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Dehumidify with Constant-Volume Systems

from the editor…
ASHRAE Standard 62, "Ventilation for Acceptable Indoor Air Quality," recommends that the relative humidity not exceed 60 percent at any load condition. This can be problematic because the Standard increases the minimum outdoor-air requirement. Many HVAC designers prefer a low-cost constant-volume solution, believing that it also simplifies ventilation and inherently provides sufficient dehumidification.

This newsletter reveals the flaw in that belief. Dennis Stanke, Trane staff engineer and member of ASHRAE SSPC 62.1, uses psychrometric analyses to demonstrate the difficulty of providing proper dehumidification—particularly at part load, when dry-bulb temperature determines system capacity. He also discusses several design options that improve the latent capacity of a constant-volume system and compares their effectiveness.

The Difficulty with CV Dehumidification

Contrary to popular belief, indoor moisture control is an issue in almost all geographic locations, not just in areas where hot, humid conditions prevail. Whenever a high relative humidity exists at or near a cold, porous surface, moisture absorption increases and moisture-related problems (increased maintenance, premature replacement of equipment and furnishings, and increased health risks) become likely.

"Ironically, the widely used single-zone CV system is particularly problematic for dehumidification."

If properly designed and controlled, the HVAC system can significantly reduce the moisture content of indoor air. Ironically, the most widely used means of ventilation—the single-zone, constant-volume (CV) system—is also the most problematic when it comes to dehumidification.

A basic CV system consists of an air handler that serves a single thermal zone. The air handler supplies the zone with a constant volume of air, usually a mixture of outdoor air and recirculated return air, at a variable temperature.

A thermostat senses the zone dry-bulb temperature and compares it to the set point. The thermostat then modulates the capacity of the cooling coil, adjusting the supply-air temperature until the sensible capacity of the cooling coil matches the sensible load and the zone temperature matches the set point.

Designers typically (and appropriately) size cooling coils based on the peak sensible load, that is, when it is hottest outdoors. In many climates, however, the *latent* load on the cooling coil—and often the total load (sensible plus latent)—peaks when outdoor dew point, not dry bulb, is highest.

Consequently, in some air-handler arrangements, coils selected for the highest sensible load may not provide sufficient cooling capacity when the highest latent load occurs. More importantly, however, coils *controlled to maintain the dry-bulb temperature* in the space often operate without adequate latent capacity at part-load conditions. Here's why…

At peak sensible load, the cooling coil removes both sensible and latent heat, directly controlling zone temperature and indirectly affecting the relative humidity. (A colder coil increases the rate at which moisture condenses from the air.) At partial sensible load, however, the control system reduces the capacity of the cooling coil by allowing the coil temperature to rise. Although this action successfully maintains the zone dry bulb, it also slows the rate of condensation: relative humidity in the zone rises.

Sizing the cooling coil for the highest *total* load will not prevent this shortfall in latent capacity if system control is based solely on sensible conditions. Whenever a part-load condition exists, the thermostat throttles the coil, latent capacity drops, and zone relative humidity increases.

An Example. Let's consider a 10,000-cubic-foot classroom in Jacksonville, Florida, that accommodates 30 people. The thermal comfort *target* is 74°F DB and 50%RH, with nine air changes per hour (9 ACH). To provide adequate ventilation, 450 cfm (15 cfm per person) of the 1,500 cfm of supply air must be introduced from outdoors.

Note: Some codes require a specific air-change rate for classrooms, so we assumed an airflow and calculated the supply-air temperature for this example. Alternatively (and perhaps more commonly), we could assume a supply-air temperature and calculate the supply airflow.

Basic system at sensible design and full load. Chapter 26 in the 1997 *ASHRAE Handbook—Fundamentals* indicates that Jacksonville's outdoor dry bulb equals or exceeds 96°F during 0.4 percent (35 hours) of an average year; the coincident wet bulb averages 76°F.

At this design condition, the sensible and latent loads calculated for the space—29,750 Btu/h and 5,250 Btu/h, respectively—yield a sensible heat ratio of 0.85. Given the supply airflow of 1,500 cfm, a supply-air temperature of 55.7°F is required to meet the sensible load and cool the space to 74°F.

But does this supply-air condition achieve the target relative humidity of 50 percent? The psychrometric analysis summarized in Figure 1 illustrates the answer. Simply controlling the zone temperature to 74°F results in a comfortable 52.4%RH and requires 4.78 tons of cooling to satisfy both the sensible and latent loads on the coil.

Sensible versus Latent Design

Chapter 26 of the 1997 *ASHRAE Handbook—Fundamentals* is a popular source for tabular, climatic data that represents the outdoor design conditions for many locations. **Sensible design** conditions for cooling systems appear under the heading "Cooling DB/MWB" (cooling dry bulb/mean coincident wet bulb). The tables also indicate the frequency of occurrence for each condition. For example, weather conditions exceed the values listed in the "0.4%" column for just 35 hours in an average year.

Latent design conditions, labeled "DP/MDB and HR" (dew point/mean coincident dry bulb and humidity ratio), were added to the 1997 edition of the handbook to aid the design of dehumidification systems.

Because the peak sensible load rarely occurs at the same time as the peak latent load, cooling equipment that is selected and controlled to deliver full capacity at sensible design is likely to deliver less-than-required capacity at latent design. Therefore, system performance must be analyzed at sensible *and* latent design conditions, based on the moisture control needs of the application.

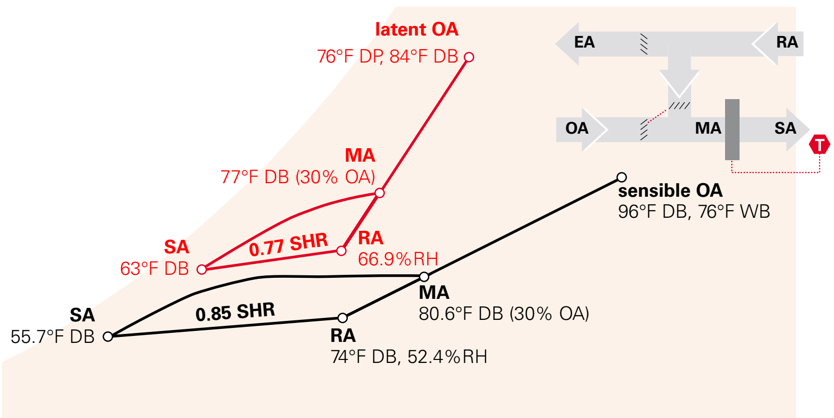
As the table below illustrates, latent design conditions can be similar in many locations. Ignoring system operation at latent design can lead to poor dehumidification in buildings across the country, not just in the South and Southeast. ■

U.S. Cooling and Dehumidification Design Conditions, 0.4% Frequency of Occurrence[a]

	Cooling DB/MWB		DP/MDB and HR		
Station	DB	MWB	DP	HR	MDB
Chicago, Illinois	91°F	74°F	**74°F**	130	84°F
Dayton, Ohio	90°F	74°F	**73°F**	129	82°F
Harrisburg, Pennsylvania	92°F	74°F	**74°F**	130	82°F
Jacksonville, Florida	96°F	76°F	**76°F**	138	84°F

[a]Excerpt from Table 1B in Chapter 26 of the 1997 *ASHRAE Handbook–Fundamentals.*

Figure 1. Basic CV System (Classroom Example)



Basic system at latent design and part load. The 1997 ASHRAE handbook also shows that, for 0.4 percent of the time, the outdoor dew point equals or exceeds 76°F while the coincident dry bulb averages 84°F. Let's see what happens in the zone if the sensible load drops to 60 percent of sensible design (17,850 Btu/h) as a result of a lower outdoor-air temperature and the correspondingly lower solar and conducted heat gains.

If we also assume that the latent load due to occupants remains unchanged (5,250 Btu/h), the sensible heat ratio drops to 0.77. Now 1,500 cfm of supply air at 63°F satisfies the sensible load.

As Figure 1 illustrates, the warmer, moister supply air raises the relative humidity in the classroom from 52.4%RH to 66.9%RH—well above the 60%RH maximum that ASHRAE recommends. Although the coil could provide additional cooling (up to 4.78 tons, if sized for the sensible design load), the thermostat reduces coil capacity to 3.68 tons.This action maintains the dry-bulb temperature in the classroom at set point, but at the expense of the system's ability to dehumidify.

Packaged Air Conditioning Compounds the Problem. A chilled water coil can be selected to deliver 4.78 tons of cooling at 1,500 cfm, a flow-to-capacity ratio of 314 cfm per ton. Most packaged air conditioners, however, must operate within a narrow range of flow-to-capacity ratios, usually between 350 and 450 cfm per ton.

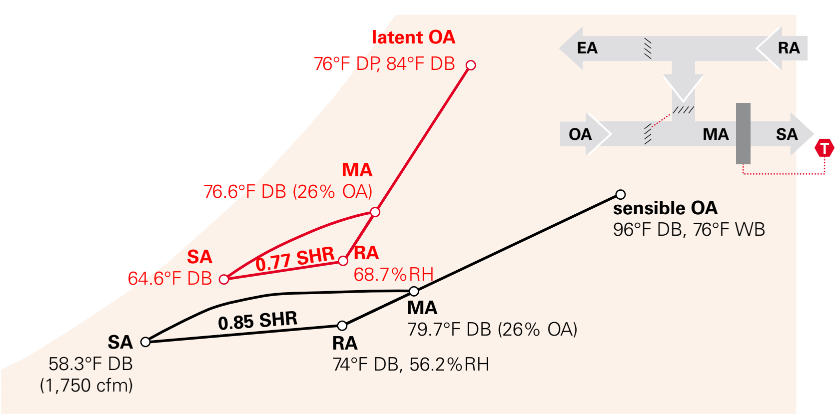
The classroom in our example requires a five-ton unit that delivers no less than 1,750 cfm (350 cfm per ton). To assure adequate cooling capacity, the designer must accept an air-change rate of 10.5 ACH instead of the desired 9 ACH.

The higher-than-required supply airflow (1,750 cfm) increases the supply-air temperature to 58.3°F; see Figure 2. As the total coil load drops from 4.78 to 4.66 tons, the humidity in the classroom increases from 52.4%RH to 56.2%RH at sensible load.

Not surprisingly, the classroom becomes even more humid when the sensible load drops to 60 percent (the latent design condition). With the thermostat throttling the coil capacity to 3.62 tons, the 64.6°F supply air removes even less of the latent load and the relative humidity climbs to 68.7%RH.

Note: An overly conservative estimate of the sensible load in the zone also results in too much supply airflow, along with the attendant increase in relative humidity.

Figure 2. Packaged Air Conditioner (Classroom Example)



Enhancing Indirect Dehumidification

A typical, constant-volume, mixed-air system uses a single cooling coil to cool and dehumidify a mixture of recirculated air and outdoor air. A thermostat modulates the cooling capacity of the coil to directly control zone temperature in response to changes in the sensible load. Coil capacity *indirectly* controls space humidity: less cooling capacity means less dehumidification and vice versa.

Various options are used to improve the indirect dehumidification of a typical CV system (Figure 1). Let's examine the effectiveness of three of them:

- Total energy recovery
- Mixed-air (MA) bypass
- Return-air (RA) bypass

Note: One of the best enhancements for indirect dehumidification controls zone temperature by varying the flow of supply air rather than its temperature (see "What About VAV Systems?" on page 6). This article, however, purposely limits the discussion to constant-volume systems.

Total Energy Recovery. Some designers find that passive energy-recovery systems provide adequate dehumidification. A passive, *total*-energy-recovery wheel (ERW), for instance, preconditions the outdoor air and reduces the cooling capacity needed to maintain the zone temperature. It removes both latent and sensible heat from the outdoor-air stream entering the building, which indirectly reduces zone relative humidity while saving significant operating energy.

Adding an ERW also helps the designer select a packaged system based on less airflow. That is, the designer can use an airflow that more closely matches the zone requirement (within the constraints of the flow-to-capacity ratio) by raising the unit airflow per ton.

Figure 3 illustrates the psychrometric effect of adding an energy-recovery wheel to the packaged air conditioner represented in Figure 2. When the sensible design condition exists, preconditioning the outdoor air reduces the coil load from 4.66 tons to 3.5 tons and permits an equipment selection based on 1,500 cfm rather than 1,750 cfm.

This reduction of supply airflow permits colder supply air (55.7°F rather than 58.3°F), increasing the latent capacity of the coil at all loads. As a result, the relative humidity in the classroom drops from 56.2%RH (Figure 2) to 50.4%RH.

Notice, however, that the latent-design, part-load condition still requires a supply-air temperature of 63°F. Although it reduces the coil load from 3.62 tons to 2.47 tons, *the energy-recovery wheel does little to improve indirect dehumidification*. The resulting relative humidity (now 65%RH rather than 68.7%RH) still exceeds the 60%RH maximum recommended by ASHRAE.

Further dehumidification cannot occur without making the mixed-air humidity ratio less than that of the return air. A total-energy-recovery device such as the ERW cannot perform this task without the help of a cooling coil.

Mixed-Air Bypass. Face-and-bypass dampers arranged to bypass mixed air are often used to extend the "indirect" dehumidification range of a constant-volume air handler. Simple and inexpensive, this option blends cold, dry air leaving the cooling coil with warm, moist, mixed air (return air and

Figure 3. Basic CV System with Total Energy Recovery (Classroom Example)

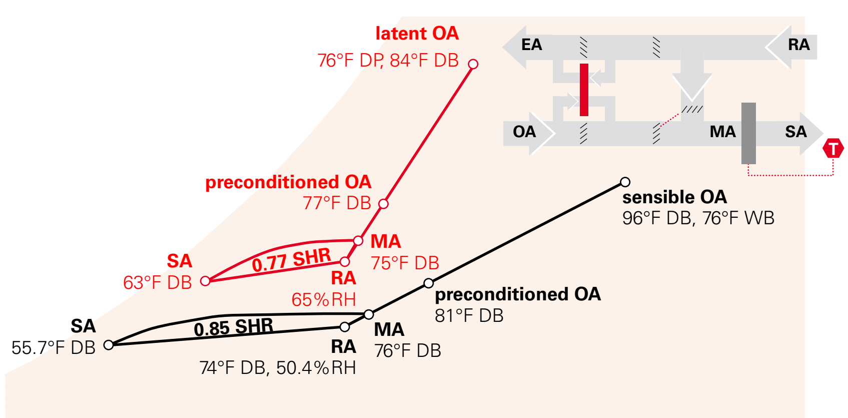
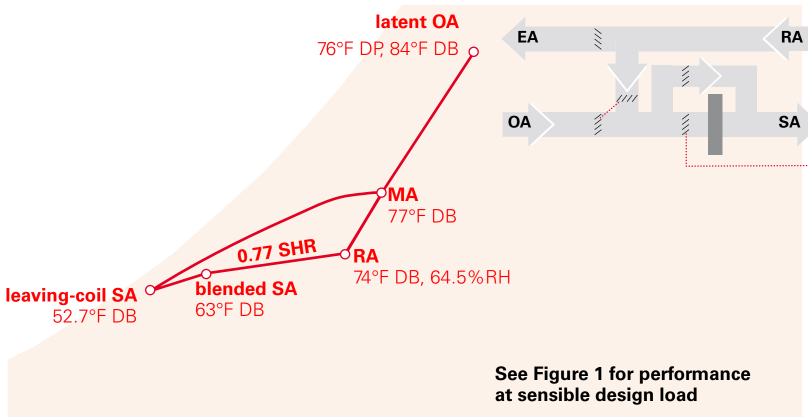


Figure 4. Basic CV System with Mixed-Air Bypass (Classroom Example)



outdoor air) to achieve the proper supply-air temperature. The zone thermostat controls capacity by adjusting the face-and-bypass dampers, regulating airflow through and around the coil. Chilled water flow through the coil is constant, not modulated.

All mixed air passes through the cooling coil when a sensible design load exists, making dehumidification performance identical to that shown in Figure 1.

Figure 4 illustrates the classroom condition that results at the part-load latent-design condition when the blended supply-air temperature is 63°F.

Using a coil performance program, we determined that the leaving-coil temperature falls to 52.7°F. Moisture in the bypassed air prevents more than a slight decrease in relative humidity (from 66.9%RH to 64.5%RH) and increases the total coil load from 3.68 to 3.74 tons.

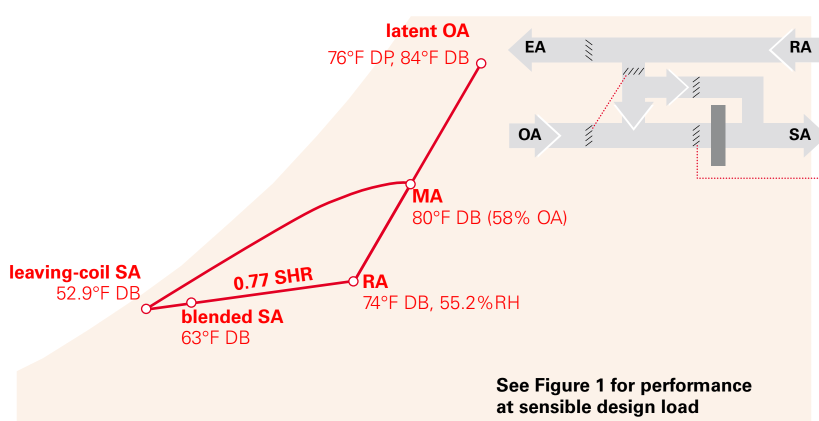
Return-Air Bypass. In many climates, face-and-bypass dampers arranged to bypass return air provide a cost-effective way to extend the indirect dehumidification range of a CV air handler. Although ducting may increase its cost slightly compared with mixed-air bypass, return-air bypass limits relative humidity better than any other indirect dehumidification enhancement at both sensible and latent (part-load) design conditions.

Like the mixed-air version, the return-air bypass modulates coil capacity by adjusting airflow rather than water flow. This means that the coil surface can be very cold, enhancing the ability of the system to dehumidify the zone without directly controlling humidity.

What makes the return-air bypass more effective, however, is that it directs *all* of the moist outdoor air through the cooling coil. Relatively dry return air (rather than moist mixed air) reheats the cold air stream leaving the coil. When a sensible design load exists, the entire mixed-air stream passes through the cooling coil. Psychrometric performance matches Figure 1.

Figure 5 summarizes the effect of adding less moisture at latent design. Again, we used a coil performance program to determine the leaving-coil temperature of 52.9°F. Relative humidity in the classroom falls below the ASHRAE-recommended high limit, dropping from 66.9%RH to 55.2%RH. Maintaining this level of dehumidification requires a total cooling capacity of 3.92 tons.

Figure 5. Basic CV System with Return-Air Bypass (Classroom Example)



What About VAV Systems?

Variable-air-volume (VAV) systems provide effective, indirect dehumidification over a very wide range of indoor load conditions (that is, sensible heat ratios). As long as any zone needs cooling, the VAV air handler supplies dry (low-dew-point) air to all terminal units. The dry supply airflow, modulated to control the *sensible* indoor load directly, removes the latent indoor load *indirectly* by absorbing space-generated moisture and removing it with the return air.

If the sensible indoor load drops below the minimum cooling capacity provided by the minimum flow of supply air, sensible heat must be added either at the terminal unit (to temper the supply air) or within the zone. Failing to temper the supply air (or to increase the sensible load) overcools the zone without dehumidifying it, making it feel clammy.

Typically, some VAV zones require tempering heat even when high sensible loads exist in the others. Always consider on-site recovered energy as the source for supply-air tempering, whether the system is VAV or constant volume. ■

Direct Dehumidification

Indirect dehumidification enhancements may work well for some indoor environments in some climates, even though latent and sensible load peaks occur independently. But when latent and sensible loads vary significantly, or when it is necessary to maintain a low relative humidity, both sensible and latent capacity must be controlled *directly* from both zone temperature *and* zone relative humidity.

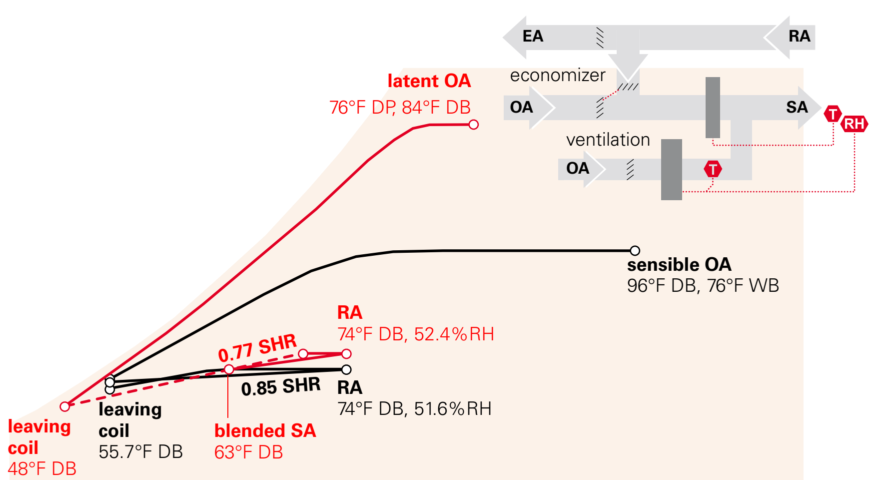
Separate Paths. One way to directly control dehumidification is to individually treat the return air and outdoor air streams before mixing them. This can be accomplished with two entirely separate air handlers, or with a single, "dual-path" air handler (usually in a stacked configuration for a smaller footprint) that accommodates both airflow paths. Air treatment may include various degrees of cooling, dehumidifying, heating, and filtering.

Together, individual cooling coils in the return air and outdoor air streams maintain the target condition in the zone. That is:

- The humidistat directly controls the latent capacity of the outdoor-air coil to maintain the desired relative humidity limit. It provides sufficiently dry air to remove the latent load, both outdoor and zone.
- The thermostat directly controls the sensible capacity of the return-air coil, providing the balance of cooling needed to assure that the supply-air temperature satisfies the sensible load. In effect, heat in the return air tempers the preconditioned outdoor air.

Figure 6 summarizes the psychrometric effect of the separate air-treatment paths in our classroom example. With both coils delivering blended 55.7°F supply air at the sensible design load, humidity drops to 51.6%RH, slightly drier than the zone condition resulting from the simple, single-coil system (Figure 1). Total coil load, which rises from 4.78 to 4.81 tons, is split between the outdoor coil (2.15 tons) and the return coil (2.66 tons).

Figure 6. CV System with Separate Paths for Air Treatment (Classroom Example)



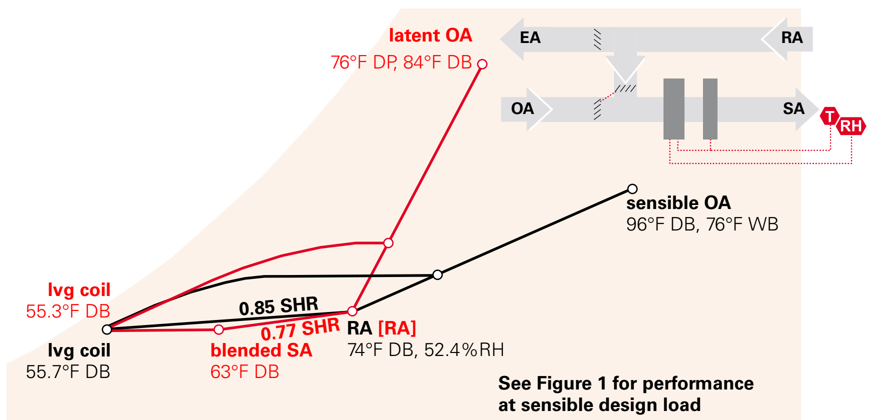
As the latent load rises, the humidistat increases the capacity of the outdoor coil by reducing the leaving-coil temperature and maintaining the indoor humidity at the 52.4%RH limit. As the indoor sensible load drops, the zone thermostat reduces the capacity of the return coil accordingly to maintain the room temperature at set point, 74°F. Total coil load rises from 3.68 tons (Figure 1) to 4.16 tons, again split between the outdoor coil (3.72 tons) and the return coil (0.44 tons).

In this arrangement, the cooling system/chiller plant can be sized for "block" load (4.81 tons) rather than "peak" load (6.38 tons) because the sensible and latent loads do not peak simultaneously. Each coil, however, must be sized for its individual peak load—the return-air coil for 2.66 tons at sensible design and the outdoor-air coil for 3.72 tons at latent design.

Supply-Air Tempering.
Dehumidification can also be directly controlled by applying a single cooling coil in series with a heating device (Figure 7). This approach assures that the supply air is always dry enough to neutralize both the outdoor and indoor latent loads and cool enough to maintain the desired zone temperature.

Note: More commonly known as supply-air "reheat," we choose to describe this configuration as supply-air "tempering" because the heating device simply moderates the cooling effect of the dry supply air.

Figure 7. CV System with Supply-Air Tempering (Classroom Example)



To understand how supply-air tempering works, let's turn again to our classroom example. A coil load of 4.78 tons maintains the desired classroom condition—without tempering—at the sensible design load; air conditions are identical to those shown in Figure 1.

As the latent load rises, however, the zone humidistat increases the capacity of the cooling coil, which reduces the supply-air temperature and maintains the relative humidity at the 52.4% RH limit. As the sensible load in the classroom falls, the zone thermostat increases the tempering capacity of the heating device to maintain the zone temperature at set point, 74°F; see Figure 7. Total coil load rises from 3.68 tons to 5.21 tons.

When designing a CV system that includes supply-air tempering, size the cooling system and coil to handle both outdoor and zone loads at the sensible *or* latent design condition. (In our example, the peak cooling load of 5.21 tons occurs at latent-design conditions.) Remember, too, that on-site energy-recovery enhancements can provide the minimal heat required for tempering. In some cases, building codes or energy standards *require* the use of on-site recovered energy for tempering.

Because the chilled water system or the packaged-unit refrigeration system removes the latent load when the sensible load is low, always consider waterside or condenser heat-recovery options when using supply-air tempering to control relative humidity.

To Recap

Simple, constant-volume HVAC systems inevitably cause high zone humidity at sensible part-load conditions. Any of several design options can improve part-load dehumidification. Table 1 ranks the dehumidification enhancements considered for our example classroom from "best" (separate paths) to "poorest" (basic cooling only) based on how well they control humidity.

Ultimately, however, the "best" system choice must also consider first cost and operating cost as well as climate and zone loads.

To design a constant-volume system for effective dehumidification:

- Limit zone relative humidity at all load conditions.

  Analyze system performance at both sensible and latent design conditions when sizing the cooling coil. The maximum coil load may occur at the wettest, not the hottest, outdoor condition.

  Determine the need for dehumidification during unoccupied periods. Moisture from infiltration, wet-process cleaning, or vapor pressure diffusion can significantly increase the relative humidity indoors.

- Consider the advantages and disadvantages of each dehumidification enhancement.

  Remember that *indirect* dehumidification, which depends on a sensible load, is often ineffective during unoccupied periods.

- Pick the dehumidification enhancement that works best for the application, given the budget.

Table 1. Comparison of Dehumidification Enhancements[a]

Enhancement		Effectiveness[b]	Sensible Design		Latent Design	
			Zone RH	Cooling Req'd	Zone RH	Cooling Req'd
Basic CV system		**6 (poorest)**	**52.4%**	**4.78 tons**	**66.9%**	**3.68 tons**
Indirect:	total energy recovery	4	50.4%	3.50 tons	65.0%	2.47 tons
	mixed-air bypass	5	52.4%	4.78 tons	64.5%	3.74 tons
	return-air bypass	3	52.4%	4.78 tons	55.2%	3.92 tons
Direct:	**separate paths**	**1 (best)**	**51.6%**	**4.81 tons**	**52.4%**	**4.16 tons**
	supply-air tempering	2	52.4%	4.78 tons	52.4%	5.21 tons

[a] Comparison is based on a 30-person, 10,000-cu-ft classroom in Jacksonville, Florida. Supply airflow is 1,500 cfm of which 450 cfm is outdoor air for ventilation.

[b] "Effectiveness" ranks the dehumidification enhancements based on the lowest relative humidity (RH) in the zone and fewest cooling tons.

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*For more information about humidity management, refer to **Managing Building Moisture** (SYS-AM-15). To review the fundamentals of psychrometric analysis, see **Air Conditioning Clinic: Psychrometry** (TRG-TRC001-EN). You can order either publication from www.trane.com/bookstore.*

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