Dehumidification Performance Of HVAC Systems

By John Murphy, Member ASHRAE

Microbial contamination is a common cause of occupant complaints and indoor air quality (IAQ) problems in buildings. ANSI/ASHRAE Standard 62-2001, *Ventilation for Acceptable Indoor Air Quality* and the United States Environmental Protection Agency (EPA) both recommend that indoor relative humidity be maintained below 60% to minimize the risks of microbial growth.

Historically, mechanical HVAC systems have focused on controlling the dry-bulb temperature within an occupied space. Space humidity has not been actively controlled and has often been described as coincidental.

This article uses basic psychrometric analyses to discuss the dehumidification performance of various cold-coil HVAC systems in non-residential comfort-cooling applications—particularly at part-load conditions. The dehumidification performance of a system hinges on its ability to reduce the temperature of the air passing through the cooling coil below the dew point of the air. Ironically, the widely used single-zone, constant-volume system can also be the most problematic when it comes to dehumidification at part load.

Constant-Volume Systems

The basic constant-volume (CV) system consists of an air handler (containing a fan and coil) that supplies a constant volume of air to a single thermal zone. A thermostat compares the zone dry-bulb temperature to the setpoint and modulates the capacity of the cooling coil, adjusting the supply-air temperature until the zone temperature matches the setpoint. This type of system *indirectly* (or coincidentally) controls space humidity. Water vapor condenses on the coil whenever its surface temperature is lower than the dew point of the air passing through it. Less cooling capacity, and therefore a warmer coil surface, means less dehumidification.

The peak sensible load on the cooling coil does not typically occur at the same time as the peak latent load. Cooling coils that are *controlled* to maintain the dry-bulb temperature in the zone often operate without adequate latent capacity at peak latent load conditions. For a complete understanding of a system's dehumidification performance, the system must be analyzed at both full- and part-load conditions.

Since it was added to the *ASHRAE Handbook—Fundamentals*, many designers use the peak dew point condition to analyze the part-load dehumidification performance of a system. However, do *not* assume that this peak dew point represents the worst-case condition for space humidity control. Space humidity depends as much on space sensible load, space sensible heat ratio (SHR), and the way the HVAC system is controlled, as it does on the condition of the outdoor air.

To demonstrate, consider a 10,000 ft^3 (283 m^3) classroom in Jacksonville, Fla., that accommodates 30 people. The basic CV system serving this classroom contains a chilled-water cooling coil with a modulating control valve for capacity control. For thermal comfort, the space setpoint is 74°F (23.3°C) dry bulb. Supply airflow is 1,500 cfm (0.7 m^3/s), which equates to nine air changes per hour. To provide adequate ventilation, Standard 62-2001requires 15 cfm (8 L/s) of outdoor air for each person, or 450 cfm (0.2 m^3/s) for this space.

About the Author

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Location	Peak Dew-Point Condition	Resulting Space RH	Cool, Rainy Day	Resulting Space RH
Baltimore	75°F DP, 83°F DB (23.8°C DP, 28.1°C DB)	62%	70°F DB, 69°F WB (21.2°C DB, 20.6°C WB)	65%
Dallas	75°F DP, 82°F DB (23.7°C DP, 28.0°C DB)	66%	70°F DB, 69°F WB (21.2°C DB, 20.6°C WB)	68%
Denver	60°F DP, 69°F DB (15.6°C DP, 20.4°C DB)	55%	63°F DB, 61°F WB (17.2°C DB, 16.1°C WB)	58%
Jacksonville, Fla.	76°F DP, 84°F DB (24.6°C DP, 28.8°C DB)	67%	70°F DB, 69°F WB (21.2°C DB, 20.6°C WB)	73%
Los Angeles	67°F DP, 75°F DB (19.4°C DP, 23.6°C DB)	62%	63°F DB, 62°F WB (17.2°C DB, 16.7°C WB)	65%
Minneapolis	73°F DP, 83°F DB (22.5°C DP, 28.5°C DB)	66%	70°F DB, 69°F WB (21.2°C DB, 20.6°C WB)	70%
San Francisco	59°F DP, 76°F DB (15.2°C DP, 19.4°C DB)	56%	54°F DB, 53°F WB (12.2°C DB, 11.7°C WB)	56%

Table 1: Basic CV system performance for various cities.

System	Resulting Space RH at Peak Dry-Bulb Condition	Resulting Space RH at Peak Dew-Point Condition	Resulting Space RH on Cool, Rainy Day
Basic, Constant-Volume System	52%	67%	73%
With Fan-Speed Adjustment	52%	60%	68%
With Mixed-Air Bypass	52%	65%	68%
With Mixed-Air Bypass and Fan-Speed Adjustment	52%	58%	65%
With Return-Air Bypass (Full Coil Face at Part Load)	52%	55%	60%
With Return-Air Bypass (Reduced Coil Face at Part Load)	52%	64%	66%

Table 2: Coincidental dehumidification performance for various enhancements to CV systems.

Performance at Peak Dry-Bulb Condition

The peak outdoor dry-bulb condition for Jacksonville is 96°F dry bulb, with an average coincident wet bulb of 76°F (35.7°C DB, 24.5°C WB). At this condition, the sensible and latent loads calculated for the space—29,750 Btu/h (8.7 kW) and 5,250 Btu/h (1.5 kW), respectively—yield a space sensible heat ratio (SHR) of 0.85. These are space loads only; the load due to the introduction of outdoor air for ventilation is intended to be offset by the cooling coil. Also, only the latent (moisture) load due to occupants is considered in this example. For simplicity, other sources of indoor moisture, such as infiltration and vapor pressure diffusion, are neglected. If included, these additional moisture sources would result in even higher space humidity levels. Reference 4 includes more detail on indoor sources of moisture.

Given the supply airflow of 1,500 cfm (0.7 m^3/s), the system must deliver air at 55.7°F (13.1°C) to offset the sensible load in the space and maintain setpoint.

$$Q_s = 1.085 \times 1{,}500\ \text{cfm} \times (74°\text{F} - T_{sa})$$
$$= 29{,}750\ \text{Btu/h} \therefore T_{sa} = 55.7°\text{F}$$
$$\left[\begin{array}{c} Q_s = 1.21 \times 0.7\ \text{m}^3/\text{s} \times (23.3°\text{C} - T_{sa}) \\ = 8.7\ \text{kW} \therefore T_{sa} = 13.1°\text{C} \end{array}\right]$$

At this condition, the resulting space relative humidity is 52% (*Figure 1*). The cooling coil removes both sensible heat and moisture, directly controlling space temperature and indirectly reducing space humidity.

Performance at Peak Dew-Point Condition

As the space sensible load drops, however, this system allows the supply-air temperature to rise by reducing the capacity of the cooling coil. Although this control action successfully maintains the space dry-bulb temperature, it also reduces the amount of moisture that condenses on the coil, and space humidity rises.

The peak outdoor dew-point condition for Jacksonville is 76°F dew point, with an average coincident dry bulb of 84°F (24.6°C DP, 28.8°C DB). At this condition, the sensible load in the classroom drops to 17,850 Btu/h (5.2 kW) as a result of a lower outdoor dry-bulb temperature and the lower solar and conducted heat gains. The latent load due to occupants remains unchanged (5,250 Btu/h [1.5 kW]), however, and the space SHR drops to 0.77. Due to the lower space sensible load, the 1,500 cfm (0.7 m^3/s) of supply air must be delivered at a warmer temperature—63°F (17.2°C)—to prevent overcooling the space.

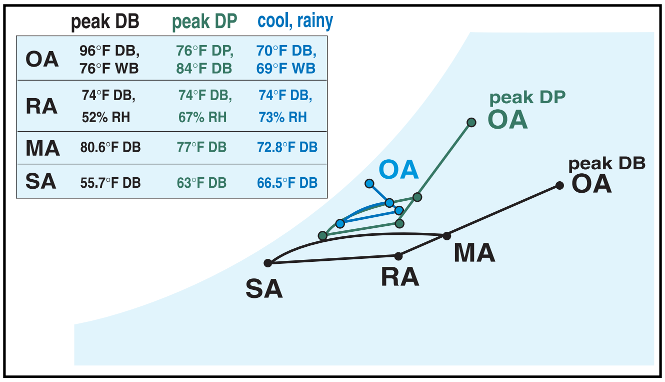

Figure 1: Basic constant-volume system.

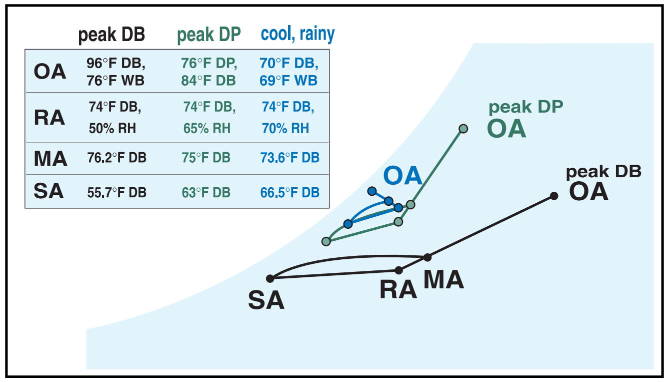

Figure 2: Basic CV system (underventilated).

$$Q_s = 1.085 \times 1{,}500 \text{ cfm} \times \left(74°\text{F} - T_{sa}\right)$$
$$= 17{,}850 \text{ Btu/h} \therefore T_{sa} = 63°\text{F}$$
$$\left[\begin{array}{l} Q_s = 1.21 \times 0.7 \text{ m}^3/\text{s} \times \left(23.3°\text{C} - T_{sa}\right) \\ \quad = 5.2 \text{ kW} \therefore T_{sa} = 17.2°\text{C} \end{array}\right]$$

This warmer air, combined with a lower space SHR, raises the relative humidity in the classroom from 52% to 67%—well above the 60% limit recommended by ASHRAE.

This is not simply a coil sizing issue. Whenever a partial-sensible-load condition exists, the thermostat reduces the capacity of the cooling coil. Less moisture is removed from the air and space humidity rises. Oversizing the cooling coil will not prevent this shortfall in latent capacity if system control is based solely on sensible conditions (space dry-bulb temperature).

Performance on a Cool, Rainy Day

Finally, we will consider a cool, rainy day—70°F dry bulb, 69°F wet bulb (21.2°C DB, 20.6°C WB). At this condition, the sensible load in the classroom drops even further to 12,250 Btu/h (3.6 kW). The latent load again remains unchanged, so the space SHR drops to 0.70. To avoid overcooling the space, the supply-air temperature must be 66.5°F (19.2°C).

$$Q_s = 1.085 \times 1{,}500 \text{ cfm} \times \left(74°\text{F} - T_{sa}\right)$$
$$= 12{,}250 \text{ Btu/h} \therefore T_{sa} = 66.5°\text{F}$$
$$\left[\begin{array}{l} Q_s = 1.21 \times 0.7 \text{ m}^3/\text{s} \times \left(23.3°\text{C} - T_{sa}\right) \\ \quad = 3.6 \text{ kW} \therefore T_{sa} = 19.1°\text{C} \end{array}\right]$$

The result is that the relative humidity in the classroom rises to 73%. Again, space humidity can depend as much on space sensible load, space SHR, and control of the HVAC system, as it does on outdoor conditions.

Impact of Outdoor Air Quantity

Some believe that indoor humidity problems result primarily from the deliberate introduction of humid outdoor air for ventilation. However, consider what happens if the outdoor airflow for this example classroom is reduced to 150 cfm (0.07 m^3/s), or 5 cfm/person (2.67 L/s/person).

Because the *space* sensible and latent loads are unchanged (only the ventilation load changes), the supply-air temperature and space SHR are also unchanged. At the peak dry-bulb condition, the resulting space relative humidity is 50% (*Figure 2*), as compared to 52% with the proper quantity of ventilation air. But, at the peak dew point condition, the resulting space humidity is nearly 65%, and on the cool, rainy day, it is 70%.

Reducing the ventilation rate lowers space humidity slightly, but may *not* adequately solve the problem of high space humidity levels associated with CV systems that are controlled based on space dry-bulb temperature alone. More importantly, it results in underventilated spaces, possibly leading to other IAQ problems.

The use of traditional packaged, direct-expansion (DX) air-conditioning equipment can compound the indoor humidity problem in CV systems with higher ventilation rates. More outdoor air, especially in humid climates, increases the required cooling and dehumidification capacity. Because this type of equipment has a limited cfm/ton range, this increase in capacity often results in higher supply airflow, corresponding warmer supply-air temperatures, and elevated space humidity levels. The cycling of compressors in DX equipment complicates the problem because condensate re-evaporates from the coils when the compressors are off, but the fans remain on.

Impact of Climate

Contrary to popular belief, high indoor humidity levels can be an issue in nearly all geographic locations, not just in areas where hot, humid conditions prevail. Whenever high relative humidity levels exist at or near a cold, porous surface, moisture adsorption increases and moisture-related problems (such as increased health risks from mold growth and premature replacement of equipment and furnishings) become likely.

Table 1 compares the dehumidification performance of this basic, CV system serving this example classroom in various climates. Notice how similar the peak dew point condition is for many of the locations. In these regions, the part-load performance of this example system is similar. In the dry climates

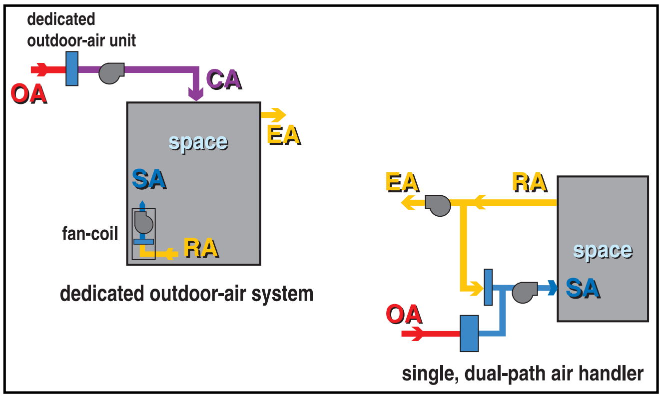

Figure 3: CV systems with separate path for OA treatment.

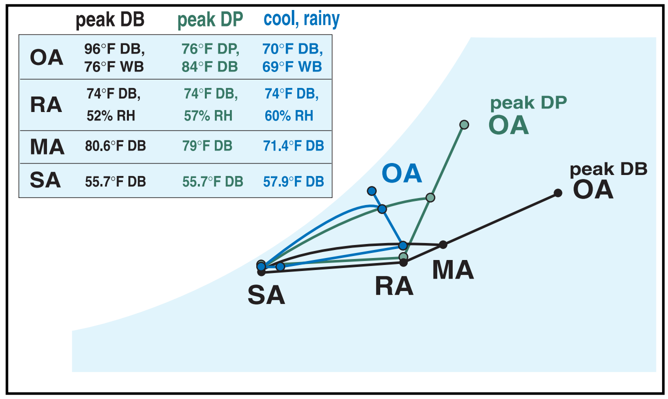

Figure 4: Basic VAV system.

(Denver and San Francisco), the system performs better because the outdoor air is dry enough to provide a dehumidifying effect. Ignoring system operation at part-load conditions can lead to high indoor humidity levels in many locations, not just hot, humid climates.

Enhancements to Constant-Volume Systems

There are ways to improve the dehumidification performance of a constant-volume system. Some enhancements directly control space humidity while others improve the system's ability to coincidentally dehumidify the air.

Supply-Air Tempering (Reheat)

The most common method used for directly controlling indoor humidity in a CV system is to overcool the air to remove moisture, and then temper (reheat) the air to avoid overcooling the space. A humidity sensor in the space controls the capacity of the cooling coil to remove moisture from the supply air and maintain space humidity below an upper limit (typically the ASHRAE-recommended limit of 60% RH).

The downstream heating coil raises the dry-bulb temperature of the supply air just enough to avoid overcooling the space. However, as long as space humidity is below the upper limit, the system performs just like the basic CV system described earlier.

Supply-air tempering may use "new" energy or heat recovered from some other part of the system. Does ANSI/ASHRAE/IESNA Standard 90.1–2001, *Energy Standard for Buildings Except Low-Rise Residential Buildings* prohibit the use of new energy for reheat in CV systems? Not necessarily.

Section 6.3.2 of the standard does not prohibit the use of new energy reheat, it only limits its use by defining the exceptions where it is allowed. Smaller terminal equipment, mid-size equipment that is capable of unloading to 50% capacity before reheat is used, and systems that serve certain space types (such as museums, surgical suites, and supermarkets) are exempt from this limitation. Additionally, reheat is always allowed if at least 75% of the reheat energy is recovered.

Treat the Outdoor Air Separately

Another method of directly controlling indoor humidity is to individually treat the outdoor and return airstreams. Separate cooling coils are controlled independently to maintain both space temperature and humidity. A space humidity sensor directly controls the capacity of the outdoor-air coil to maintain space humidity below the upper limit. A space thermostat directly controls the capacity of the return-air coil to maintain space dry-bulb temperature at setpoint.

This can be accomplished using an entirely separate air handler (a dedicated outdoor-air unit) to dehumidify all of the outdoor air, to a dew point drier than the space, before delivering it directly to the occupied spaces, or to the mixing boxes of other air handlers. *Figure 3* shows a dedicated outdoor-air unit delivering conditioned outdoor air directly to an occupied space where a fan-coil handles the space load.

Alternatively, a single "dual-path" air handler can be used to separately condition *both* airstreams in the same unit (*Figure 3*). Each airstream has its own cooling coil, but a single constant-volume fan serves both paths. A stacked configuration is often used to take advantage of the smaller footprint.

In addition to these direct humidity control enhancements, there are other enhancements that simply improve the indirect (or coincidental) dehumidification performance of a CV system.

Fan Speed Adjustment

Many in-space terminal units, such as fan-coils and classroom unit ventilators, have the ability to operate at multiple fan speeds. Automatically reducing the fan speed as the first step of cooling capacity reduction improves the dehumidification performance of these CV units. The reduced airflow results in a lower supply-air temperature for a given load condition, and therefore, more moisture is removed from the air.

Face-and-Bypass Dampers

Face-and-bypass dampers arranged to allow air to bypass the cooling coil can also improve the indirect dehumidification performance of a CV system. A space thermostat controls cooling capacity by adjusting the positions of the linked face and bypass dampers, regulating airflow through and around the coil until the appropriate supply-air temperature is achieved. Chilled-water flow through the cooling coil is held constant, not modulated. While the entering water temperature and flow rate are unchanged, the velocity of the air passing through the coil drops at part load, allowing the air to get colder and more moisture to condense. Resetting the temperature of the chilled water, or varying water flow through the cooling coil, both negatively impact the performance of this system enhancement, and should be avoided.

There are two configurations for using face-and-bypass dampers: mixed-air bypass and return-air bypass. Mixed-air bypass blends cool, dry air leaving the cooling coil with mixed air (a mixture of outdoor and return air). Return-air bypass blends cool, dry air leaving the cooling coil with return air. When the outdoor air contains more moisture than the return air, return-air bypass is more effective because it usually directs *all* of the moist outdoor air through the cooling coil.

Because of limited space, the implementation of return-air bypass in terminal units often results in reduced coil face area as the damper closes. In other words, as the load decreases, the face damper prevents mixed air from passing through part of the cooling coil. The effect is that the air passing through the coil does not slow down much at part load. This results in warmer air leaving the coil, and higher space humidity, than if the entire face of the coil was available.

While direct dehumidification enhancements (supply-air tempering and treating the outdoor air separately) can be used to control space humidity to any desired limit, the indirect enhancements simply improve the indirect (or coincidental) dehumidification performance of the CV system. *Table 2* compares the performance of these indirect enhancements for our classroom example.

VAV Systems

A variable-air-volume (VAV) system consists of a central air handler that supplies constant-temperature air to multiple thermal zones. A thermostat in each zone compares dry-bulb temperature to the setpoint, and a VAV terminal unit modulates the volume of air delivered to the zone in response to the changing sensible load. The central supply fan is modulated to maintain static pressure in the duct system and the capacity of the central cooling coil is modulated to maintain a constant supply-air temperature.

VAV systems generally provide effective, *indirect* (or coincidental) dehumidification over a wide range of indoor load conditions. As long as any space needs cooling, the VAV air handler supplies dry (low dew point) air to all of the VAV terminal units. Let's use the same example classroom to analyze the dehumidification performance of this basic VAV system.

Performance at Peak Dry-Bulb Condition

At the peak dry-bulb condition, the space sensible load and supply-air temperature are the same as for the CV system. Given the supply airflow of 1,500 cfm (0.7 m^3/s), a supply-air temperature of 55.7°F (13.1°C) is required to offset the space sensible cooling load. The resulting space relative humidity is 52% (*Figure 4*).

Performance at Peak Dew-Point Condition

At partial sensible-load conditions, the VAV system responds by reducing the quantity of air supplied to the space, while maintaining a constant supply-air temperature. At the part-load, peak dew point condition, the supply airflow is reduced to 899 cfm (0.42 m^3/s) to avoid overcooling the space.

$$Q_s = 1.085 \times V_{sa} \times (74°\text{F} - 55.7°\text{F})$$
$$= 17{,}850 \text{ Btu/h} \therefore V_{sa} = 899 \text{ cfm}$$
$$\begin{bmatrix} Q_s = 1.21 \times V_{sa} \times (23.3°\text{C} - 13.1°\text{C}) \\ = 5.2 \text{ kW} \therefore V_{sa} = 0.42 \text{ m}^3/\text{s} \end{bmatrix}$$

Because the supply air is still cool and dry, the relative humidity in the classroom only rises to 57%, as compared to 67% for the basic CV system operating at this same condition.

Impact of Minimum Airflow Settings

Eventually, the sensible load in the space drops to a point where the required airflow is below the minimum airflow setting of the VAV terminal unit. The minimum airflow setting for this example classroom is 700 cfm (0.33 m^3/s).

On the cool, rainy day, if 700 cfm (0.33 m^3/s) is supplied at 55.7°F (13.1°C), the space will be overcooled to 71.8°F (22.1°C). As the dry-bulb temperature in the space decreases, the relative humidity increases—to 66% in this example—and the space feels cool and damp.

$$Q_s = 1.085 \times 700 \text{ cfm} \times (T_{sp} - 55.7°\text{F})$$
$$= 12{,}250 \text{ Btu/h} \therefore T_{sp} = 71.8°\text{F}$$
$$\begin{bmatrix} Q_s = 1.21 \times 0.33 \text{ m}^3/\text{s} \times (T_{sp} - 13.1°\text{C}) \\ = 3.6 \text{ kW} \therefore T_{sp} = 22.1°\text{C} \end{bmatrix}$$

One solution to prevent overcooling is to lower the minimum airflow setting of the VAV box. However, this setting is likely based on either space ventilation requirements, or diffuser or terminal unit performance limitations.

Another possible solution to prevent overcooling is to reset the temperature of the supply air upward at low-load conditions. On the cool, rainy day, raising the supply-air temperature to 57.9°F (14.3°C) would avoid overcooling the space and reduce the energy consumed by the mechanical cooling equipment.

$$Q_s = 1.085 \times 700\,\text{cfm} \times (74°\text{F} - T_{sa})$$
$$= 12{,}250\,\text{Btu/h} \therefore T_{sa} = 57.9°\text{F}$$
$$\left[\begin{array}{l} Q_s = 1.21 \times 0.33\,\text{m}^3/\text{s} \times (23.3°\text{C} - T_{sa}) \\ \quad = 3.6\,\text{kW} \therefore T_{sa} = 14.3°\text{C} \end{array}\right]$$

However, less moisture condenses out of the air and the space relative humidity rises to 65%. Each system must be analyzed to determine if the increase in space humidity levels, and fan energy consumption, outweigh the savings in mechanical cooling and reheat energy.

Adding sensible heat at the VAV terminal unit to temper (reheat) the supply air is the most common method of avoiding both overcooling the space and rising space humidity levels. When the supply airflow drops to the minimum setting, sensible heat is added either at the terminal unit or within the space itself. This might involve radiant heat in the space, a heating coil mounted on the VAV terminal unit, fan-powered VAV units, or a dual-duct VAV system.

On the cool, rainy day, a heating coil in the VAV terminal unit is used to warm the 55.7°F (13.1°C) supply air to 57.9°F (14.3°C) before delivering it to the space. This avoids overcooling the space and results in a space relative humidity of 60% (*Figure 4*).

Supply-air tempering at the VAV terminals may use "new" energy or heat recovered from some other part of the system. Does Standard 90.1–2001 prohibit the use of new energy for reheat in VAV terminals? The answer is generally no. Section 6.3.2 of the standard does not prohibit the use of new energy reheat, it only limits its use by defining the exceptions where it is allowed. Most zones in a VAV system have a minimum airflow setting below 50% of design supply airflow. Therefore, due to Exception A in this section, new energy would be allowed for reheat after the airflow is reduced to the minimum setting.

Enhancements to VAV Systems

Even though VAV systems generally provide effective, *indirect* dehumidification over a wide range of indoor load conditions, there are ways to improve their dehumidification performance.

Treat the Outdoor Air Separately

One method is to separately treat the outdoor and return airstreams. This is typically accomplished using a dedicated outdoor-air unit to cool and dehumidify all of the outdoor air to a dew point drier than the space. This conditioned outdoor air is then delivered directly to the spaces, to the "ventilation damper" of individual dual-duct VAV terminal units, or to one or more VAV air handlers. A humidity sensor in the space controls the capacity of the dedicated outdoor-air unit to maintain humidity in all spaces below an upper limit.

Colder Supply Air

Lowering the temperature of the air leaving the central cooling coil in a VAV system results in more moisture being condensed out of the supply air. At the peak dry-bulb condition, designing the VAV system serving this example classroom for a 50°F (10°C) supply-air temperature, rather than 55.7°F (13.3°C), results in lower supply airflow.

$$Q_s = 1.085 \times V_{sa} \times (74°\text{F} - 50°\text{F})$$
$$= 29{,}750\,\text{Btu/h} \therefore V_{sa} = 1{,}142\,\text{cfm}$$
$$\left[\begin{array}{l} Q_s = 1.21 \times V_{sa} \times (23.3°\text{C} - 10°\text{C}) \\ \quad = 8.7\,\text{kW} \therefore V_{sa} = 0.54\,\text{m}^3/\text{s} \end{array}\right]$$

This colder, drier supply air results in a drier space at all load conditions. For example, at the peak dry-bulb condition, the space relative humidity is 47%, compared to 52% with a more-traditional supply-air temperature.

Summary

HVAC systems have historically focused on controlling the space dry-bulb temperature, while space dehumidification was coincidental. The widely used single-zone, constant-volume system can be the most problematic when it comes to dehumidification, particularly at part-load conditions. VAV systems, however, generally provide effective, indirect dehumidification over a wide range of indoor load conditions.

When properly designed and controlled, the HVAC system can significantly reduce the moisture content of indoor air. Analyze system dehumidification performance at both full- and part-load conditions, and consider the advantages and disadvantages of each system enhancement. The enhancements discussed in this article are detailed further in *Reference 3*. The right choice for a given project depends on the climate, building use, available budget, and operating cost goals.

Bibliography

1. ANSI/ASHRAE Standard 62.1–2001, *Ventilation for Acceptable Indoor Air Quality.*
2. ANSI/ASHRAE/IESNA Standard 90.1–2001, *Energy Standard for Buildings Except Low-Rise Residential Buildings.*
3. Murphy, J. 2002. *Dehumidification in HVAC Systems*, applications engineering manual, *SYS-APM004-EN*, Trane.
4. Stanke, D., et al. 1998. *Managing Building Moisture*, applications engineering manual, *SYS-AM-15*, Trane.
5. Stanke, D. and B. Bradley. 2000. "Dehumidify with constant-volume systems." *Engineers Newsletter*, 29(4) Trane.
6. Stanke, D. and B. Bradley. 2001. "Dedicated ventilation systems." *Engineers Newsletter*, 30(3) Trane.
7. U.S. Environmental Protection Agency. 2001. *Mold Remediation in Schools and Commercial Buildings*, EPA 402-K-01-001, March (www.epa.gov/iaq/molds).