Providing insights for today's HVAC system designer

ENGINEERS NEWSLETTER

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Using Hot Gas Reheat for Humidity Control

Humidity control in buildings is a core component of occupant comfort. For decades, many buildings would simply rely upon coincidental (or "passive") dehumidification, which attempts to maintain zone humidity through the active control of zone temperature. This meant that systems were controlled to maintain the zone dry-bulb temperature at a desired setpoint and trust the resulting flow of cool, dry supply air was sufficient to offset both zone latent and sensible loads.

Some buildings require active humidity control because of occupant activities or purpose. Examples include zones where humidity-sensitive medical procedures are performed, the storage of rare artwork or items of historical significance, and zones like dining rooms, gyms, grocery stores and other dynamic applications with varying occupancy and/or high outdoor air loads. Another consideration is moisture management and the need to maintain a dry environment to protect the building structure and its furnishings from moisture damage that can occur with consistently high zone humidity levels.

The climate in some areas would necessitate humidity control. Traditional rules of thumb are dated and should be revisited on an application-specific basis.

Active on/off dehumidification control with hot gas reheat has been commercially available for many years and has become a popular feature for dehumidification applications. Advancements in packaged rooftop units now offer the designer increased flexibility: variable speed supply fans, variable speed compressors, and modulating hot gas reheat providing greater control over a broader range of applications impacting comfort.

Humidity Requirements in Standard 62.1 and 90.1

ASHRAE® Standard 62.1 "Ventilation and Acceptable Indoor Air Quality" has long discussed humidity requirements for occupied zones. Prior to the 2019 publication, the standard prescribed a maximum zone relative humidity of 65 percent at a specific design dehumidification condition.

In the 2019 revision of Standard 62.1, the zone humidity requirements in section 5.10 (section 5.12 in Standard 62.1-2022) have been significantly updated[1]. In that publication, and the subsequent 2022 publication, a maximum zone dew point temperature of 60°F is required, during both occupied and unoccupied periods, whenever the outdoor air dew point temperature is 60°F or higher.

[1] As of this publishing, the topic of indoor zone humidity is still being discussed by the Standard 62.1 project comittee.

There are several exceptions to the requirement:

1. Spaces equipped with materials, assemblies, coatings, and furnishing that resist microbial growth.
2. The dew point limit shall not apply during overnight, unoccupied periods that do not exceed 12 hours provided a maximum space relative humidity does not exceed 65 percent.

Figure 1 illustrates Standard 62.1-2022 exception 2 . The shaded region shows where zone humidity operation is prohibited, except during those overnight periods less than 12 hours.

See the Engineers Newsletter "ASHRAE Standard 62.1 Update" for more detail on the maximum zone humidity requirement.

Figure 1. Maximum indoor dew point temperature limit

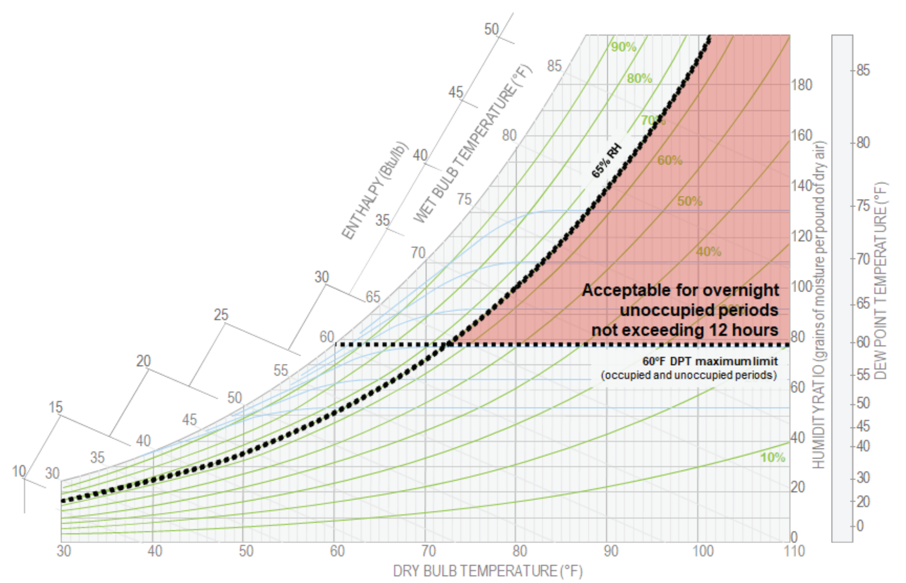


Figure 2. Example zone humidity when dehumidification is required

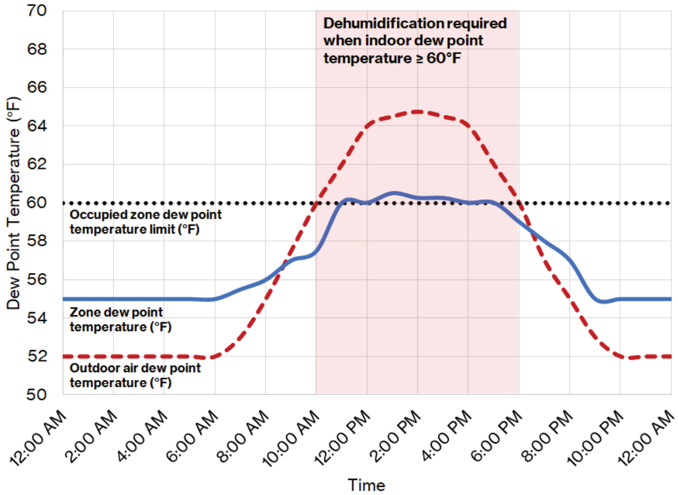
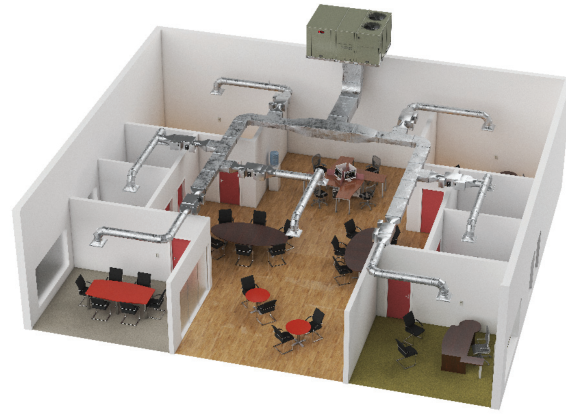


Figure 3. Multiple-zone system



This updated requirement means numerous systems will require active humidity control and cannot simply rely on passive dehumidification to meet the standard's intent. Figure 2 illustrates an example of this. The shaded region shows when the system must be capable of providing dehumidification because the outdoor air dew point temperature is greater than 60°F. The system must provide dehumidification to prevent the indoor dew point temperature from exceeding 60°F.

ASHRAE® Standard 90.1-2019 and -2022 include mandatory requirements which prevent humidistatic controls from lowering the humidity level lower than 55°F dew point, or 60 percent RH in the coldest zone (section 6.4.3.6). There are several exceptions to exempt systems that serve zones with specific humidity needs, such as museums or hospitals. There is also an informative note explaining that lower humidity levels are allowed if they are a result of mechanical cooling for zone temperature control.

The standard also includes prescriptive requirements limiting the use of reheat for humidity control (section 6.5.2.3). There are several exceptions covering a variety of permissible reheat applications:

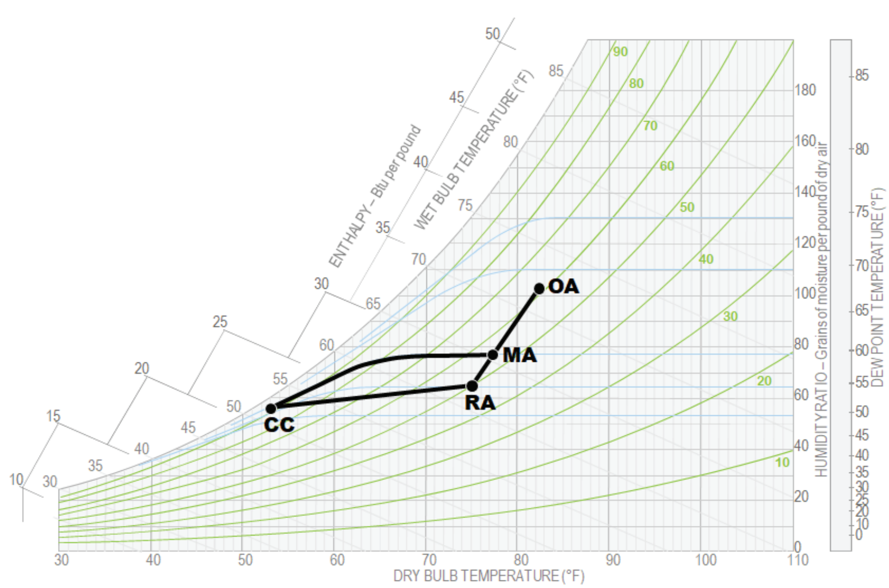
- Exception 1: variable air volume systems which reheat 50 percent or less of the design airflow rate, or the minimum outdoor airflow rate required by Standard 62.1 to meet ventilation requirements.
- Exception 4: unique system and zone types which have specific dehumidification requirements (e.g., museums, surgical suites, pharmacies).
- Exception 5: at least 90 percent of the annual reheat energy is sourced from site-recovered energy, which includes condenser heat (which includes hot gas reheat).

Humidity Control in a Single Zone Direct Expansion System

A single-zone system is designed to maintain comfort in one thermal zone. Common examples of equipment used in a single-zone system include packaged terminal air conditioners, unit ventilators, fan coils, water-source heat pumps, individual variable refrigerant flow terminal units, and packaged rooftop units. The focus of this Engineers Newsletter is packaged rooftops. In the occupied zone, a zone sensor compares the air dry-bulb temperature to a setpoint. The unit controller uses this information to modulate the system's cooling capacity to maintain zone temperature. The supply fan may be constant volume, always providing the same quantity of conditioned air to the zone. Alternatively, a two-speed fan or a variable speed drive may be used to vary the quantity of conditioned air flowing to the zone.

Figure 4 shows cooling operation, where the system mixes outdoor air (OA) for ventilation [(95°F dry bulb temperature (DBT)/78°F wet bulb temperature (WBT)] with recirculated return air (RA, 75°F DBT/50 percent relative humidity). The air mixture (MA) passes through an evaporator (cooling coil), where it is sensibly cooled and dehumidified to 52°F DBT (CC). This cooled air is then supplied to the zone where it absorbs sensible heat and moisture, maintaining the zone dry-bulb temperature at the desired 75°F setpoint with a resulting zone relative humidity of 50 percent (which equates to a 55°F dew point). At full load, zone temperature and humidity are satisfied. At part-load, when the total load is reduced, the system must alter operation to ensure the zone temperature setpoint is maintained.

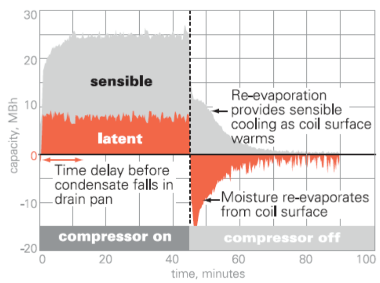
Figure 4. Full-load single-zone, constant volume psychrometric chart



Unit cycling and resulting zone humidity levels

Research has shown how dehumidification performance is affected when direct expansion systems cycle compressor operation as shown in Figure 5. When the compressor starts, the evaporator coil surface becomes cold enough to provide sensible cooling and dehumidification (latent) capacity. After the compressor is shut off, there is a quick reduction of sensible cooling capacity. At the same time, latent capacity goes negative, indicating moisture on the coil surface is re-evaporating into the supply air stream. When the compressor is shut off, the system is adding humidity back into the zone.

Figure 5. Part-load dehumidification (cycling compressor, constant-volume fan)



Shirey, D., H. Henderson, and R. Raustad. 2003. DOE/NETL Project DE-FC26-01NT41253

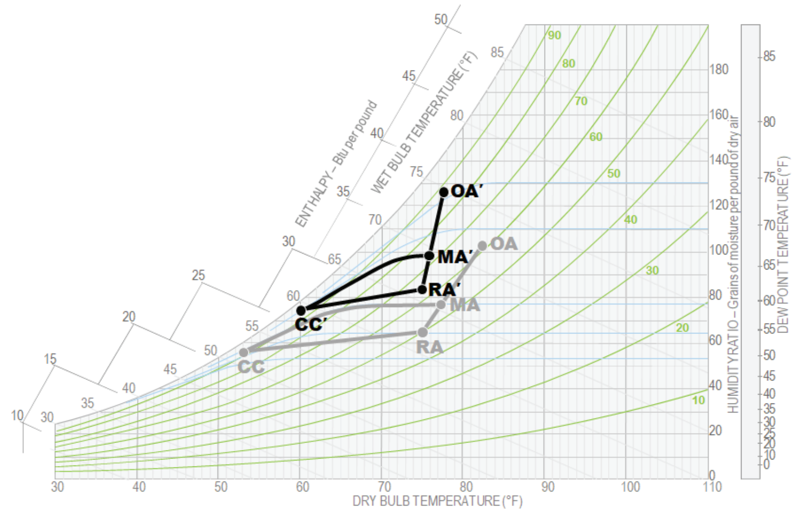
Part-load operation: single zone constant volume systems

The constant volume single-zone system delivers the same amount of air, regardless of any change in zone load. Therefore, during periods of reduced load, the system must deliver warmer supply air to prevent the zone from being overcooled. To accomplish this, a direct-expansion (DX) system cycles the compressor(s) on and off to maintain zone temperature.

When the zone setpoint temperature is met, the compressor(s) cycle off, but the supply fan typically remains on to continue delivering supply air and satisfy ventilation needs. If the outdoor air conditions are mild but moisture-laden, the zone loads will be impacted resulting in rising zone humidity.

Figure 6 shows the same constant volume system operating at part-load. Design dehumidification weather is used in this analysis because it uses a more extreme design dew point temperature compared to cooling design weather. For more information on this weather, see the sidebar titled "Which weather should I use for analysis?"

Figure 6. Part-load single-zone, constant volume psychrometric plot



Which weather should I use for analysis?

Cooling design vs. design dehumidification weather.

ASHRAE® provides design weather for many locations around the world in its Fundamentals handbook. Examples include:

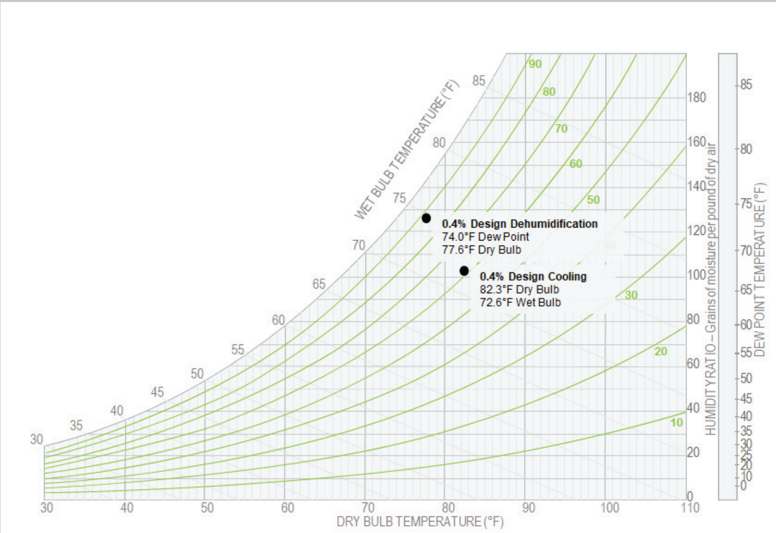
- Heating design conditions
- Humidification design conditions
- Cooling design conditions
- Dehumidification design conditions

For a cooling design analysis, designers can choose from 0.4%, 1%, or 2% cooling dry bulb and mean-coincident wet bulb conditions. The percentage correlates to the number of hours, expressed as a percentage, that exceed the design value. For example, 0.4% design conditions means the actual weather would be expected to exceed the design value 35 hours per year.

For a dehumidification analysis, designers can choose design dehumidification weather, which identifies the conditions when the outdoor dew point temperature is highest. This weather allows the designer to choose from 0.4%, 1%, or 2% dehumidification dew point and mean-coincident dry bulb conditions. Often, design cooling and design dehumidification weather occur at different times and months, which means a design analysis would provide different conduction and solar loads.

To illustrate, the design cooling and design dehumidification conditions have been plotted on a psychrometric chart for Hilo, Hawaii in Figure 7.

Figure 7. Hilo, Hawaii design cooling and design humidification weather conditions



Outdoor air (OA’) mixes with return air (RA’), this mixture (MA’) then passes through the evaporator. At part-load, this air mixture is only cooled to 60°F DBT (CC’), compared to 52°F DBT at full load. While this elevated supply air temperature is sufficient to satisfy the reduced sensible cooling load, the air leaving the unit (CC’) has only been dehumidified to a dew point temperature of 58°F compared to 51°F at design. As a result, the supply air’s ability to offset latent load in the zone is reduced and the resulting zone humidity at part-load is 64 percent compared to 50 percent at full load.

Part-load operation: single zone variable volume systems

The single-zone variable air volume (VAV) system modulates the flow of conditioned air to maintain zone dry-bulb temperature. Therefore, during periods of reduced load, the system reduces the supply fan speed and corresponding airflow rate to deliver enough air to maintain zone temperature.

Systems often have a minimum airflow dictated by the ventilation requirement or a mechanical limitation, such as minimum fan speed. Some units may increase the discharge air temperature setpoint when the minimum fan speed is reached, to prevent the zone from overcooling—see Figure 8 for an example of single-zone VAV fan control.

Compared to the single-zone constant volume system, a single-zone VAV system typically produces lower part-load zone humidity levels because the supply air is dehumidified to a dew point temperature that is lower than the constant-volume system. Figure 9 shows an example of a single-zone VAV system operating at part-load. At part-load, this system maintains relative humidity at 56 percent in the zone, compared to 50 percent at full-load.

Figure 8. Fan speed and discharge air temperature setpoint control for a single-zone VAV system

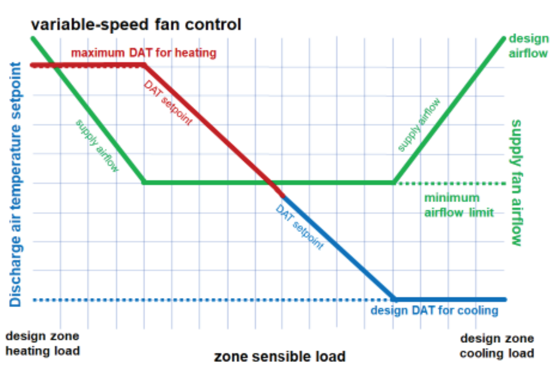
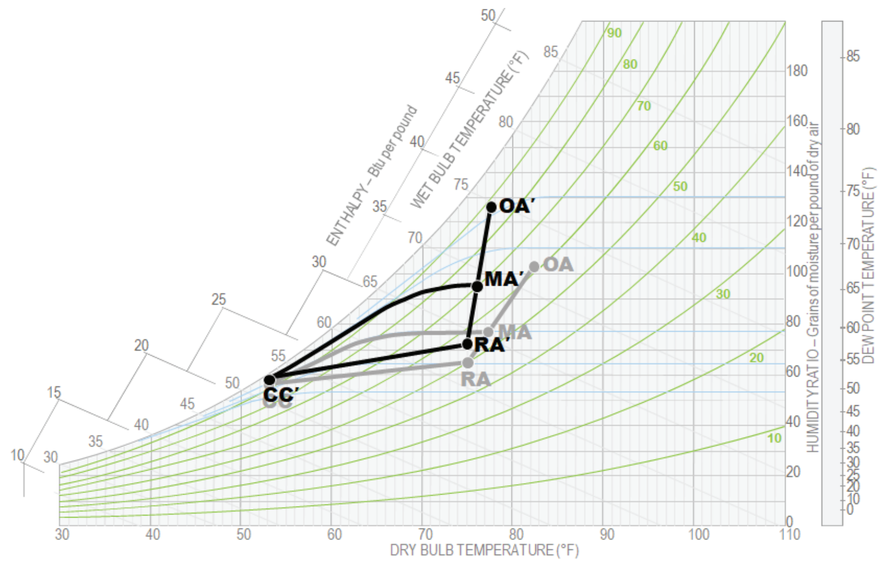


Figure 9. Part-load, single-zone VAV psychrometric plot



Cool and Reheat

The previously discussed system configurations rely upon coincidental dehumidification. When air passes through the cooling coil to be cooled, dehumidification is a by-product of the cooling process. Adding a humidity sensor to the zone and a heat source downstream of the unit cooling coil enables the ability to directly control both humidity and temperature.

When humidity in the zone exceeds a pre-determined limit, the cooling coil capacity is increased to dehumidify the supply air. The downstream heating coil reheats the supply air to prevent overcooling.

During periods of part-load, this method of subcooling ensures the supply air can be sufficiently dehumidified at part-load conditions, when the system needs to deliver warmer discharge air to avoid overcooling the zone. This means the system can provide better part-load dehumidification performance. In Figure 10, a reheat coil has been added to the single-zone, constant volume example. The resulting zone relative humidity is 50 percent with reheat, compared to 64 percent without reheat at the same part-load conditions. As a result, the system with a reheat coil can better maintain zone humidity. Similarly, Figure 11 illustrates a reheat coil added to the single-zone VAV system. Again, the zone relative humidity can be maintained at 50 percent with reheat.

Figure 10. Constant volume single-zone cool-reheat psychrometric plot

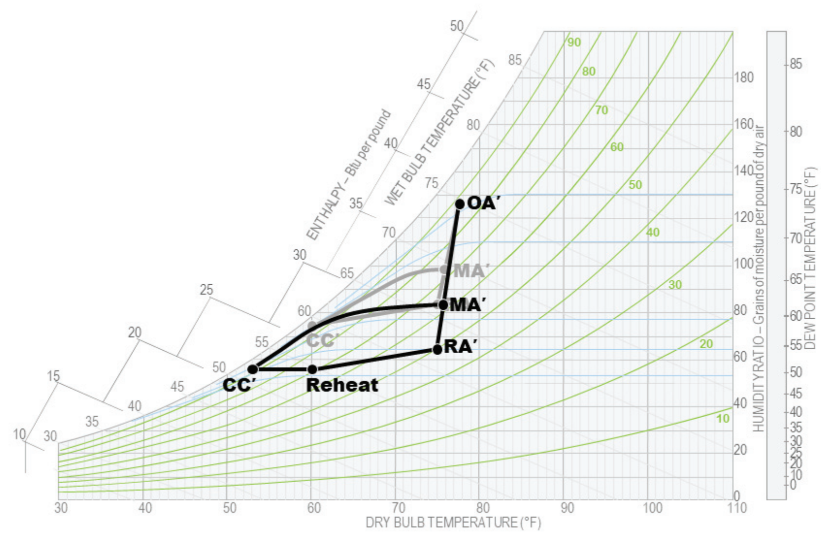


Figure 11. Single-zone VAV cool-reheat psychrometric plot

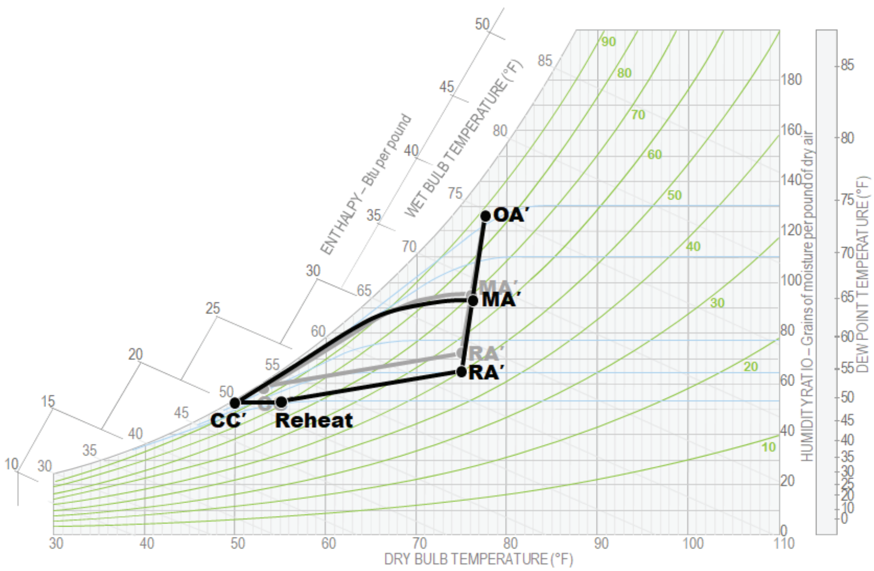


Table 1 shows a summary of the constant- and single-zone VAV psychrometric analyses. The table shows that each system can maintain the zone dry-bulb temperature setpoint—the zone sensor is always satisfied under these load conditions. However, zone relative humidity and dew point temperature changes based upon system type and operation. Adding reheat allows the system to subcool the supply air and maintain better space relative humidity and dew point temperature conditions

Table 1. Comparison of single-zone constant volume and single-zone variable air volume systems at design- and part-load

	Design	Part-load			
	Single-zone constant volume and single-zone VAV	**Single-zone constant volume**	**Single-zone constant volume with hot gas reheat**	**Single-zone VAV**	**Single-zone VAV with hot gas reheat**
Design weather	0.4% Design cooling	0.4% Design dehumidification			
Cooling coil leaving air conditions	53°F DBT	60°F DBT	53°F DBT	53°F DBT	49°F DBT
Resulting zone dry bulb temperature	75°F	75°F	75°F	75°F	75°F
Resulting zone relative humidity	50%	64%	50%	56%	50%
Resulting zone dew point temperature	55°F	62°F	55°F	58°F	55°F
supply fan airflow	5,000 cfm	5,000 cfm	5,000 cfm	3,515 cfm	3,515 cfm
Cooling coil load	17 tons	14 tons	19 tons	15 tons	16 tons
Reheat load			38 MBh		22 MBh

Humidity Control in a Multiple-Zone VAV System

A multiple-zone VAV system is designed to offset the loads in two or more thermal zones. Systems typically consist of a packaged rooftop air conditioner that serves multiple, individually controlled zones. Each zone has a VAV terminal unit that varies the quantity of conditioned air delivered to maintain the desired zone temperature. The rooftop unit is controlled to maintain a discharge air temperature setpoint. The discharge air temperature setpoint is often maintained at or under 55°F to ensure the varying loads of multiple thermal zones can be offset. Systems are typically sized based on ASHRAE® design conditions. Systems may be oversized for most operational hours per year because of design assumptions (climate, varying internal loads), which makes active dehumidification with hot gas reheat a viable option to maintain zone temperature and humidity comfort.

VAV terminal units are equipped with a damper to modulate the flow of cool air from the rooftop unit and may include a source of heat, such as an electric heater or hot water coil. The individual VAV box activates this local source of heat to reheat the cool supply air when needed to prevent the zone from overcooling or to supply warm air to the zone when it needs heat.

At part-load conditions, the system continues to deliver cool air to offset the zone loads. The VAV box modulates its damper to reduce the supply of conditioned air, to ensure the zone temperature is maintained and there is no overcooling. When the VAV box dampers reach their minimum point, reheat will be utilized to ensure the zones are not overcooled.

VAV systems with terminal heat typically do an effective job of maintaining indoor humidity levels for most comfort cooling applications because the system supplies cool, dry supply air at both full- and part-load conditions. Additionally, the following strategies can be utilized to ensure adequate dehumidification:

- **Avoid discharge air temperature reset during humid weather.** This control scheme is designed to reset the rooftop unit's discharge air temperature setpoint upward during mild weather to increase the duration of airside economizing and reduce compressor energy usage. A warmer discharge air temperature setpoint means the system's cooling coil dehumidifies the air less which typically results in higher humidity levels in the zones.

- **Consider equipping VAV terminal units with heat.** The rooftop unit can continue dehumidifying the air to a lower discharge air temperature setpoint and lower dew point temperature, even at part-load conditions when it is humid outside. Heat in the VAV terminal unit can be used to reheat the cool supply air as necessary to avoid overcooling the zone.

Hot Gas Reheat

Heat generated from the vapor compression refrigeration cycle can be recovered from direct expansion equipment rather than rejecting it to the ambient environment. This is typically accomplished by placing a hot gas reheat coil downstream of the unit evaporator (Figure 12). Hot refrigerant from the compressor is routed through a valve to the hot gas reheat coil to reheat the airstream. Heat is transferred from the hot refrigerant vapor to the cooled and dehumidified supply air. The cooled refrigerant vapor continues its journey to the unit's condenser coil for heat rejection.

Hot gas reheat allows the system to dehumidify the supply air to a sufficient dew point temperature for dehumidification, then reheat the air without the use of additional new energy, like fossil fuel heat or electric heat. The usage of new energy may be prohibited by standards or codes.

Staged hot gas reheat

Units equipped with staged hot gas reheat cycle the flow of hot refrigerant to the hot gas reheat coil. Generally, all or most of the hot refrigerant gas is sent to the hot gas reheat coil to provide reheat. As a result, the unit discharge air temperature fluctuates. Some unit controls will monitor zone temperature during hot gas reheat operation. When the zone dry-bulb temperature exceeds a pre-determined limit, such as 75°F, hot gas reheat will be stopped and the unit will resume normal cooling operation to ensure there is no overheating.

While this typically provides sufficient humidity and temperature control, cycling between modes can result in temperature and humidity fluctuations.

Modulating hot gas reheat

Modulating hot gas reheat uses a three-way valve to modulate the flow of hot refrigerant to the hot gas reheat coil. This allows the unit to maintain a more-consistent discharge air temperature at a pre-determined setpoint by regulating the flow of hot refrigerant from the compressor.

Hot gas reheat in a single zone system

In a single zone system, when dehumidification is needed, the evaporator (cooling coil) dehumidifies the supply air to a sufficiently low dew point temperature and the hot gas reheat coil tempers the cold, dry supply air to prevent overcooling the zone.

If equipped with modulating hot gas reheat, the system will dehumidify the air leaving the evaporator coil then reheat to a calculated discharge air temperature setpoint. This setpoint can be reset periodically to match the zone load requirements.

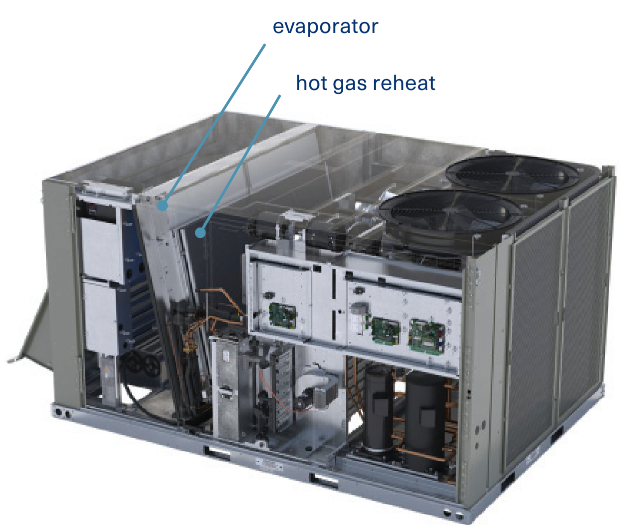
Hot gas reheat in a multiple-zone system

In a multiple-zone VAV system, hot gas reheat may not be needed because the rooftop unit is typically dehumidifying the air to a low dew point temperature at both full- and part-load conditions. Typically, there are several methods that can be used to address high zone humidity:

- **Increase the flow of dry conditioned air to the zone.** Temporarily increase the supply of dry supply air by forcing the VAV damper open. This will increase the flow of dry conditioned air from the rooftop unit. As a result, the rooftop fan will increase it's operating speed. To prevent overcooling the zone, that VAV terminal unit may activate its reheat coil.
- **For rooftops without hot gas reheat: reset rooftop discharge air temperature setpoint downward.** Controls can temporarily reset the rooftop unit's discharge air temperature setpoint downward to increase supply air dehumidification by delivering air at a lower dew point temperature. The entire quantity of supply air will be dehumidified, resulting in a colder supply air dry bulb temperature. As a result, VAV boxes may close their dampers to restrict airflow and maintain zone temperature. This will result in more compressor energy to dehumidify the entire quantity of supply air. Additional reheat energy may be used to prevent overcooling.
- **For rooftops with hot gas reheat: reset the rooftop evaporator leaving air temperature setpoint downward.** Controls can temporarily reset the rooftop unit's leaving evaporator air temperature setpoint downward to increase supply air dehumidification. Hot gas reheat is then used to reheat the air to the standard discharge air temperature setpoint. This will result in more compressor energy to dehumidify the entire quantity of supply air and little or no impact on fan or zone-level reheat energy.

Some multiple-zone applications may benefit from hot gas reheat to improve dehumidification performance. If the humidity level in a zone were ever to rise higher than desired, the unit controller can lower the air temperature leaving the evaporator to increase dehumidification. This allows the evaporator to remove more moisture from the air. Modulating hot gas reheat is then used to temper (reheat) the air back up to the desired discharge air temperature setpoint. For example, during cooling mode, the unit may discharge air at 55°F. During dehumidification mode, the leaving-evaporator temperature setpoint may be reset downward from 55°F to 50°F. Modulating hot gas reheat then reheats the cool, dehumidified air from 50°F to 55°F. VAV terminal unit heaters may then be used to provide additional zone reheat to maintain temperature.

Figure 12. Example packaged rooftop unit with hot gas reheat



Load and Energy Modeling

Designers have access to robust tools that can be used to model buildings and their HVAC systems. Many of these tools can simulate the usage of dehumidification and hot gas reheat and provide component load sizing and estimated energy costs. The tools can be used to evaluate buildings and systems using cooling design and design dehumidification weather (see "Which weather should I use for analysis?" sidebar). Additionally, a full 8,760-hour analysis can help predict zone humidity levels.

Designers should use care when simulating the system to ensure the equipment is properly sized. Often, the usage of hot gas reheat will cost more in annual energy, but the benefits often outweigh the additional energy cost due to the ability to actively control humidity, maintain zone comfort, and reduce the potential of moisture-related damage.

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Conclusion

System dehumidification capabilities will continue to play an increasingly important role in air conditioning systems. The latest versions of Standard 62.1 now require active control of indoor humidity in a variety of climates when it is humid outside. Modulating hot gas reheat can be effectively used to maintain zone temperature control during dehumidification to maintain the interior zone dew point temperature below the desired limit.

By Eric Sturm and Greg DuChane, Trane. To subscribe or view previous issues of the Engineers Newsletter visit trane.com. Send comments to ENL@trane.com.

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