engineers newsletter

volume 33–2 ● providing insights for today's hvac system designer

Better Part-Load Dehumidification

[it's *not* a pipe dream]

from the editor …
While "humidity control" is apt to imply special applications, such as museums or printing plants, managing humidity should *be a key design consideration in* any *HVAC application. This EN outlines the challenge of dehumidifying at part load and describes ways to enhance the performance of three commonly used types of HVAC systems: chilled water terminal, single-zone direct expansion (DX), and central VAV.*

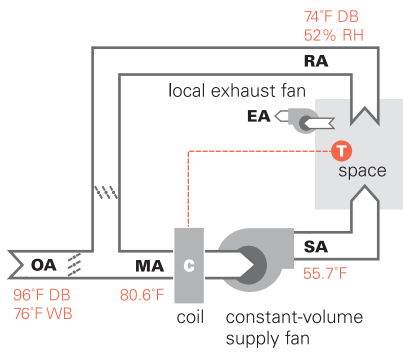
Why it's hard to manage humidity at part load

The most basic HVAC system supplies a constant quantity of air to a single zone; common examples include packaged terminal air conditioners, small packaged rooftop units, unit ventilators, and fan-coils. A thermostat compares the dry-bulb temperature in the occupied space to a setpoint and then modulates the system's cooling capacity accordingly (Figure 1).

At full load, the system mixes outdoor air with recirculated return air. The mixture then passes through a cooling coil, where it's sensibly cooled and dehumidified. When the supply air reaches the space, it absorbs sensible heat and moisture—lowering the dry-bulb temperature to the target (74°F in Figure 1) and maintaining the relative humidity (in this case, at 52% RH). This full-load dehumidification performance certainly *seems* adequate, but what happens at part load?

The basic constant-volume system supplies an unchanging amount of air, regardless of the cooling load. The system therefore must deliver *warmer* supply air at part load to avoid overcooling the space.

Figure 1. Single-zone, constant-volume, mixed-air system



Chilled water vs. DX coils. In a typical chilled water application, a modulating valve reduces system capacity by throttling the rate of water flow through the cooling coil. Figure 2 (p. 2) shows how this affects the supply air leaving the coil—the warmer coil surface that results from less water flow provides less sensible cooling (raising the supply-air temperature) and removes less moisture from the passing air stream.

In DX applications, the compressors cycle off regularly to avoid overcooling. Recent research clearly shows how compressor cycling affects part-load dehumidification (Figure 3, p. 2). When the compressor starts, the coil surface quickly becomes cold enough to provide both sensible cooling (gray region) and *latent cooling* (also known as "dehumidification"; red region).

Notice that several minutes elapse after the compressor starts but before the moisture that condenses on the

Oversized airflow and dehumidification don't mix

Determining the proper relationship between airflow and cooling capacity is key to designing a system with adequate dehumidification capacity at all load conditions. The flexibility of applied systems, such as chilled water air handlers, normally lets you select equipment based on a specific airflow rate (cfm) *and* a specific cooling capacity (tons). By contrast, the prematched components of packaged DX equipment typically limit your selection to a finite cfm/ton range of application.

Selecting a larger unit to provide the required cooling capacity sometimes yields a higher-than-necessary supply airflow. The result is warmer supply air and, in non-arid climates, less dehumidification.

Higher-than-necessary airflow also can result if you base the design airflow on a rule-of-thumb rather than a calculation of the space sensible load. The old standbys of 400 cfm/ton, 1 cfm/ft², and 6 air changes/hr were formulated before the advent of higher ventilation rates, tighter building envelopes, and high-efficiency lighting.

Of course, excess supply airflow and the increased humidity that accompanies it also can result from the addition of "safety factors" or a conservative estimate of the space sensible load.

Selecting equipment based on accurate load calculations is the best way to assure thermal comfort at all load conditions. ●

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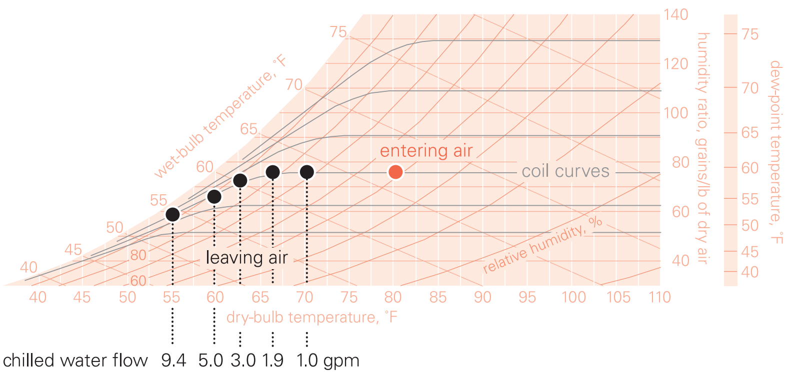
Figure 2. Part-load dehumidification with modulated chilled water flow



coil surface actually falls to the drain pan below. Like raindrops on a window, droplets on the coil fins must accumulate enough mass for gravity to overcome surface tension.

When the compressor stops, sensible cooling drops off dramatically; meanwhile, latent cooling not only falls to zero but actually becomes *negative* as the condensate on the coil's surface re-evaporates into the supply air stream.

Figure 4 shows the psychrometric effect of compressor cycling on the air leaving the coil. While the compressor is on, the leaving air is cold and dry. But when the compressor stops, the leaving air quickly warms. Due to re-evaporation of moisture from the coil surface, the humidity ratio of the air is higher when it leaves the coil than when it entered. Then the leaving-air wet-bulb temperature remains relatively constant as the moisture evaporates; if the coil dries completely, the leaving and entering air conditions match.

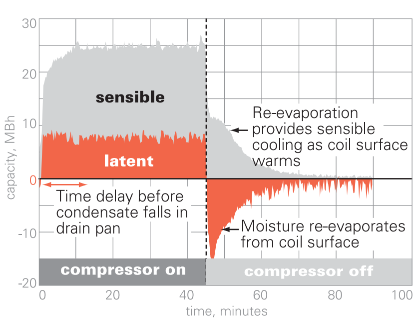
Bear in mind that the test data in Figure 3 and Figure 4 was based on 45-minute on/off cycles. In actual operation, compressor off-time depends on sensible load and may last only a few minutes—too briefly to permit the coil to dry completely between compressor starts.

Figure 5 shows *net* dehumidification capacity as a function of compressor duty cycle or percent run-time.[1] (Net dehumidification equals all of the moisture that condenses on the coil *minus* the condensate that re-evaporates.) Notice that as the compressor operates for less of each hour, the system provides little or no dehumidification. Because of the brief "on" periods, the coil doesn't generate enough condensate to drain into the pan below; and long "off" periods allow more condensate to re-evaporate from the coil.

Note: The relationship between compressor run-time and dehumidification capacity (Figure 5) also demonstrates the effect of oversizing constant-volume DX equipment. The compressor runs even less of each hour, resulting in warmer supply air and less dehumidification.

Figure 6 compares basic, constant-volume dehumidification at two outdoor conditions: full load (design) and part load (cooler but more humid).[2] Notice that space humidity climbs from 52% RH at full load to 67% RH at part load. That's because the *sensible* cooling capacity of a constant-volume system decreases to match the smaller *sensible* cooling

Figure 3. Part-load dehumidification (cycling compressor, constant-volume fan)



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

[1] To aid coil selections and performance analyses, the Trane psychrometric chart includes coil curves that approximate the part-load dehumidification of cooling coils (whether chilled water or refrigerant). These curves can be invaluable for analyzing part-load performance.

[2] This example, which appeared in a previous *Engineers Newsletter* ("It may take more than you think to dehumidify with constant-volume systems," volume 29-4), used an elementary school classroom to demonstrate full-load versus part-load dehumidification performance of a basic, constant-volume system.

Figure 4. Effect of compressor cycling on air leaving the cooling coil

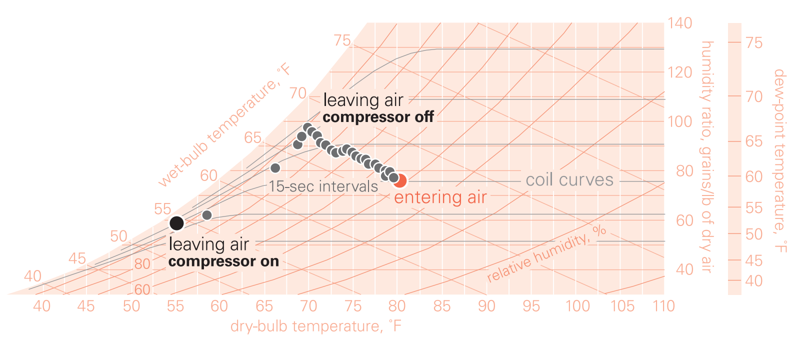
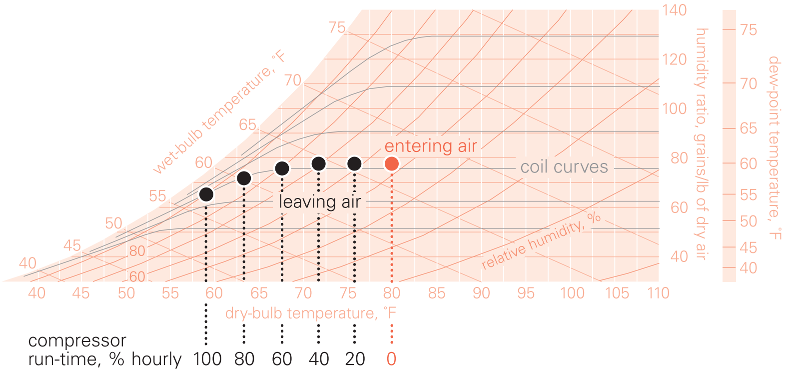


Figure 5. Net dehumidification as a function of compressor run-time



load. Any latent cooling (dehumidification) capacity is purely coincidental, whether the cooling-coil medium is chilled water or refrigerant. As the load diminishes, the system delivers ever warmer supply air. *Some* dehumidification may occur … but only if the sensible load is high enough.

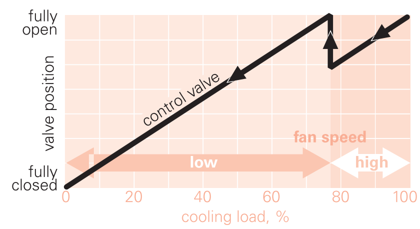
To avoid moisture problems, it's important to understand how well the HVAC system will dehumidify at both full load and part load, *and to know how to improve its performance*. Here's a brief overview of ways to improve the dehumidification performance of three familiar system types—chilled water terminal, single-zone DX, and central VAV.

Chilled water terminal systems

In a chilled water terminal system, a central plant distributes cold water to terminal units (such as unit ventilators, fan-coils, and blower-coils) located within or near each conditioned space. Each terminal unit is a single-zone, constant-volume, mixed-air system whose part-load dehumidification can be improved by adding automatic fan-speed adjustment, face-and-bypass dampers, a subcool–reheat strategy, and/or a dedicated outdoor-air system.

Automatic fan-speed adjustment. Terminal units with multispeed supply fans first reduce cooling capacity by automatically switching to a slower fan speed. At full load (Figure 7), the fan operates at high speed and the control valve is wide open. As the cooling load decreases, the thermostat modulates the valve to throttle the rate of chilled water flow through the coil. At some point, based on valve position (for example), the unit controller switches the fan to low speed. Less airflow means that colder supply air is needed to maintain the target space temperature. The control valve opens, chilling the coil surface and allowing the coil to remove more moisture from the passing air stream.

Figure 7. Automatic fan-speed adjustment for chilled-water terminal units



Note: When the unit controller switches the fan to low speed, it also should automatically adjust the position of the outdoor air damper to assure proper ventilation.

Face-and-bypass dampers provide an alternate means of controlling coil capacity. The cooling coil "runs wild" (constant water flow rate, constant entering water temperature) and the zone thermostat modulates the position of the linked dampers.

By itself, *mixed-air bypass* typically offers little benefit in non-arid climates because the diverted air (Figure 8, p. 4) is relatively humid at many part-load conditions. However, combining mixed-air bypass with automatic fan-speed adjustment can improve dehumidification significantly.

Ideal *return-air bypass* (full coil face available as the face damper closes) directs all incoming outdoor air through the cooling coil. Consequently, it's a better choice in situations where the humidity ratio of the outdoor air is higher than that of the return air.

Figure 6. Comparison of full- and part-load dehumidification (basic, constant-volume system)

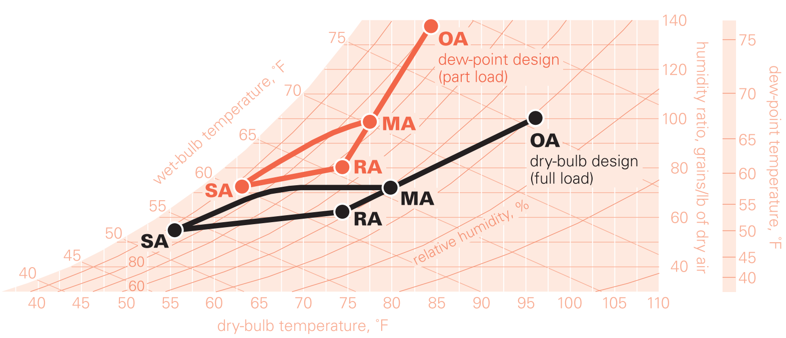
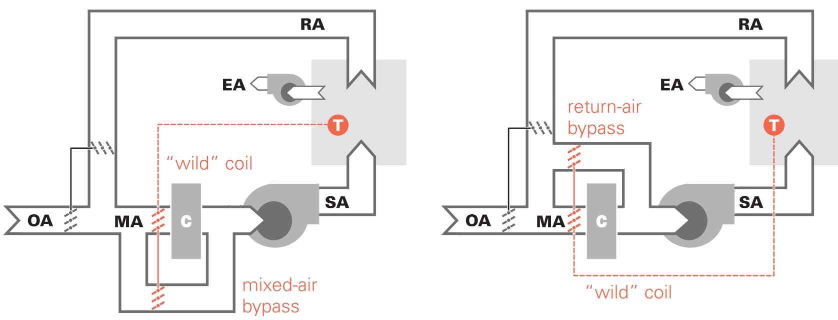


Figure 8. Mixed-air and return-air bypass arrangements



Note: The design of terminal units, such as unit ventilators, affords little space for return-air bypass. Often, face dampers in this arrangement block part of the cooling coil as they close. Less face area at part load severely limits the coil's ability to dehumidify.

Subcool–reheat. Adding a heat source downstream of the cooling coil and a humidity sensor in the space makes it possible to directly control both temperature *and humidity*. Here's how …

If the detected humidity in the space exceeds a preset limit, cooling-coil capacity increases to dehumidify the supply air. The downstream heating coil then warms (reheats) the supply air to avoid overcooling.

A boiler can serve the reheat coil—but only if it's available to operate during the cooling season. Recovering heat from the chilled water system for this purpose offers several benefits. It may permit seasonal shutdown of the boiler; reduce system operating costs by avoiding the use of "new" energy for reheat; and enable compliance with local codes and energy standards, such as ASHRAE Standard 90.1.[3]

Dedicated outdoor-air system. Directing all outdoor air through a dedicated air handler can eliminate the sensible and latent cooling loads associated with ventilation. The conditioned air then passes from the air handler directly to the space or to local terminal units (Figure 9). Each local terminal unit conditions only the recirculated return air to maintain the target temperature in the space.

For most efficient performance, make sure that the conditioned outdoor air is:

- *drier than the space,* which offsets the latent load in the space and eases the dehumidification burden on the terminal units; and,
- *cold enough, whenever possible,* to offset part of the sensible load in the space. "Neutral" conditioned air requires not only more capacity from the terminal units but also reheat at the dedicated outdoor-air unit.

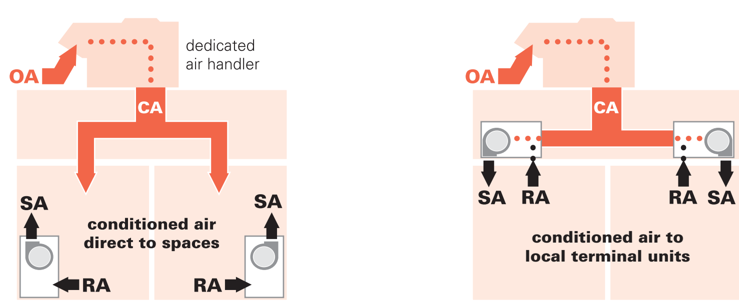
Specify the dedicated outdoor-air unit to limit relative humidity to some maximum level (typically 60–65% RH) at worst-case conditions. Designing for drier indoor air requires larger equipment and increases operating costs. Also consider communicating controls to optimize energy use, improve comfort (two-pipe systems), and independently limit humidity during unoccupied periods.[4]

Single-zone DX systems

Equipment in this category is installed in or near the space and includes packaged rooftop units, split systems, packaged terminal air conditioners, and water-source heat pumps. Each of these units is a single-zone, mixed-air device that contains a constant-volume supply fan and a compressor that cycles on and off.

Single-zone DX systems are notorious for losing humidity control at part load due to the natural cycling operation of DX equipment and the need for constant ventilation with outdoor air. Oversizing often compounds the problem (see inset, p. 1.) In applications that require an unusually high percentage of outdoor air, the cfm/ton limitations of packaged units often make it necessary to select a unit that delivers more airflow than the cooling load requires at design. But don't forget—more airflow means

Figure 9. Dedicated outdoor-air arrangements for chilled water terminal systems



[3] Schwedler, M. and D. Brunsvold. 2003. *Waterside Heat Recovery in HVAC Systems* (SYS-APM005-EN). La Crosse, WI: Trane.

[4] See Trane *Engineers Newsletter* 30-3 (2001), "Design Tips for Effective, Efficient Dedicated Ventilation Systems."

warmer supply air and less dehumidification at *all* loads. Such situations demand particular design care to assure that the DX system will dehumidify effectively over a wide range of conditions.

Automatic fan-speed adjustment. In a traditional DX unit with a constant-speed supply fan, the fan operates whenever the space is occupied. By contrast, in a unit with a multiple-speed fan, the fan operates at low speed whenever possible and only switches to high speed when the load is large. Less supply airflow at part load results in a lower supply-air temperature and improves dehumidification. (For proper ventilation at low airflow, the unit controller should appropriately and automatically adjust the position of the outdoor-air damper.)

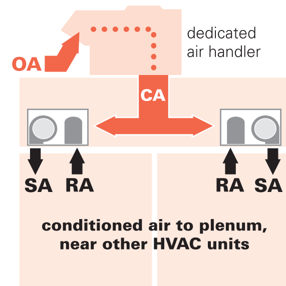
Combining a multispeed supply fan with multiple compressors (or with a compressor that unloads) further improves dehumidification because it lengthens the compressor "on" cycle when supply airflow (and temperature) are low. It also helps prevent the suction temperature from becoming too cold, and perhaps frosting the coil, during low-airflow operation.

Note: A variation of this concept is the single-zone VAV system, which modulates fan capacity (supply airflow) to maintain the desired space temperature and cooling capacity to maintain the supply-air temperature at setpoint.

Subcool–reheat. Adding a space-humidity sensor and a reheat coil permits direct control of both space temperature and space humidity. When humidity becomes excessive, cooling capacity increases to adequately dehumidify the air. The reheat coil then warms the supply air to avoid overcooling.

Reheating the supply air with recovered heat from the refrigeration cycle can reduce operating costs by avoiding the use of new energy. It also allows the system to meet local energy codes and ASHRAE Standard 90.1. Typically, "hot gas reheat" is accomplished by adding a refrigerant-to-air heat exchanger to the refrigeration circuit, between the compressor and the condenser. Whenever it's necessary to avoid overcooling the space, the heat exchanger transfers sensible heat from the hot refrigerant vapor to the supply air downstream of the cooling coil.

Figure 10. Dedicated outdoor air system with single-zone water-source heat pumps



Dedicated treatment of outdoor air. As in chilled water systems, you can improve the single-zone DX system's ability to control humidity by adding a dedicated outdoor-air handler … but with an important difference. It's usually preferable to deliver *cold* air to the local units, *but not when the local units are plenum-mounted water-source heat pumps.* Most WSHPs do not include mixing boxes, so the conditioned outdoor air must be delivered *in proximity to* the inlet of each heat pump (Figure 10). To avoid unwanted condensation, especially during unoccupied periods, dehumidify the outdoor air so that it's drier than the space and reheat it to assure that the dry-bulb temperature is well above the worst-case dew point in the plenum.

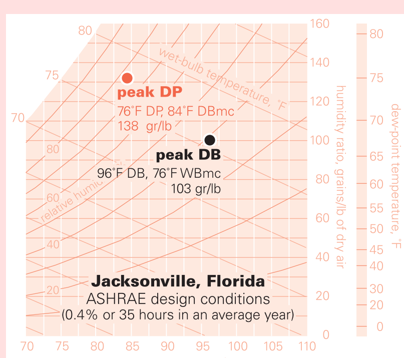
Make use of ASHRAE's design dew-point condition

With publication of the 1997 *Fundamentals* handbook, ASHRAE expanded outdoor design conditions to include "peak dew point." Unlike the traditionally used "peak dry bulb" condition, which represents hot, mostly sunny days, "peak dew point" conditions occur when the outdoor air is cooler but loaded with moisture.

"Peak dew point" can be valuable when designing a dedicated dehumidification system. It can be equally useful for analyzing coincidental dehumidification at part load, as provided by an HVAC system that's controlled on the basis of space dry-bulb temperature alone.

Realize, however, that the "peak dew point" seldom occurs during the same month, nor at the same time of day, as the "peak dry bulb." Because the month and/or time of occurrence (which affect solar and roof loads) may be different—as is the outdoor dry-bulb temperature (which affects conduction loads)—the sensible load in the space at each outdoor design condition may be quite different.

For HVAC systems that are controlled based on space temperature, it is the space *sensible* load that dictates how the system will dehumidify at part load. ●

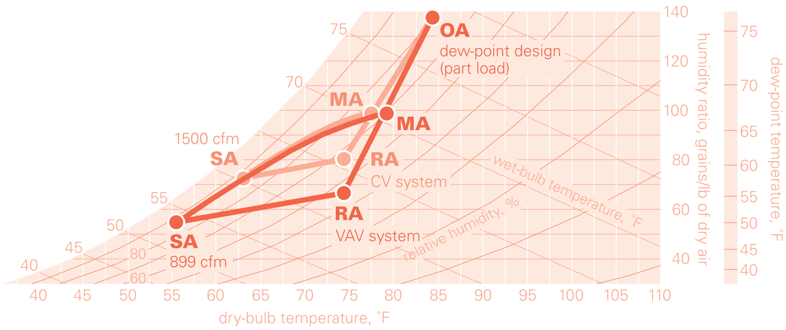


Central VAV systems

VAV systems typically dehumidify effectively over a wide range of indoor loads. As long as any space needs cooling, the VAV air handler will provide supply air at a dew point that's low enough (sufficiently cool and dry) to offset the latent load.

A basic system consists of an indoor or outdoor central air handler

Figure 11. VAV versus CV dehumidification at part load



and a VAV box for each zone. A thermostat maintains the target space temperature by modulating the volume of supply air through the VAV box. Meanwhile, the central supply fan delivers only the necessary amount of constant-temperature air.

As the sensible cooling load in the space decreases, the basic VAV system delivers less supply air. Because the supply air is cool and dry, the relative humidity in our classroom example only rises to 57% at the part-load condition, compared to 67% RH with the basic constant-volume system (Figure 11).

To improve the dehumidification of a central VAV system:

- **Avoid using SAT reset during the cooling season.** It's tempting to raise the supply-air temperature (SAT) at part-load conditions in an attempt to save cooling or reheat energy. But warmer supply air means less dehumidification at the coil and higher humidity in the space. It also increases the supply fan's energy consumption—perhaps enough to negate the cooling energy savings.

  For applications that include SAT reset, provide one or more humidity sensors to override the reset function if space humidity exceeds the desired limit.

- **Provide a tempering heat source at the VAV boxes.** Each VAV box has a minimum airflow setting that's based on the performance limits of the diffusers or terminal, or on the required ventilation in the space. Densely occupied spaces may require relatively high minimum airflow settings to assure proper ventilation.

  Eventually, the sensible cooling load becomes small enough that the space requires less supply airflow than the VAV terminal can deliver. Without tempering (reheat), the system will overcool the space, causing the relative humidity to rise and the space to feel cold and damp.

  If using a boiler as the heat source for tempering, make sure that it's available for cooling season operation. Alternatively, use recovered heat from another part of the system; this option allows the boiler to be shut down and avoids using new energy for reheat.

- **Investigate the practicality of delivering colder supply air.** Lowering the supply-air temperature reduces supply airflow and ultimately lowers the humidity in the space because more moisture condenses on the coil.

Closing thoughts

Dehumidification performance varies with the type of HVAC system. A basic, single-zone, constant-volume system may suffice for some applications in certain climates. But more demanding applications—particularly densely occupied spaces and humid climates—may require enhanced designs (or a different system type) to adequately manage or limit indoor humidity.

If properly designed, controlled, and operated, the HVAC system can effectively dehumidify over a wide range of conditions. The key to success is careful analysis of full-load and part-load performance.

Note: The dehumidification enhancements mentioned here are discussed in greater detail in Trane application manual SYS-APM004-EN, Dehumidification in HVAC Systems. To obtain a copy, contact your local Trane office or visit www.trane.com/bookstore. ●

By John Murphy, applications engineer, and Brenda Bradley, information designer, Trane. You can find this and previous issues of the *Engineers Newsletter* at http://www.trane.com/commercial/newsletters.asp. To comment, e-mail us at comfort@trane.com.

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Dehumidification in HVAC Systems uses basic psychrometric analyses to review the dehumidification provided by "cold coil" commercial comfort-cooling HVAC systems. It also identifies ways to improve dehumidification, particularly at part-load conditions.

- Chapter topics: sources and effects of indoor moisture; a dehumidification primer; dehumidifying with constant-volume mixed air, variable-volume mixed air, and dedicated outdoor air
- "How to" appendices for psychrometrically analyzing dehumidification performance and designing a dedicated outdoor-air system
- 140 pages; indexed for easy reference
- Published December 2002

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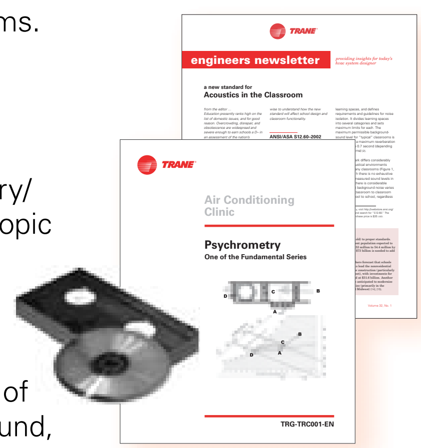

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