the rated voltage. See the equipment nameplate for ratings.

If the frequency variation does not exceed 5% of rated frequency, there can be a combined variation of 10% (the sum of absolute values) rated values in the voltage and frequency. Equipment with electric heat requires a minimum of 0.1 iwg downstream static pressure (SP).

Before start-up, engineers obtain and thoroughly understand the project control sequence or wiring diagram. If they are using factory supplied analog or direct digital control (DDC) controls, they consult the applicable operation manual for start-up and balancing information.

Service

Some rooftop units (RTUs) come with a number of features and functions that assist servicing so that it is simpler, easier, and more efficient:

- A start-up wizard
- Fault diagnostics detection (FDD)
- Access doors and lights
- A variable frequency drive (VFD) bypass
- Serviceable filter driers
- Isolation valves
- A mobile access portal (MAP)
- An end user interface
- Dirty filter transducers
- A grease line
- Belt guards
- Various condenser functions, such as a cleaning hatch, roof safety tie-off, and overflow switch

Start-Up Wizard

An RTU's initial start-up is critical for the unit to have a long trouble-free life. During the start-up process, technicians check areas such as voltage, compressor rotation, and amp draws of various components. Most manufacturers require trained service technicians to start units with a manufacturer-based start-up checklist.

However, even with such detailed paper checklists, service technicians often miss or overlook steps within the start-up process. To avoid this, and to simplify the step-by-step process, modern technology includes a start-up wizard. The wizard presents a technician with a well-defined sequence of steps that the technician must perform to start the RTU, for example:

- A series of questions for configuring the RTU
- A sequence of automatic self-tests
- A sequence of manually verified tests (required for start-up documentation)

For each of the automatic self-tests, the system records the following data:

- A self-test pass or fail
- A time stamp for the test
- Other relevant system data

The wizard also requires the technician perform measurements to confirm proper unit operation. Some values are automatically collected by the controller. For other values, the wizard prompts the user to measure and enter them.

After the testing sequence completes, a report with the recorded results is stored in non-volatile memory on the controller. These results can be exported via USB.

All system safeties (for example, the compressor safety chain, compressor low pressure, or high duct pressure) remain in effect. During the start-up wizard, these safeties can shut down the unit at all times.

Fault Diagnostics Detection (FDD)

FDD is a tool for maintaining RTUs (and HVAC equipment generally) at an optimal level of performance and reliability. FDD can significantly save energy on an ongoing basis, supporting efficient maintenance, extending equipment life, and providing more consistent occupant comfort and indoor air quality (IAQ).

FDD pinpoints root causes of problems so that they can be corrected. It detects faults and isolates them. It enables an RTU controller to predict faults before they result in such things as occupant discomfort, problems with efficiency, or even major failure.

Embedded FDD algorithms continuously run within the controller. The algorithms monitor the different types of input to precisely indicate faults and recommend ways of correcting them.

The feature is optional with Smart Equipment Controls (SEC) product lines. FDD algorithms can meet California Energy Code regulations (Title 24). FDD algorithms used to meet Title 24 are specific to economizer operation. However, FDD algorithms can also be used for the refrigeration side of the RTU.

During the start-up process, FDD calculates the efficiency and capacity of the equipment and generates a baseline for future measurements. The algorithms provide this type of information during the equipment life cycle. You can view this information from the unit control board local display, MAP, or by connecting through a building management system.

These two calculated values (efficiency and capacity) enable owners to make smart decisions regarding their equipment. Owners can quickly and easily see when the equipment is not performing to the start-up and commissioning baseline.

Here are some examples of FDD status values:

- Refrigerant Low
- Excessive Refrigerant Flow
- Inefficient Compressor
- Refrigerant Flow Restriction
- High Side Heat Transfer Problem
- Low Side Heat Transfer Problem
- Reduce Evaporator Airflow
- Add Charge
- Insufficient Refrigerant Flow
- Recover Charge
- Safe And Reasonable Performance
- Non-Condensable Present
- Efficiency Index
- Capacity Index

Access Doors

Technicians often need to access internal sections within an RTU. They can do this in multiple ways. The simplest is a removable panel. However, removable panels are unsuitable for larger cabinet units because the panels get bigger and become more difficult for service technicians to manage. Instead, access doors are typically supplied on the larger pieces of equipment, especially those that require periodic access.

Access doors come in many different sizes, commonly ranging from 4–36 inches. The size frequently depends on what the door provides access for. For parts of the RTU such as side-loaded filters, a 4-inch door may be enough. For parts such as a fan or motor, larger doors may be required. Larger doors enable access for inspection but also enable technicians to remove or pull the motor when there's a failure.

Access doors on RTUs are fabricated to reduce the chance of injury to technicians. Most manufacturers require RTUs be shut down before technicians access them. However, technicians can act differently in practice in the field. On pressurized compartment doors, a latch with secondary pawl at 90° enables the door to open about 1 inch before contacting the door column, followed by a 90° rotation back to release the second pawl and fully release the door. When the cabinet pressure is relieved, this design reduces the chance of the door blowing open and hitting the technician.

Doors should be supplied with a method to secure the door open, such as a chain and slotted anchor. These chains and anchors keep doors in a fully secured open position for maintenance access to the unit's control box and moving parts. Take caution when conducting maintenance in windy conditions, especially where there are no stronger means of keeping the door open.

As RTUs are often set on curbs, door handle access is important for proper service. It is common to use multiple independently operated handles to close doors on large units. For 6–8 feet high units with 24–48 inch high curbs, the top handle can be out of reach of service technicians standing on the roof. Independent handles require technicians to bring ladders to the roof every time they want to inspect the unit. Alternatively, manufacturers offer single handles that can latch at multiple points, yet also be moved up and down. This eases the access challenges from multiple point handles.

In some situations, a locking mechanism may be required that prevents a basic level of entry. Entry can be limited with handles requiring a tool to operate or by suppling pad-lockable handles.

Service windows or viewports can also be supplied to allow a technician to view and confirm the operation of an RTU. At a minimum, viewports are designed to withstand static pressure (SP) developed by the unit, temperature, and UV exposure without leaking or degrading.

Lights

RTUs can have interior lighting to assist technicians in servicing the unit. The light system is often separate to the RTU's main power circuits. The power circuits are locked and tagged for normal maintenance, and the light system can need to run during such maintenance.

Lighting technology has changed over the years and interior lights can come in many shapes and forms. To service parts (such as cleaning coils, actuators, fan motors, or wiring) it is recommended that there is interior light – for example, an LED network bright enough for common maintenance and service without a hand-held flashlight.

Variable Frequency Drive (VFD) Bypass

If a VFD fails, a VFD bypass allows the motor to continue operating on line power without modulating or conditioning the power going to the motor. Bypass operation is a short-term solution for operation while the VFD is replaced. It is short term because the start-up voltage to the motor in bypass mode is significantly higher than a normal VFD start-up.

The bypass mode is integral to the drive and is unlikely to be configured so that it starts from an external source. The bypass mode only suits if the driven motor can operate at 100%.

A redundant VFD can be provided as an alternative to the (primary) VFD bypass. In this case, if the primary VFD is faulty or unresponsive, the unit controller automatically uses the backup VFD to run the motor. This is a superior backup method because the device can continue running at the desired speed.

Condenser Roof Safety Tie-Off

RTUs require condenser fans and motors to overcome start/stops and long operational hours, all while the RTU is subject to the natural elements. These factors can often lead to failures in components such as the condenser fan and motor. The rooftop manufacturer is responsible for allowing repair and replacement of these components.

Frequently, on large RTUs, the repair and replacement of the condenser fan and motor is completed from the roof of the unit. The combination of the heights of the roof curb and the unit can force the technician to work on the condenser fan and motor at heights of 6-8 feet or more off the building roof deck.

It is imperative that, during repair and replacement, the technicians are able to work in a safe environment. There are multiple ways to help ensure the workers are safe. However, one method is to provide a safety tie-off on the RTU that the technician can lock into.

Convenience Outlet

The National Electric Code® (NFPA 2017, sec. 210.63) requires requires a service receptacle outlet for all outdoor HVAC equipment installations:

> A 15A or 20A, single-phase 125V receptacle outlet must be installed at an accessible location for the servicing of heating, air-conditioning, and refrigeration equipment (HACR) on rooftops and in attics and crawl spaces. The receptacle must be located within 25 ft and on the same level of the heating, air-conditioning, and refrigeration equipment. The receptacle outlet cannot be connected to the load side of the equipment disconnecting means.

The unit manufacturer provides this receptacle outlet as an optional feature. Factory installing the receptacle outlet on the RTU is an economical installation alternative to having the electrical contractor provide an independent dedicated outlet branch circuit on the building roof.

A 15 amp, single phase 125V receptacle outlet is mounted on the unit exterior. The outlet is easily accessible to service personnel. It is factory wired to the line side of the unit power terminal block or disconnect option, with ground fault circuit interrupter (GFCI) protection for use in outdoor applications. The outlet is factory provided with the necessary fuses and step down transformer to maintain the single point power connection to the RTU.

Serviceable Filter Driers

Filter driers are a key component in any refrigeration or air conditioning system. Filter driers have two functions that improve the RTU's longevity:

- They remove moisture and acids
- They provide physical filtration to trap contaminants and all types of foreign substances

Of these functions, the primary one is to remove moisture from a refrigeration system. Moisture can come from many sources. Here are some examples:

- Trapped air from improper evacuation
- System leaks
- Motor windings of hermetic compressors.
- Improper handling of polyolester (POE) and polyvinyl ether (PVE) lubricants.

In particular, POE and PVE are hygroscopic and can pick up moisture from their surroundings. Moisture in the system can hydrolyze the POE lubricant, forming organic acids. These acids, if they exist in significant quantity, react with system materials and can adversely affect the component operation. The surface of the desiccant is charged positively with cations, that act as a magnet and therefore adsorb polarized molecules, such as water, and hold them tightly on the structure.

A filter drier with a molecular sieve (with a honeycombed core and activated alumina desiccant) can minimize such moisture and acid inside the system. The filter drier can be a spring load desiccant design that uses multiple layers of a fibrous media to capture circulating particulate.

Replaceable core filter drier shells are designed for flexibility over a wide range of applications. In single or multiple-core applications, cores can be loaded individually. This enables easy installation in tight spots. Furthermore, by installing isolation valves by both connection ends, cores can be easily replaced without any major brazing work at the site.

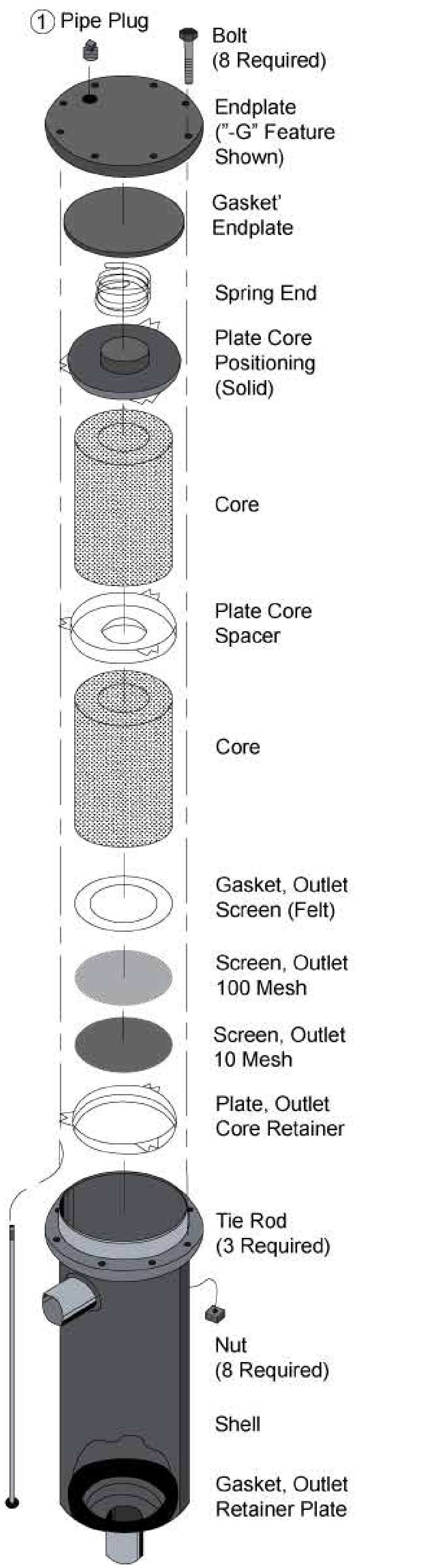


Figure 50: Serviceable Filter Drier

Isolation Valves

Also known as shutoff valves, isolation valves open or shut fluid flow in a system, usually for maintenance or safety. These valves suit applications in which bidirectional fluid flow is required, for example, liquid, suction, and hot gas lines. They are designed to operate over a broad temperature range.

Isolation valves are made of a forged brass body with extended copper fittings that can be fully closed or open in 1/4 turn. See *Figure 51* for the valve's internal function (the figure is for functional illustration only).

The installation and function of isolation valves can vary by refrigerant line, for example:

- **Liquid Line:** The valves isolate a replaceable filter drier used to close the flow replacing the dried core.
- **Discharge Line:** Isolation valves are optional for HGRH. A discharge line controls or services (as required) a 3-way valve and a hot gas bypass (HGBP) valve branching from it. Isolation valves modulate these valves.

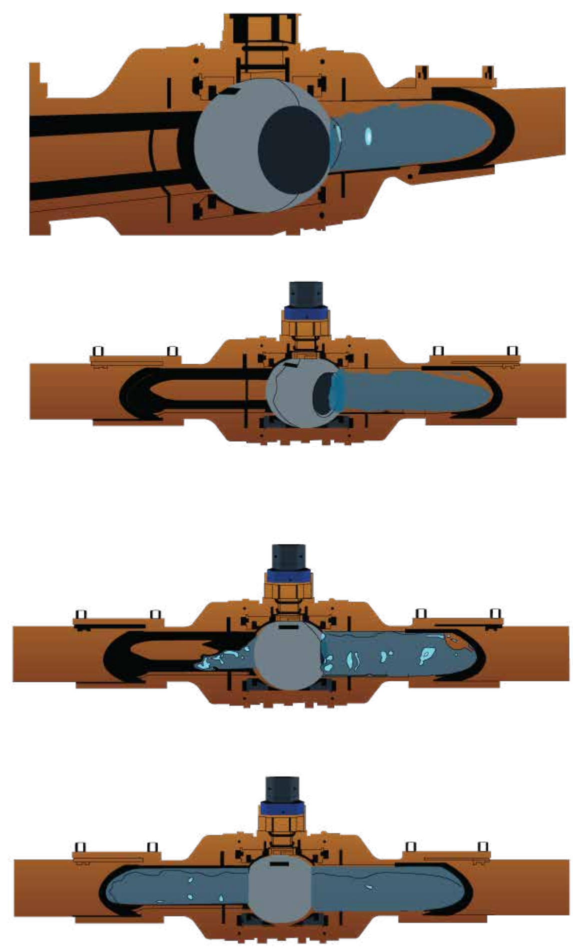


Figure 51: Valve Internal Function

- **Suction Line:** When replacing hermetic compressors and servicing semi-hermetic compressors, isolation valves isolate the compressor from the system routing. This helps replace the compressor without evacuating the full charge from the system.

The application's requirements determine the valve choice.

Condenser Cleaning Hatch

In HVAC units, clean condenser coils are critical for energy efficiency. Pacific Gas and Electric states that a dirty condenser coil that "raises condensing temperature from 95.0–105.0°F cuts cooling capacity 7% and increases power consumption 10%, with a net (compressor) efficiency reduction of 16%" (PG&E 1997, 3).

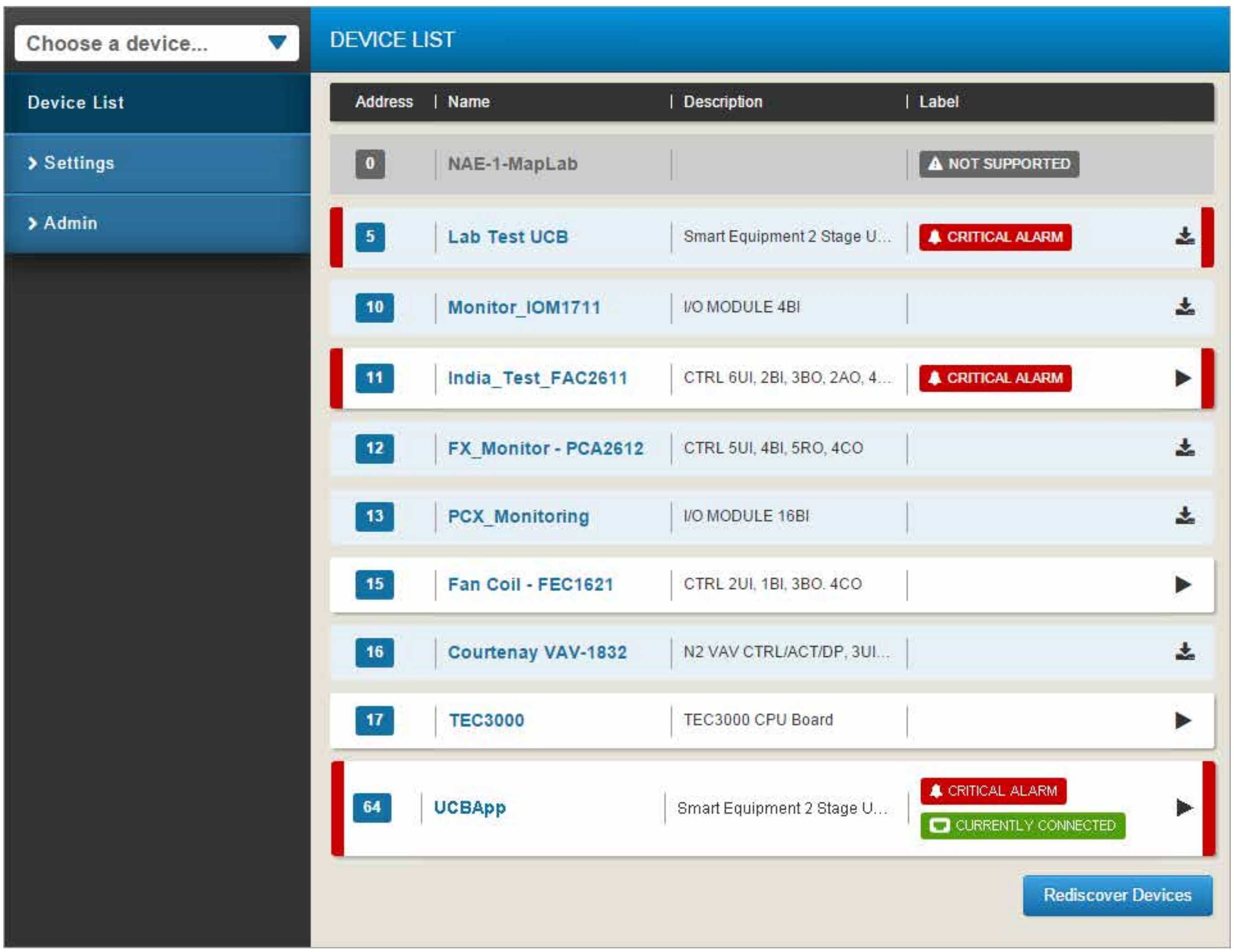
Condenser coil sides that face away from the condenser fans are typically easy to access and clean. Condenser coil sides that face the fans are often difficult to access and clean. A condenser cleaning hatch can be provided for access to the condenser coil side facing the fans.

Mobile Access Portal (MAP)

A MAP Gateway is a pocket-sized web server that provides a wireless mobile user interface (UI) to Metasys® field controllers, Facility Explorer® (FX) field controllers, TEC3000 Series thermostats, and Smart Equipment RTUs. The UI can be viewed on a browser of a phone, tablet, or computer. See *Figure 52* for an example of the MAP UI.

The MAP Gateway ships from the factory with a base set of features that enable users to access, view, edit, and override key information from all devices connected on a common BACnet® MS/TP field bus. Along the line of sight, the wireless connection on the MAP Gateway enables users to be at a distance up to 100 feet (31 meter) indoors and up to 300 feet (91 meter) outdoors.

The MAP Gateway can also be permanently mounted, powered with an optional separate power supply, and, for remote connection to a MS/TP field bus of devices, connected to an Ethernet access point.



LD27246

Figure 52: Mobile Access Portal (MAP) – Device List Menu Screenshot

There are features and benefits to the functions of the standard factory release MAP 4.0:

- **Simple Browser-Based UI**: The MAP Gateway has a consistent, intuitive, menu-driven set of browser-based views. These views enable quick navigation between connected devices and drill-down into point and feature data supported by each device type.
- **Secure Wi-Fi Connectivity to Multiple Platforms:** The MAP Gateway can communicate securely through Wi-Fi to smart phones, tablets, and personal computers. The browser-based user interface automatically conforms to the connected platform's size constraints.
- **Carry with You or Leave on Site:** The MAP Gateway can be a basic, portable commissioning tool to view, adjust, or override key points on supported field controllers connected by an MS/TP network. For connected devices, when permanently mounted and connected to an MS/TP network, the MAP is a secure local display option.
- **Access to Advanced Application Field Controller Schedules, Alarms and Trends:** The MAP Gateway allows the viewing of alarms, events, and trends, and the modification of schedules for advanced application controllers.
- **Permanent Audit Log:** Allow the export and viewing of a log file of all user logins, transactions, and any events generated from the connected controllers.

Standard and High End User Interface

RTU VAV systems include UI devices to obtain status and operational details from both the RTU and the VAV boxes. There is normally a low end UI that provides detail on unit operation. This low end UI is a text-based UI. Normally, it has a membrane keypad for navigating the various data points of the unit.

A higher-end UI can be touchscreen and provide a graphical representation of the rooftop or, in some cases, the entire VAV system. The higher-end UI provides rich detail and can feature point animation as well as color changes for alarm data points. In some cases, these graphics are customizable.

Dirty Filter Transducers

Dirty filter transducers supply a signal back to the unit controller to identify that the filters are dirty and need to be changed. If the unit is tied into a BAS, the dirty filter transducer signals the unit controller that communicate to the BAS about the issue. If the unit is standalone, the transducer simply communicates to the unit controller that can be viewed through the RTU user interface.

Grease Lines

Grease lines keep the fan bearings lubricated. The grease lines are frequently placed in such a way that you can easily access them. This can lead to grease lines capable of being greased from one side of the fan. It is common to mount the connection to the grease lines on the fan frame on the side closest to the RTU access door.

Another approach is to extend the grease lines to the external of the cabinet. It is important that the service technician can visually inspect the grease lines to ensure that there is not a blockage within the grease line, and to ensure that lines are not over-greased.

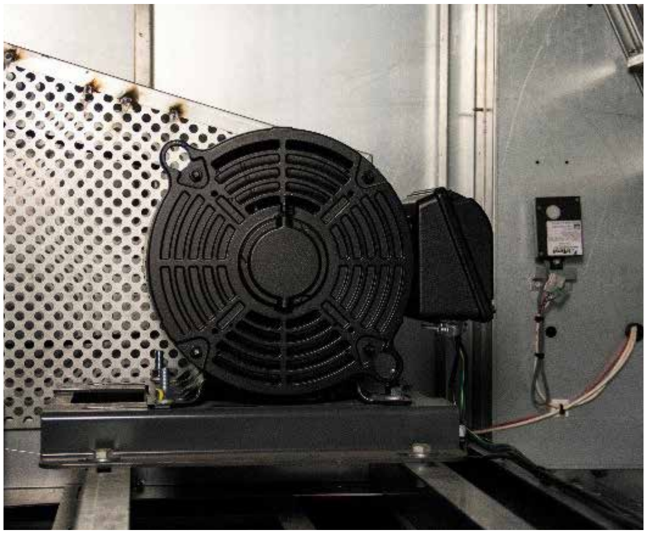
Direct-driven fans only have a motor bearing and do not have a fan bearing. The presence of only a motor bearing removes the need for a greasable fan bearing and the need for a grease line.

LD27203

Figure 53: Grease Line Connection Mounted on Fan Frame

LD27204

Figure 54: Grease Line



LD27205

Figure 55: Belt Guard

LD27237

Figure 56: Fan Inlet Guards

Belt Guards

Belt guards are supplied on most air-handler equipment. They protect the service technician from contacting the moving belt used to transfer motor energy into fan energy. The definition of satisfactory protection for a service technician from contacting the belts can vary based on customer desires.

It is commonly accepted that a service technician is sufficiently protected from contact with an operational belt by access doors that must be opened. However, an owner may want an additional layer of protection for the service technician. Belt guards can provide the next level of protection by reducing the chance of inadvertent contact with a moving belt.

Fan Inlet Guards

Before entering and servicing a unit, RTUs are shut down, locked out, and tagged out. Fan inlet guards provide an additional layer of protection for service technicians. These inlets can prevent major items from damaging the fans. However, they are not supplied as the only reliable protection from injury for an individual. When the customer specifies and requires it, the unit fan performance must be properly de-rated so that the design team can properly analyze horsepower, CFM, and SP requirements.

Condensate Overflow Switch

Where water collects from cooling coils or other components, drain pans can catch the condensate. In standard operation, the drain pans collect then remove the water from the unit. Drain pans are typically connected to gravity-fed drain seals, also known as P-traps. P-traps must be designed for the application's specific negative SP. If not, they do not work correctly and prevent the condensate from draining from the unit (see the manufacturer's P-trap design information in the installation manual).

If P-traps are not properly serviced and maintained, they can become clogged. When the traps are clogged, the drain pan can also become clogged. Water can spill out of the drain pan, out of the unit, and potentially down through the roof curb to penetrate the building. Water penetration can lead to repairs costing hundreds or thousands of dollars. Condensate overflow switches reduce the chance of a clogged drain pan leading to such water damage. Often, the condensate overflow switch is wired to send an alarm across the building automation systems (BAS) or shut down the unit and alarm the BAS.

For drains that are periodically serviced and inspected, it is rare to get a clogged line. However, for the rare occasions when it occurs, condensate overflow switches are a relatively low cost solution that provides an additional layer of protection.

Maintenance

VAV terminal units have some general maintenance features.

Damper actuator (optional) – A damper can have a factory-mounted floating actuator. The actuator mounts directly to the damper operating shaft. The actuator is not provided with and does not require any limit switches but is electronically protected against overload.

Manual override – To allow manual movement of the drain shaft, a button on the side of the actuator cover disengages the gear train. Releasing the button re-engages the gear train.

Mechanical angle of rotation stops – To halt the rotation of the damper blade before the damper blade reaches the damper stops, the stops can be field adjusted. The actuator can indefinitely stall in any position without harm.

External terminal strip – The external terminal strip is located on the top of the actuator. The terminals are designed for 26 to 16 gauge wires. Connections are numbered. For most installations, 18 or 16 gauge wire work well with the actuator.

Overload protection – The actuators are electronically protected against mechanical overload. In the actuator, an electronic circuit maintains the current at a level that, while providing adequate holding torque, does not damage the motor.

Adjusting the Stops

1. With a number 2 Phillips head screwdriver, loosen the two end stop screws. Be careful not to unscrew the captive nut under the slot.
2. In 2.5° steps, move the stops to the desired position and retighten the screws.

Inspecting the Actuator

1. Disconnect the actuator from the controller.
2. Rotate the actuator in a clockwise (CW) direction by applying 24 VAC to its COM and CW terminals.
3. Rotate the actuator in a counter-clockwise (CCW) direction by applying 24 VAC to its COM and CCW terminals.

   If the actuator moves in both directions, it is operational. If the actuator does not rotate, it is either at an end stop or there is a problem with the damper.
4. Loosen the set screw. This frees the actuator from the damper shaft.
5. Check that the damper shaft freely rotates and check that the actuator is not against the stop.
6. Repeat steps 2 and 3.
7. Replace the actuator if it does not rotate.

Damper Shaft – An indicator on the end of the damper shaft assists in determining the position of the damper blade. If the indicator is horizontal, the damper is completely open. The damper shaft is 1.2 inches in diameter.

Coil – The frequency of required cleaning depends on the system's operating hours, filter maintenance, efficiency, and dirt load. Coils can become externally dirty as a result of normal operation. Dirt on the surface of the coil reduces its ability to transfer heat, resulting in reduced performance and increased operating energy cost.

If the dirt on the surface of the coil becomes wet, there can be microbial growth (mold), possibly causing unpleasant odors and serious health related IAQ problems. Also, fins are fragile and their edges are sharp. Care must be taken to avoid damaging fins. Solutions are not used to clean coils. There are no drain pans to remove the collected solution.

Cleaning the Coil

1. Disconnect all electrical power to the equipment.
2. Tag and lock out the power source.
3. Access the coil through ductwork (or an optional coil access panel).
4. Use a soft brush and vacuum to remove loose debris from the coil sides. Do not use fluid or solvents to clean the coils. On this type of equipment, there are no provisions for collecting liquids.
5. Straighten any coil fins damaged during cleaning process with a fin comb.
6. Replace the ductwork (or access panel) and restore electrical power to the equipment.

Electric Heat

The manufacturer's electric heaters require little or no maintenance. Electric heaters come equipped with a primary auto-reset limit switch. These limit switches protect against overheating. When overheating occurs, the auto-reset limit switches automatically cut the heater off. When the elements have cooled down, they turn the heater back on.

Electric heaters also come equipped with a secondary one-time trip limit switch. If the secondary limit switches trip, they need to be replaced with a limit switch that has the same trip temperature as the one-time trip limit switch originally supplied with the electric heater. An optional manual reset secondary is available. This is reset by depressing the reset switch.

Minimum Operating Conditions

VAV requires the following minimum operating conditions:

- Airflow is at least 70 CFM/kW.
- External pressure is at least 0.1 iwg.

Replacing the Electric Heater Rack

1. Turn off the power supply.
2. Locate the element rack T-Plate inside the heater control enclosure.
3. Mark where the wires are connected. This ensures they can be reconnected on the new element rack.
4. Remove the wires and screws holding the T-plate in the control enclosure.
5. Remove the old element rack from the enclosure.
6. Insert the new element rack into the enclosure.
7. To secure the element rack to the enclosure, replace the screws and wires.
8. Close the enclosure cover.
9. Turn on the power.

Performance

RTU performance data is based on rigorous development testing, including tests for cooling, heating, electrical, airflow, and sound performance.

These are tests of only some of the possible performance points. However, the manufacturer's selection tools includes sophisticated algorithms that predict the RTU's performance. They also make it possible to select sophisticated RTUs with a variety of options for meeting application requirements.

Airflow is a critical performance criterion in RTUs. Factory testing is available on select families of RTUs to verify their performance against specific design conditions reported by the selection software. The performance is under laboratory conditions and is reported in terms of the following values:

- Supply fan BHP
- RPM
- Total SP
- Resulting airflow at standard conditions

This testing is usually available as a special quotation or request. It can add incremental lead time to the delivery of the RTU to the job site.

Alternatively, a copy of the end of line test report may also be available, showing the parameters checked on the RTU. The parameters can depend on the RTU configuration. The configuration can include the following parameters:

- Compressor
- Suction
- Discharge
- Heater
- Condenser fan
- Supply fan
- Exhaust/return

Additionally, the end of the line test report informs the RTU controller configuration and other useful data. This testing is also available as a special quotation.

Glossary

AFMS: Airflow measuring station. See Piezometer ring definition.

ASCE/SEI 7-16: Minimum design loads and associated criteria for buildings and other structures. The latest edition is 7-16 and is up dated every 6 years.

BHP: Brake horsepower. A measure of the power required to operate a fan at a given design point. Motor size should be selected based off of BHP as it accounts for fan efficiency.

CFM: Cubic feet per minute. The imperial unit for volumetric flow rate (for example, airflow).

Direct drive fan: A fan turned by connecting the fan wheel directly to the motor shaft. The RPM of the fan will equal the RPM of the motor.

Dual fans: On larger units, the fan segment is constructed with two fans operating in a 50/50 parallel arrangement. The dual supply fan offering is for unhoused plenum fans.

Fan curve: A unique plot of specific RPM curves at various airflow (CFM) and static pressure (iwg) points for a given fan type and size. The fan curve also contains intersecting lines for motor size (in HP). The point at which the design RPM curve intersects the system curve is the design point.

Fan inlet screen: A safety screen installed in the inlet cone of a fan. This device is primarily used in situations where operator safety is a concern. Inlet screens have a significant impact on the system performance.

Flex connector: A flexible fabric that is airtight, watertight and fire retardant and used in the interior of an RTU to connect the inlet or outlet of a fan to the intake or discharge wall (depending on fan type).

A flexible connector may also be used by a Contractor to connect the external ductwork to the RTU at the RA or SA openings.

Hurricane: A tropical cyclone with wind speeds occurring from 74 mph (119 kmh) to 190 mph (289 kmh).

IBC: International Building Code. The first edition of IBC Iwas issued in 2000 and it is updated triennially. The IBC replaces the three major US building codes: the NBC, the UBC, and the SBC. The code covers seismic and snow loads as well as wind loads (2017b).

ICC: International Code Council. Organization that develops and enforces building codes in the US.

Motor size (or HP): The point at which the design RPM curve intersects the system curve is the design point.

Motor synchronous speed: A term synonymous with direct drive fans, the motor synchronous speed represents the point at which the fan RPM is equal to the motor nameplate RPM.

NOA: Notice of Acceptance. Products that have satisfied and passed the tests (TAS) are given an NOA number. This certifies that the product can be used in hurricane prone areas.

Piezometer ring: A means to measure airflow across a fan, the piezometer ring consists of a ring mounted at the throat of the inlet cone and a static pressure tap mounted at the face of the inlet cone. Airflow is calculated by measuring the differential pressure between taps with a transducer.

Plenum fan: An SWSI type fan with an airfoil, backward inclined or backward curved impeller that typically does not have a fan housing around the wheel. Plenum fans are ideal to use in RTU applications where the fan is not the last component in the unit.

Shaft grounding ring (SGR): A device installed internal or external to the motor, the shaft grounding ring prevents damage to the motor bearings by diverting harmful shaft voltages and bearing currents commonly caused by a VFD to a ground.

Spring isolators: Springs attached to the fan skid to reduce the transmission of vibration and noise of the fan and motor to the building structure.

Static pressure (SP):

- **Internal:** A sum of all the pressure drops of the internal components of an RTU including, but not limited to, dampers, filters, coils, and specialty components. This is the responsibility of the RTU manufacturer and may or may not include opening pressure drops where the ductwork connects to the RTU.
- **External:** A sum of all the pressure drops of the external duct system for an RTU including, but not limited to, friction losses through straight duct and fittings, external filters or components, and diffusers. This is the responsibility of the Consulting Engineer and may or may not include opening pressure drops where the ductwork connects to the RTU.
- **Total:** The total combined sum of internal and external pressure drops. The fans in an RTU should be selected based on this number.

System curve: A plot of various airflow and SP conditions on a fan curve for a given “System." The system is defined as the RTU and all connecting ductwork, fittings, etc.

TAS: Testing Application Standard. Test procedures determine whether a particular product can withstand the hurricane load.

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