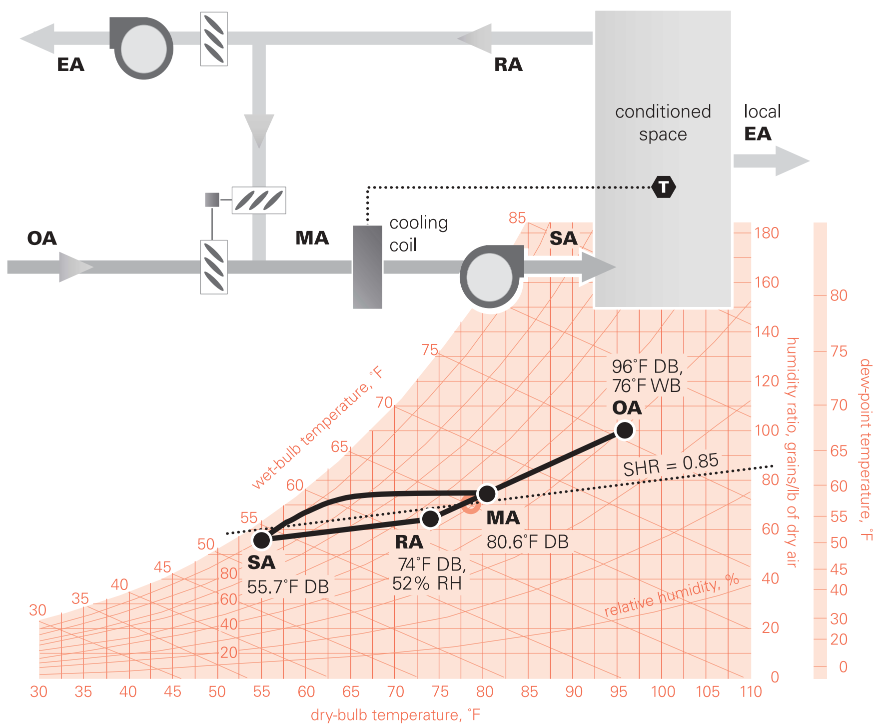
Applications Engineering Manual

Dehumidification in HVAC Systems



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Dehumidification in HVAC Systems

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Preface

As a leading HVAC manufacturer, we believe that it is our responsibility to serve the building industry by regularly disseminating information gathered through laboratory research, testing programs, and practical experience. Trane publishes a variety of educational materials for this purpose. Applications engineering manuals, such as this document, can serve as comprehensive reference guides for professionals who design building comfort systems.

This manual focuses on **dehumidification** (the process of removing moisture from air), as performed by HVAC systems in commercial comfort-cooling applications. Using basic psychrometric analyses, it reviews the dehumidification performance of various types of "cold-coil" HVAC systems, including constant-volume, variable-volume, and dedicated outdoor-air systems. In each case, full-load and part-load dehumidification performance is compared with the 60 percent-relative-humidity limit that is currently recommended by ANSI/ASHRAE/IESNA Standard 62–2001. This manual also identifies ways to improve dehumidification performance, particularly at part-load conditions.

We encourage you to familiarize yourself with the contents of this manual and to review the appropriate sections when designing a comfort-system application with specific dehumidification requirements.

Note: This manual does not address residential applications, nor does it discuss the particular dehumidification requirements for process applications, such as supermarkets, manufacturing, or industrial drying. ■

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Introduction

Uncontrolled moisture can reduce the quality of indoor air, make occupants uncomfortable, and damage a building's structure and furnishings. One form of moisture is water vapor entrained in the air.

Before the widespread use of air conditioning, humid weather meant high moisture levels indoors; indoor relative humidity remained acceptable, however, because the dry-bulb temperature indoors also increased. During warm weather, interior surfaces were only slightly cooler than the ambient temperature, so indoor condensation seldom occurred. The presence of any microbial growth primarily resulted from water leaks or spills, or from condensation on poorly insulated walls during cold weather.

Until 1970, designers typically chose constant-volume reheat or dual-duct systems to provide mechanical ventilation and air conditioning in commercial and institutional buildings. Both types of systems effectively (albeit coincidentally) controlled indoor humidity while regulating dry-bulb temperature. As the 1970s drew to a close, heightened concern about the availability and cost of energy prompted designers to choose system designs that neither used "wasteful" reheat energy nor mixed hot and cold air streams.

Although many of today's HVAC systems adequately control the indoor dry-bulb temperature, the lack of reheat or mixing allows humidity in the space to "float." High humidity levels can develop, especially during part-load operation. When coupled with the cold indoor surfaces that result from mechanical cooling, high humidity may lead to unwanted condensation on building surfaces.

The HVAC system and application influence the severity and duration of high indoor humidity. This manual therefore compares the dehumidification performance of several common types of HVAC systems. ■

Sources and Effects of Indoor Moisture

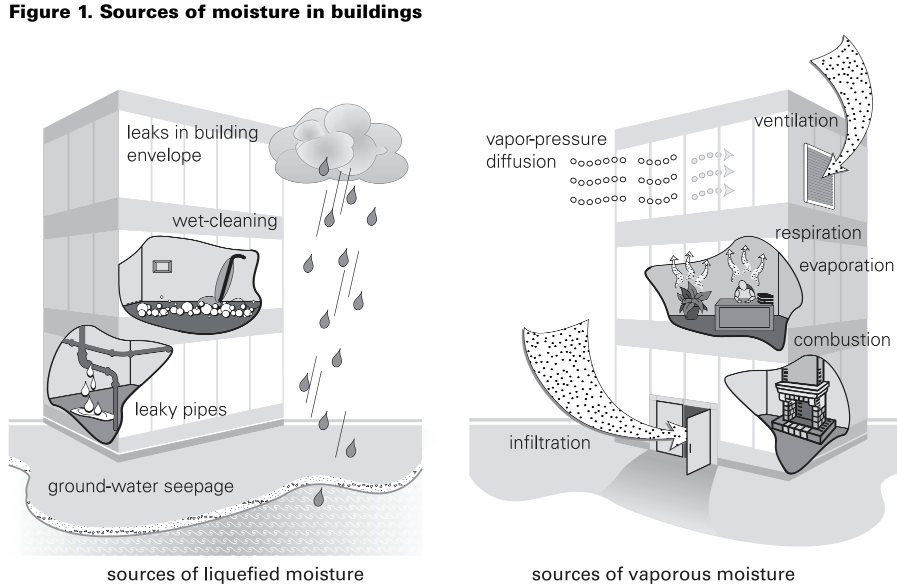
Refer to *Managing Building Moisture,* Trane applications engineering manual SYS-AM-15, for more information on sources of moisture in buildings, methods for calculating moisture-related HVAC loads, and techniques for managing moisture in the building envelope, occupied space, and mechanical equipment room. ■

Moisture can enter a building as a liquid or a vapor via several paths (Figure 1). It can cause problems in either form, and after it is inside the building, it can change readily from liquid to vapor (evaporation) or from vapor to liquid (condensation). To assure that the conditioned environment inside the building remains within the acceptable range, carefully evaluate *all* sources of moisture at *all* operating conditions when designing the HVAC system.

Liquid sources include ground-water seepage, leaks in the building envelope, spills, condensation on cold surfaces, and wet-cleaning processes (such as carpet shampooing). Roof leaks are a common source of unwanted water, especially in large low-rise buildings like schools. Leaking pipes, another common source, can be particularly troublesome because the leaks often develop in inaccessible areas of the building.

Water vapor develops inside the building or it can enter the building from outdoors. Indoor sources include respiration from people, evaporation from open water surfaces (such as pools, fountains, and aquariums), combustion, cooking, and evaporation from wet-cleaning. Outdoor sources include vapor pressure diffusion through the building envelope, outdoor air brought in by the HVAC system for ventilation, and air infiltration through cracks and other openings in the building envelope, including open doors and windows.

Figure 1. Sources of moisture in buildings



Sources and Effects of Indoor Moisture

Proper practices of design, construction, and operation can help minimize unwanted moisture inside the building. For example, proper landscaping can provide good drainage, periodic roof maintenance can help eliminate roof leaks, the building envelope can include a weather barrier to keep rain from penetrating the wall structure, and (depending on the season and climate) positive building pressurization can minimize the infiltration of humid outdoor air.

Why be Concerned about Indoor Humidity?

Indoor Air Quality

Scientists agree that excess water or "dampness" can contribute significantly to mold growth inside buildings. An article in the November 2002 issue of the *ASHRAE Journal* notes that:

> While it has been difficult for epidemioligic studies to definitively link indoor mold and human illness, there are indications that indoor mold is responsible for such health concerns as nasal irritation, allergic and non-allergic rhinitis, malaise, and hypersensitivity pneumonitis.[1]

The Web site hosted by the U.S. Environmental Protection Agency (EPA) is a good source for information about indoor air quality and related health effects (www.epa.gov/iaq). ■

It is virtually impossible to avoid contact with the spores produced by fungi (including molds). Fungi exist everywhere: in the air, in and on plants and animals, on soil, and inside buildings. They extract the nutrients that they need to survive from almost any carbon-based material, including dust. Excessive indoor humidity, especially at surfaces, encourages fungi and other microorganisms, such as bacteria and dust mites, to colonize and grow.

Minimizing sources of moisture is the best way to help minimize microbial growth. Scientist/authors Sarah Armstrong and Jane Liaw recommend that:

> In the absence of clear guidance regarding what types of indoor fungi, or concentrations thereof in air, are safe or risky, one may wish simply to prevent mold from growing in buildings by acting quickly [drying water-damaged areas within 24 to 48 hours] when water leaks, spills, or floods occur indoors, being alert to condensation, and filtering air.

[1] S. Armstrong and J. Liaw. "The Fundamentals of Fungi," *ASHRAE Journal* 44 no. 11: 18–23.

Sources and Effects of Indoor Moisture

> If approved, a proposed addendum to Standard 62 would require that systems be designed to limit the relative humidity in occupied spaces to 65 percent or less at the design outdoor dew-point condition. The design dew-point condition, however, does *not* necessarily coincide with the worst-case condition for indoor relative humidity. As the examples presented later in this manual demonstrate, even higher indoor relative humidities can occur on mild, rainy days during the cooling season. The proposal was still under debate when this manual went to press. Check ASHRAE's Web site, www.ashrae.org, for more information. ■

ANSI/ASHRAE Standard 62–2001, *Ventilation for Acceptable Indoor Air Quality*, addresses the link between indoor moisture and microbial growth in this recommendation:

> Relative humidity in habitable spaces preferably should be maintained between 30 percent and 60 percent to minimize the growth of allergenic and pathogenic organisms. (Section 5.10)

The U.S. Environmental Protection Agency (EPA) adopts a similar stance in its publication titled *Mold Remediation in Schools and Commercial Buildings:*

> The key to mold control is moisture control. Solve moisture problems before they become mold problems! … [One way to help prevent mold is to] maintain low indoor humidity, below 60 percent relative humidity (ideally 30–50 percent, if possible).

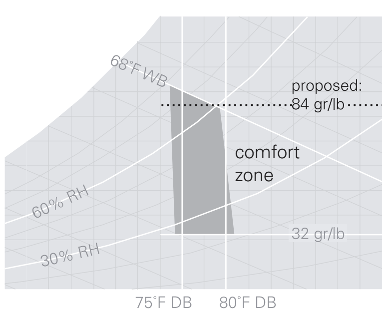
This publication, which was published in March 2001 and is identified as EPA 402-K-01-001, is available from www.epa.gov/iaq/molds. For more information about the mechanics of mold growth and how it affects buildings and HVAC systems, review Chapter 7 in *Humidity Control Design Guide for Commercial and Institutional Buildings* (ISBN 1-883413-98-2). It was published by ASHRAE in 2001, and is available from their online bookstore at www.ashrae.org.

Occupant Comfort and Productivity

In addition to curbing microbial growth, limiting indoor humidity to an acceptable level helps assure consistent thermal comfort within occupied spaces, which:

- Reduces occupant complaints
- Improves worker productivity
- Increases rental potential and market value

Figure 2. Summer "comfort zone" defined by ASHRAE Standard 55–1992



ANSI/ASHRAE Standard 55–1992, *Thermal Environmental Conditions for Human Occupancy*, specifies thermal environmental conditions that are acceptable to 80 percent or more of the occupants within a space. The "comfort zone" (Figure 2) defined by Standard 55 represents a range of environmental conditions based on dry-bulb temperature, humidity, thermal radiation, and air movement. Depending on the utility of the space, maintaining the relative humidity between 30 percent and 60 percent keeps most occupants comfortable.

Note: A proposed revision to ASHRAE Standard 55 suggests redefining the upper humidity limit for thermal comfort as a humidity ratio of 84 gr/lb (12 g/kg). This approximates a dew point of 62°F (16.7°C) or a relative humidity *of 65 percent when the dry-bulb temperature is 75°F (23.9°C). The proposal was still under debate when this manual went to press.*

Building Maintenance

For more information about problems resulting from moisture in buildings, refer to *Preventing Indoor Air Quality Problems in Hot, Humid Climates: Design and Construction Guidelines,* published by CH2M Hill, and to *Humidity Control Design Guide for Commercial and Institutional Buildings,* published by ASHRAE. ■

The same fungi (mold and mildew) that cause people discomfort and/or harm also can irreversibly damage building materials, structural components, and furnishings through premature failure, rot, corrosion, or other degeneration. Moisture-related deterioration affects maintenance costs and operating costs by increasing the frequency of normal cleaning and by requiring periodic replacement of damaged furnishings, such as moldy carpet and wallpaper.

Climate Considerations

The *ASHRAE Handbook—Fundamentals* is a popular source for tabular, climatic data representing the outdoor design conditions of many locations. **Peak dry-bulb conditions** for cooling systems appear under the heading "Cooling DB/MWB" (dry bulb and mean-coincident wet bulb). The ASHRAE weather tables also indicate how often each condition occurs. For example, the 0.4 percent, peak dry-bulb condition for Jacksonville, Florida, is 96°F DB and 76°F MWB (35.7°C DB, 24.5°C MWB). In other words, the outdoor dry-bulb temperature exceeds 96°F (35.7°C) for 0.4 percent of the time, or 35 hours, in an average year. Also, the average, coincident wet-bulb temperature at this dry bulb is 76°F (24.5°C WB).

The *sensible* load caused by the introduction of outdoor air and weather-dependent space loads, such as conduction, is greatest when the outdoor dry-bulb temperature is highest. Consequently, engineers who design HVAC systems typically and (most of the time) appropriately use the peak dry-bulb condition to determine the required capacity for the cooling coil. The peak *latent* load resulting from the introduction of outdoor air, however, does *not* coincide with the highest outdoor dry-bulb temperature; instead, it occurs when the dew point of the outdoor air is highest.

Beginning with the 1997 edition, the design weather data in the *ASHRAE Handbook—Fundamentals* includes the **peak dew-point condition** for each location. Although peak dew-point data is seldom used for design purposes, it helps designers analyze the dehumidification performance of HVAC systems and, at the same time, provides a more complete picture of the relevant weather conditions. According to the 2001 Handbook:

> The [extreme dew-point] values are used as a check point when analyzing the behavior of cooling systems at part-load conditions, particularly when such systems are used for humidity control as a secondary (or indirect) function. (p. 27.3)

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Peak dew-point design conditions for cooling systems appear under the heading "Dehumidification DP/MDB and HR" (dew point/mean-coincident dry bulb and humidity ratio). The 0.4 percent, peak dew-point condition for Jacksonville, Florida, is 76°F DP and 84°F MDB (24.6°C DP, 28.8°C MDB). Outdoor air is cooler at this condition, but contains more moisture than outdoor air at the peak dry-bulb condition.

For outdoor air used for ventilation, the peak sensible load rarely coincides with the peak latent load. Consequently, coils *selected* for the highest sensible load may not provide sufficient latent capacity when the highest latent load occurs. More often, however, coils *controlled to maintain the dry-bulb temperature* in the space (sensible capacity) operate with inadequate latent capacity at part-load conditions, even though the latent capacity may be available. Therefore, it is important to evaluate system performance at full-load *and part-load* conditions, based on the humidity-control requirements of the application.

Moisture problems aren't confined to hot, humid climates. Too often, indoor humidity problems are incorrectly associated *only* with buildings located in hot, humid climates. While it is true that such areas experience elevated outdoor humidity levels for a higher percentage of the year, the absolute amount of moisture in the air is comparable to that experienced in many other climates. To illustrate this fact, Table 1 shows the peak dry-bulb and peak dew-point conditions for several cities across the United States. Although these cities are located in different regions, the peak dew-point conditions for most of these locations are remarkably similar.

Table 1. Cooling design conditions for various U.S. cities[1]

	0.4% Peak dry-bulb condition		0.4% Peak dew-point condition	
Baltimore, Maryland	93°F DB	(34.0°C)	75°F DP	(23.8°C)
	75°F WB	(23.7°C)	83°F DB	(28.1°C)
Dallas, Texas	100°F DB	(37.8°C)	75°F DP	(23.7°C)
	74°F WB	(23.6°C)	82°F DB	(28.0°C)
Denver, Colorado	93°F DB	(33.8°C)	60°F DP	(15.6°C)
	60°F WB	(15.3°C)	69°F DB	(20.4°C)
Jacksonville, Florida	96°F DB	(35.7°C)	76°F DP	(24.6°C)
	76°F WB	(24.5°C)	84°F DB	(28.8°C)
Los Angeles, California	85°F DB	(29.2°C)	67°F DP	(19.4°C)
	64°F WB	(17.7°C)	75°F DB	(23.6°C)
Minneapolis, Minnesota	91°F DB	(32.8°C)	73°F DP	(22.5°C)
	73°F WB	(22.7°C)	83°F DB	(28.5°C)
San Francisco, California	83°F DB	(28.4°C)	59°F DP	(15.2°C)
	63°F WB	(17.0°C)	67°F DB	(19.4°C)

[1] Source: *2001 ASHRAE Handbook–Fundamentals,* Chapter 27 (Table 1B)

Sources and Effects of Indoor Moisture

It is important to understand that indoor humidity problems are not solely attributable to outdoor air brought into the building for ventilation, however. Indoor humidity levels typically depend as much on the sensible and latent loads in the space (and the resulting space sensible heat ratio), the type of HVAC system, and the method of controlling that system as they do on outdoor conditions. Moisture-related problems therefore can occur in *any* geographic region where buildings are mechanically ventilated and cooled.

Energy Use

For more information on Standard 90.1 and its effect on the design of HVAC systems, see the Trane *Engineers Newsletter* titled “90.1 Ways to Save Energy” (ENEWS-30/1). This newsletter is available at www.trane.com.

Standard 90.1 is available from ASHRAE’s online bookstore at www.ashrae.org. A user’s guide accompanies the standard. ■

Heightened concern about the cost and availability of energy is hastening the obsolescence of HVAC systems that reheat cold supply air using “new energy” or that mix hot and cold air streams to achieve the desired space temperature.

In the United States, the primary standard related to energy consumption in commercial buildings is ANSI/ASHRAE/IESNA Standard 90.1–2001, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. It provides minimum requirements for energy-efficient building design, including the building envelope, lighting system, motors, HVAC system, and service-water heating system.

Some people believe that the requirements of Standard 90.1 make it impossible to maintain indoor humidity within the ranges recommended by Standard 62 and the U.S. EPA (p. 4). Section 6.3.2.1 and Section 6.3.2.3 of Standard 90.1 restrict the use of “new energy” for reheat and limit mixing of hot and cold air streams; the intent is to restrict dehumidification systems and control strategies that waste energy.

Section 6.3.2.3, (excerpted on the next page) is particularly relevant because it specifically addresses HVAC systems that regulate indoor humidity. We address its implications throughout this manual, and describe system designs and control strategies that comply with Standard 90.1 while properly regulating indoor humidity. ■

Sources and Effects of Indoor Moisture

from ANSI/ASHRAE/IESNA Standard 90.1–2001 on Dehumidification

6.3.2.3 Dehumidification. Where humidistatic controls are provided, such controls shall prevent reheating, mixing of hot and cold airstreams, or other means of simultaneous heating and cooling of the same airstream.

Exceptions to 6.3.2.3:

a) The system is capable of reducing supply air volume to 50% or less of the design airflow rate or the minimum rate specified in 6.1.3 of ASHRAE Standard 62, whichever is larger, before simultaneous heating and cooling takes place.

b) The individual fan cooling unit has a design cooling capacity of 80,000 Btu/h (23 kW) or less and is capable of unloading to 50% capacity before simultaneous heating and cooling takes place.

c) The individual mechanical cooling unit has a design cooling capacity of 40,000 Btu/h (12 kW) or less. An individual mechanical cooling unit is a single system composed of a fan or fans and a cooling coil capable of providing mechanical cooling.

d) Systems serving spaces where specific humidity levels are required to satisfy process needs, such as computer rooms, museums, surgical suites, and buildings with refrigerating systems, such as supermarkets, refrigerated warehouses, and ice arenas. This exception also applies to other applications for which fan volume controls listed in accordance with Exception (a) are proven to be impractical to the enforcement agency.

e) At least 75% of the energy for reheating or for providing warm air in mixing systems is provided from a *site-recovered* (including condenser heat) or *site solar energy source*.

f) Systems where the heat added to the airstream is the result of the use of a desiccant system and 75% of the heat added by the desiccant system is removed by a heat exchanger, either before or after the desiccant system with energy recovery. ■

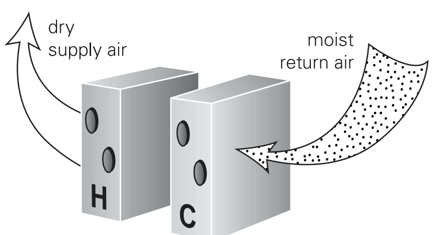
Dehumidification Primer

Types of Dehumidification

Maintaining the indoor humidity within the desired range requires a means of either locally removing moisture from the air that is already in the space, or replacing that moisture-laden air with drier air that was dehumidified elsewhere.

Local Dehumidification

Figure 3. Local dehumidification



Portable dehumidifiers, like those used in many residential basements, provide dedicated, local dehumidification. These devices (Figure 3) commonly use mechanical refrigeration to remove moisture from the air in the space: an evaporator coil dehumidifies and *coincidentally* cools the entering air, and a condenser coil reheats the leaving air. Humidity in the space decreases, while the dry-bulb temperature increases.

Simple, in-space air conditioners often coincidentally dehumidify the space as they cool it; but do not confuse these devices with dedicated dehumidification equipment. The evaporator coil in a packaged terminal air conditioner ("PTAC," Figure 4) responds to the room thermostat, directly cooling a mixture of recirculated return air and outdoor air, and removing moisture in the process. At full load, the air conditioner usually provides adequate dehumidification because the thermostat keeps the unit running and the coil cold.

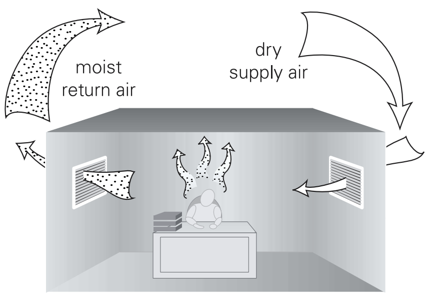
To avoid overcooling the space at part load, however, the thermostat reduces the sensible-cooling capacity of the coil by cycling it on and off. Cycling raises the average temperature of the coil, which significantly reduces its dehumidification (latent-cooling) capacity. Simple air conditioners, such as the PTAC, may provide adequate coincidental dehumidification for spaces with constant cooling loads. When the cooling load varies widely, however, additional equipment and/or controls may be required for adequate dehumidification at part-load conditions.

Figure 4. Packaged terminal air conditioner

Dehumidification Primer

Remote Dehumidification

Figure 5. Remote dehumidification



The central air-conditioning system commonly serves as a remote source of dehumidification for the occupied spaces in a commercial or industrial building. To maintain an acceptable indoor humidity, the system must be properly designed and controlled so that the air it supplies is drier than the air in the space (Figure 5). In effect, the supply air must be dry enough to “soak up” the water vapor in the space; the absorbed moisture is then carried from the space in the return air.

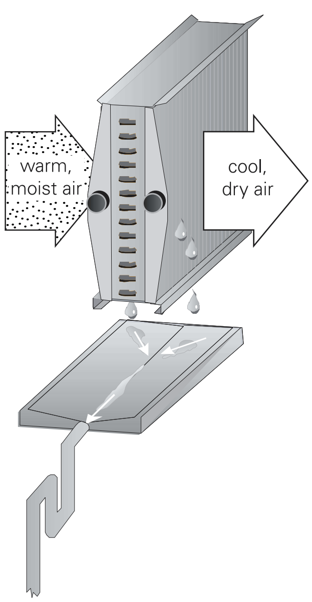
Depending on the type of system and method of control, central air-conditioning units may or may not be able to adequately dehumidify the space at all load conditions. The dehumidification performance of various system types and control methods is discussed in the next three chapters.

Processes for Dehumidification

An air-conditioning system typically uses one of two processes to dehumidify the supply air that ultimately reaches the space: **condensation** on a cold coil or **adsorption** via a desiccant.

Condensation on a Cold Coil

Figure 6. “Cold-coil” dehumidification



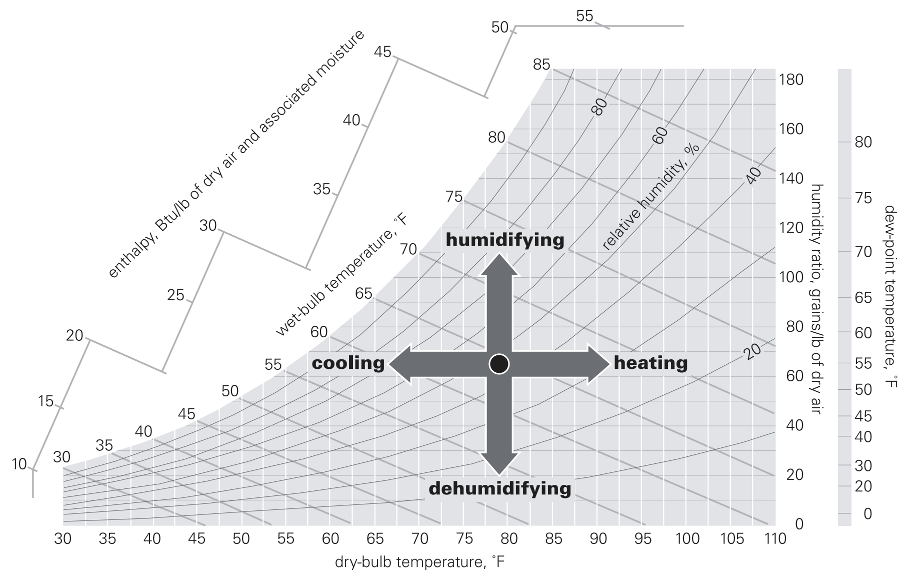
Water vapor condenses on a surface if the temperature of the surface is colder than the dew point of the moist air in contact with it. Controlled condensation dehumidifies an air stream by directing it across the cold surfaces of a finned-tube coil. Circulating either chilled water or evaporating refrigerant through the coil makes the coil surfaces cold enough to induce condensation. As warm, moist air passes through the coil, water vapor condenses on the cold surfaces (Figure 6); the condensate (liquid water) then drains down the coil fins and collects in the drain pan, where it is piped from the air handler. The air leaves the coil cooler and drier.

A psychrometric chart can illustrate how “cold-coil” dehumidification works. This special-purpose chart (Figure 7) represents the interrelated physical properties of moist air: dry-bulb (DB), wet-bulb (WB), and dew-point (DP) temperatures; relative humidity (RH), enthalpy (h), and humidity ratio (W). For example, if sensible heat is added or removed with no change in moisture content, the condition of the air moves horizontally on the chart. Conversely, if moisture is added or removed without changing the dry-bulb temperature, then the condition of the air moves vertically on the chart.

Figure 8 (p. 12) illustrates what happens when a mixture of outdoor air and recirculated return air at 80°F DB, 60°F DP (26.7°C DB, 15.6°C DP), enters a cold coil. The temperature of the coil surface is well below the dew point of the

Dehumidification Primer

Figure 7. Psychrometric chart



entering air. Sensible cooling occurs as the air passes through the coil; on the chart, the air condition moves horizontally to the left. When the condition of the air nears the saturated state (100 percent-relative humidity), moisture begins to condense on the cold surface of the coil. The condition of the air now moves diagonally down and to the left on the chart, representing the removal of both sensible heat and moisture. Cool, dry air leaves the coil in this example at 55°F DB, 53°F DP (12.8°C DB, 11.7°C DP).

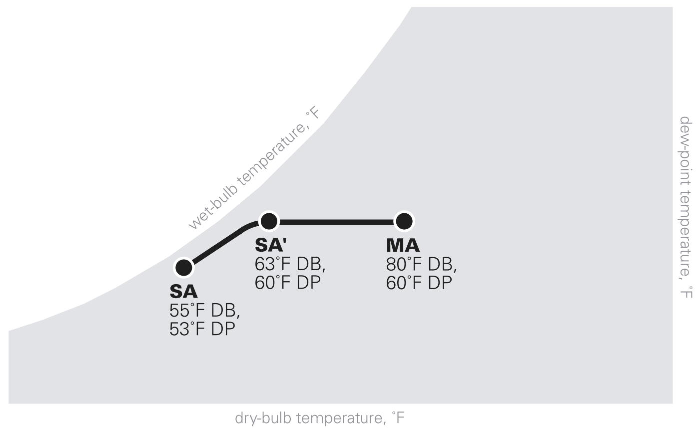
The Trane psychrometric chart includes a series of "coil curves" that depict the approximate performance of a wide range of coil configurations (Figure 19, p. 26). These curved lines, established from hundreds of laboratory tests of various coil geometries at different air and coolant temperatures, represent the changes in dry-bulb and dew-point temperatures as air passes through a "typical" cooling coil. Of course, exact coil performance depends on actual coil geometry and can be precisely determined by software that accurately models the performance of the specific coils. ■

No moisture removal occurs unless the temperature of the coil surface is lowered below the dew point of the entering air. If the coil surface is *not* colder than the dew point, only sensible cooling takes place. Sensible cooling without dehumidification is especially common during part-load operation of a constant-volume system. That's because constant-volume systems (discussed in the next chapter) respond to part-load conditions by reducing coil capacity, which raises the temperature of the coil surface and of the supply air.

For comfort-cooling applications that do not require a supply-air dew point lower than 40°F to 45°F (4.5°C to 7°C), cold-coil condensation is the traditional choice for dehumidification because of its low first cost and low operating cost. Given that decision, the next choice is whether to use chilled water or refrigerant to make the coil cold.

Dehumidification Primer

Figure 8. Psychrometric analysis of “cold-coil” dehumidification



Chilled water systems, with their individually selected components, provide the necessary design flexibility for applications that require low supply-air dew points, that is, dew points approaching 40°F to 45°F (4.5°C to 7°C).

By contrast, most DX systems are packaged. Although prematched refrigeration and air-handling components lower the initial cost of the system, they also make the system less flexible by deferring certain design decisions to the manufacturer. A traditional, “off-the-shelf” packaged DX system is optimized for operation at about 400 cfm/ton (0.054 m^3/s/kW), which prevents it from achieving “low” dew points. Specially designed DX equipment can reach dew points of 45°F to 50°F (7°C to 10°C) because they are designed to deliver less airflow (cfm) per cooling ton (L/s per kW).

When space loads or process requirements dictate an even lower supply-air dew point, moisture adsorption is preferred for dehumidification.

Condensate management

Managing Building Moisture, Trane applications engineering manual SYS-AM-15, discusses proper design and installation of condensate traps for draw-through and blow-through coil configurations. ■

When a cold coil is used for dehumidification, moisture condenses from the air onto the surface of the coil and falls into the drain pan, where it is piped from the air handler. Too often, inattention to proper trapping of the condensate line causes “spitting,” which dampens the insulation inside the air handler and ductwork, or restricts flow from the drain pan, causing it to overflow. Both situations create opportunities for microbial growth. To assure proper condensate removal under all operating conditions, comply with the manufacturer’s instructions for drain-line installation and trapping.

Adsorption Using a Desiccant

Solid desiccants are typically used for dehumidification equipment applied in commercial and institutional buildings. Liquid desiccants are also available, but they are traditionally used in industrial applications. Refer to the "Desiccant Dehumidification and Pressure-Drying Equipment" chapter of the *ASHRAE Handbook–HVAC Systems and Equipment* for more information. ■

Desiccants used for commercial dehumidification are selected for their ability to collect large quantities of water vapor. The porous surface of the desiccant attracts and retains water molecules from the passing air stream. This dehumidification process is described as **adsorption** because the collected moisture does not chemically or physically alter the desiccant.

Vapor pressure at the desiccant surface is directly proportional to the surface temperature of the desiccant and the amount of moisture adsorbed there. When the desiccant is cool and dry, its surface vapor pressure is low; when the desiccant is warm and moist, its surface vapor pressure is high. Water vapor migrates from areas of high vapor pressure to areas of low vapor pressure. Consequently, a desiccant with a low surface vapor pressure will adsorb water molecules from the surrounding air, while a desiccant with a high surface vapor pressure will reject water molecules to the surrounding air.

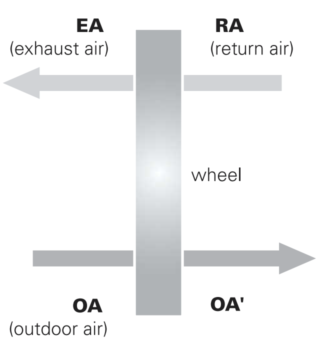
The most common application of adsorption for commercial dehumidification uses a rotating wheel that contains a fluted, desiccant-coated medium. The wheel rotates between two air streams: the "process" air stream and the "regeneration" air stream. Warm, moist *process* air enters one side of the rotating wheel, where water vapor collects on the desiccant surface. As the wheel rotates, the moisture-laden portion moves into the *regeneration* air stream, where the collected water vapor is released and transported outdoors. The cycle repeats with each rotation, providing continuous dehumidification.

The temperature of the regeneration air determines whether the adsorption process is *passive* or *active*.

Passive adsorption

When the regeneration air is drier than the process air, but is not heated to drive the moisture from the desiccant, the dehumidification process is considered *passive* adsorption.

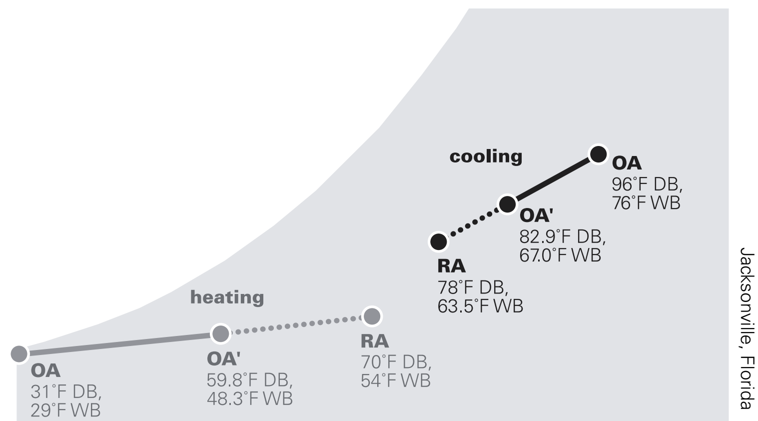
Figure 9. Total-energy wheel



An example of passive adsorption is the use of building exhaust air to regenerate the desiccant of a total-energy/enthalpy wheel (Figure 9). The wheel is mounted so that the minimum outdoor (process) airflow required for ventilation passes through half of the wheel, while exhaust (regeneration) air passes through the other half. The wheel rotates quickly — between 20 rpm and 60 rpm — alternately exposing the desiccant to process air and regeneration air.

In the summer, when the outdoor air is hot and humid, the total-energy wheel cools and dehumidifies the entering outdoor air by transferring sensible heat and moisture to the cooler, drier exhaust air (Figure 10, p. 14). Desiccant regeneration occurs at a low temperature — 78°F (25.6°C) in this example — without additional heat. In the winter, when the outdoor air is cold and dry, the

Figure 10. Example of passive adsorption performed by a total-energy wheel



total-energy wheel warms and humidifies the entering outdoor air by transferring sensible heat and moisture from the warmer, moister exhaust air.

Refer to *Air-to-Air Energy Recovery in HVAC Systems*, Trane applications engineering manual SYS-APM003-EN, for more information about using the passive adsorption of total-energy wheels to precondition outdoor air. ■

Although desiccant-coated devices, such as the total-energy wheel, reduce the sensible heat and moisture content of entering outdoor air, these *passive adsorption devices are not considered as dehumidification equipment*. Such devices are less than 100 percent effective: When it is humid outside, process air leaving the wheel always contains more moisture than regeneration air (from the space) entering the exhaust side of the wheel. By definition, a passive adsorption device cannot dehumidify the space because the air leaving the supply side of the device never can be drier than the space. As demonstrated in "Dehumidifying with Constant-Volume Mixed Air" (pp. 27–29), a space under these conditions will always require additional dehumidification.

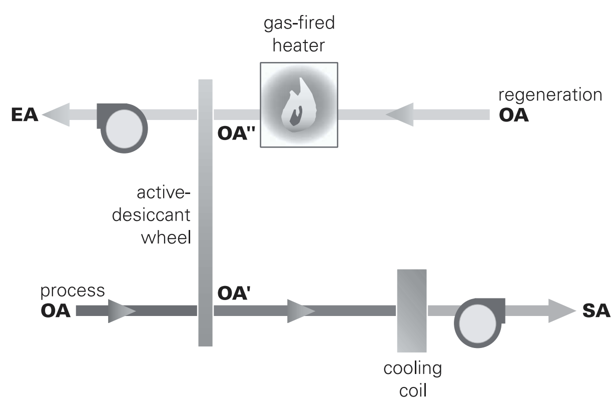
Active adsorption

In the *active* adsorption process, the moisture-collecting ability of the desiccant is improved by adding sensible heat to the regeneration air before it enters the desiccant. Figure 11 depicts the active desiccant wheel mounted so that the outdoor (process) air for ventilation passes through half of the wheel, while regeneration air (either a separate outdoor air stream or exhaust air from the building) passes through the other half.

As the active desiccant wheel slowly rotates between 10 rph and 30 rph, it removes moisture from the outdoor (process) air stream *and releases sensible heat* (Figure 12). The resulting temperature increase is directly proportional to the amount of moisture removed from the process air. In this example, active adsorption dehumidifies the process air to 44°F DP (6.7°C DP) and raises the temperature of the process air to 120°F DB (48.9°C DB). Consequently, the process air must be cooled before it is delivered to the building's occupied

Dehumidification Primer

Figure 11. Active adsorption system

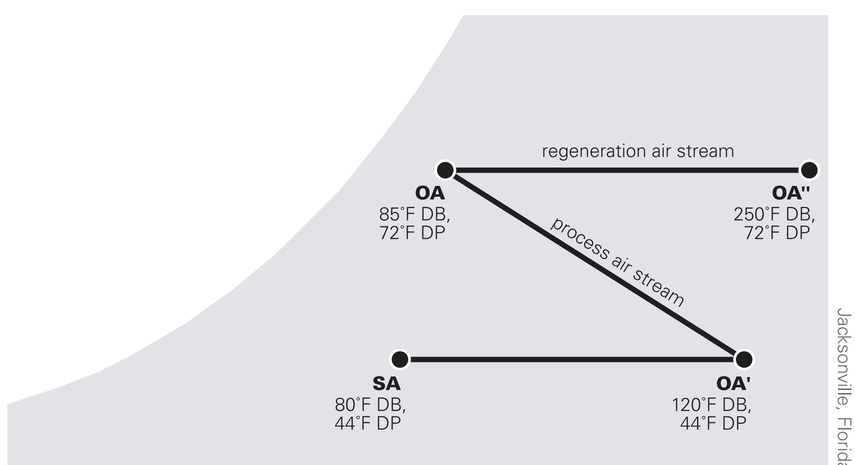


spaces. The psychrometric analysis (Figure 12) for this example system shows that the cooling coil lowers the temperature of the process air to 80°F DB (26.7°C DB).

On the regeneration side of the system, a gas-fired heater raises the temperature of the regeneration air. Depending on the dew-point target for the process air, regeneration air temperatures typically range from 130°F to 250°F (54°C to 121°C). The warmer that the regeneration air is, the drier the resulting process air will be.

Recall that Section 6.3.2.3 of ASHRAE Standard 90.1 requires that humidistatic controls prevent simultaneous heating and cooling of the same air stream. It therefore addresses active-adsorption dehumidification, which heats the process air and requires downstream cooling. Exception F of Section 6.3.2.3

Figure 12. Example performance for an active adsorption system



(p. 8 in this manual) defines the conditions for compliance; that is, an active desiccant system must recover 75 percent of the heat that adsorption adds to the process air.

Sensible heat added by the adsorption process:

$$Q_s = 1.085 \times 2{,}634\ cfm \times (120°F - 85°F)$$
$$= 100{,}000\ Btu/hr$$

$$(Q_s = 1.21 \times 1.24\ m^3/s \times [48.9°C - 29.4°C])$$
$$(= 29.3\ kW)$$

For example, if the adsorption process adds 100,000 Btu/hr (29.3 kW) of sensible heat to the process air, then 75,000 Btu/hr (22.0 kW) of energy must be removed from that same air. One possible design solution places a sensible-energy, air-to-air heat exchanger downstream of the active desiccant wheel to transfer at least 75,000 Btu/hr (22.0 kW) of heat from the hot, dry process air to the regeneration air. Another possible solution adds an air-to-air energy-recovery device, such as a total-energy wheel, upstream of the active desiccant wheel to precondition the outdoor air and transfer at least 75,000 Btu/hr (22.0 kW) of heat (sensible plus latent energy) from the process air to another air stream.

Typical applications for adsorption dehumidification

Total-energy wheels and other types of passive adsorption devices are used in all types of HVAC systems to *precondition* outdoor air. This practice enables downsizing of cooling and heating equipment, which reduces the initial cost of the system; it also saves energy by reducing the cooling and heating loads associated with ventilation.

Active adsorption systems are primarily used in applications where high internal latent loads or process requirements dictate a lower-than-normal dew point (below a threshold of 40°F to 45°F [4.5°C to 7°C]) for the supply air. Typical applications include supermarkets, ice rinks, museums, industrial drying processes, and other spaces that require exceptionally dry air. Given the relatively high first cost, the energy required to heat the regeneration air, and the additional energy needed to post-cool the process air, active adsorption systems are seldom used in comfort-cooling applications. The succeeding chapters of this manual therefore focus exclusively on comfort-cooling systems that use “cold coil” condensation for dehumidification.

Dehumidification Primer

Implications for HVAC Control

Even when the average relative humidity in a conditioned space is low, high relative humidities can develop near cold surfaces and increase the likelihood of condensation. Enforcing a maximum relative humidity of 60 percent or 65 percent should make most surfaces 12°F to 15°F (6.7°C to 8.3°C) warmer than the space dew point and generally avoid concentrations of water vapor near surfaces. ■

The next three chapters examine three types of HVAC systems, which are distinguished from one another by how each system delivers ventilation air to the space: constant-volume mixed air, variable-volume mixed air, and dedicated outdoor air. In each case, the central theme is "cold coil" dehumidification during full-load and part-load comfort cooling. The performance benchmark is a relative humidity of 60 percent, which is the upper limit currently recommended by ASHRAE Standard 62.

Certain control strategies will affect the dehumidification performance of *any* of these HVAC systems:

- Humidity control during unoccupied periods
- Building pressurization
- Airside economizing

Brief descriptions of how each of these control strategies affects dehumidification performance follow. Specific application considerations by system type are discussed within the appropriate chapter.

Humidity Control during Unoccupied Periods

Latent loads associated with occupants and their activities make humidity control important during scheduled operation. But after-hours humidity control is also important in facilities, such as schools, with few or no occupants for extended periods. ASHRAE offers the following recommendation:

> In humid climates, serious consideration should be given to dehumidification during the summer months, when the school is unoccupied, to prevent the growth of mold and mildew. (*1999 ASHRAE Handbook–Applications,* Chapter 6, p. 6.3)

Controlling humidity at *all* times of the day can greatly reduce the risk of microbial growth on building surfaces and furnishings. Wet-cleaning procedures (mopping floors, shampooing carpets) bring large amounts of moisture into the building and usually take place when the building is unoccupied. Drying wet surfaces is critical to prevent microbial growth. For shampooed carpets, this is best accomplished by providing adequate air motion and dehumidification during unoccupied hours.

Dehumidification Primer

Building Pressurization

Refer to *Building Pressurization Control,* Trane applications engineering manual AM-CON-17, for additional information about how to regulate building pressure through design and control of the HVAC system. ■

HVAC systems do more than provide heating, cooling, and ventilation; they also bring makeup air into the building to replace the air removed by local exhaust fans (in restrooms and kitchens, for example) and combustion equipment (furnaces, fireplaces). Turning off the ventilation system during unoccupied periods while allowing these devices to continue operating creates negative pressure inside the building. Unconditioned outdoor air infiltrates the building, which can raise the dew point in the envelope (risking condensation) and increase the humidity in the occupied space (perhaps beyond the limit recommended by ASHRAE).

One solution is to design the building control system so that it turns off all local exhaust fans and combustion equipment whenever the ventilation system is off. However, this approach may require a manual override to accommodate after-hours cleaning.

Wind, variable operation of local exhaust fans, and “stack effect” in multistory buildings can create building pressure fluctuations despite a properly balanced HVAC system. Therefore, controlling building pressure directly may be desirable to prevent negative pressure from developing inside the building… and it may be *necessary* during economizer operation to prevent overpressurization.

Airside Economizing

An airside economizer can lower operating costs by using outdoor air to help offset building cooling loads. When outdoor conditions are suitable for natural cooling, the outdoor-air damper opens fully, assisting the mechanical cooling equipment by offsetting as much of the cooling load as possible. At cooler outdoor conditions, the outdoor-air damper maintains the target temperature in the space by modulating between its full-open and minimum-open positions.

When the outdoor air is too warm or too cold for economizing, the outdoor-air damper remains at the minimum-open position to provide the necessary quantity of outdoor air for ventilation; meanwhile, the cooling or heating coil satisfies the space load.

Proper control of the airside economizer is critical to maximize energy savings without creating potential humidity problems. ■

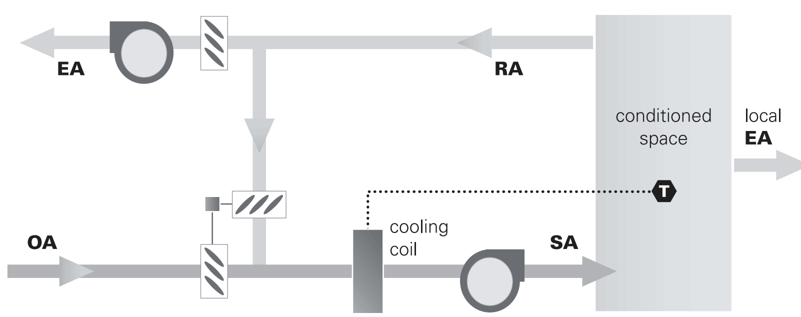
Dehumidifying with Constant-Volume Mixed Air

Mixed-air systems use an air handler to condition a combination of outdoor air and recirculated return air before delivering this mixed air to each space. A *constant-volume,* mixed-air system supplies an unchanging quantity of air, usually to a single space or thermal zone. The temperature of the supply air modulates in response to the varying sensible-cooling load in the space.

"Basic" constant-volume systems, which consist of an air handler containing a fan and a cold coil (Figure 13), *indirectly* affect indoor humidity. A thermostat compares the dry-bulb temperature in the space to the setpoint; it then modulates the cooling coil until the cooling capacity matches the sensible load — that is, until the space temperature and setpoint match. Reducing the capacity of the cooling coil results in a warmer coil surface and less dehumidification. Similarly, increasing the coil capacity makes the coil surface colder and provides more dehumidification.

The peak *sensible* load on the cooling coil rarely coincides with the peak *latent* load. So, a cooling coil selected for the highest sensible load (in some air-handling arrangements) may not provide sufficient capacity when the highest latent load occurs. More often, however, a cooling coil that is *controlled* to maintain the space dry-bulb temperature often operates without adequate moisture-removal capacity at peak latent-load conditions. As the following examples reveal, accurate predictions of dehumidification performance require an analysis of system operation at both full-load *and* part-load conditions.

Figure 13. Basic, constant-volume HVAC system



Analysis of Dehumidification Performance

$$V_{sa} = \frac{10{,}000\ cfm \times 9\ air\ changes/hr}{60\ min/hr} = 1{,}500\ cfm$$

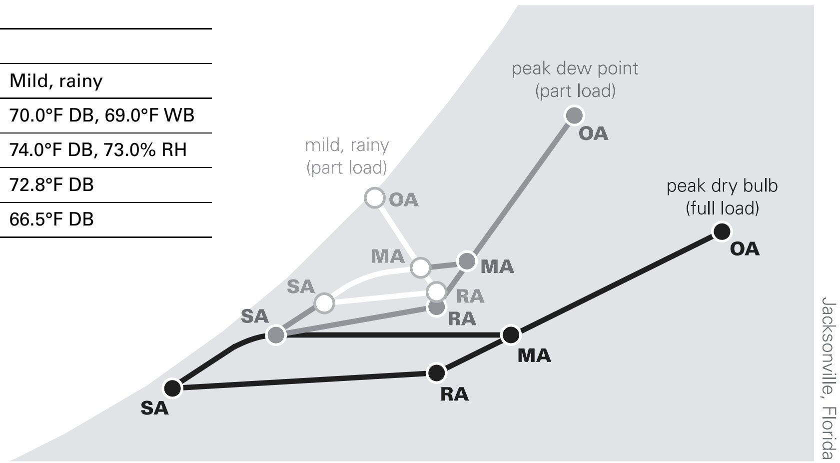
$$\left(V_{sa} = \frac{283\ m^3 \times 9\ air\ changes/hr}{3{,}600\ sec/hr}\right)$$
$$(= 0.7\ m^3/s)$$

Consider a 10,000 ft³ (283 m³), 30-occupant classroom in Jacksonville, Florida. For thermal comfort, the space setpoint is 74°F DB (23.3°C DB). Supply airflow V_{sa} is based on nine air changes per hour and is 1,500 cfm (0.7 m³/s). ASHRAE Standard 62 requires 15 cfm (8 L/s) of outdoor air per person for adequate ventilation; so, 450 cfm (0.21 m³/s) of the supply air must be outdoor air.

Dehumidifying with Constant-Volume Mixed Air

Figure 14. Dehumidification performance of a basic, constant-volume HVAC system at various outdoor conditions

Design condition	Full load	Part load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96.0°F DB, 76.0°F WB	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 52.4% RH	74.0°F DB, 67.0% RH	74.0°F DB, 73.0% RH
MA	80.6°F DB	77.0°F DB	72.8°F DB
SA	55.7°F DB	63.0°F DB	66.5°F DB



The classroom is air conditioned by a basic constant-volume system, which uses a chilled water coil to cool and dehumidify the supply air. A modulating valve controls coil capacity.

At the peak dry-bulb condition:

$$SHR = \frac{29{,}750\ Btu/hr}{29{,}750\ Btu/hr + 5{,}250\ Btu/hr} = 0.85$$

$$\left(SHR = \frac{8.7\ kW}{8.7\ kW + 1.5\ kW}\right) (= 0.85)$$

Performance at peak dry-bulb (full-load) condition. According to the 2001 *ASHRAE Handbook—Fundamentals*, the peak dry-bulb condition for Jacksonville is 96°F DB, 76°F WB (35.7°C DB, 24.5°C WB). At this condition, the sensible and latent loads calculated for the classroom — 29,750 Btu/hr (8.7 kW) and 5,250 Btu/hr (1.5 kW), respectively — yield a sensible-heat ratio (SHR) of 0.85 in that space. Given the supply airflow of 1,500 cfm (0.7 m³/s), satisfying the sensible-cooling load and maintaining the space at 74°F DB (23.3°C DB) requires 55.7°F (13.1°C) supply air.

$$Q_s = 1.085 \times 1{,}500\ cfm \times (74°F - T_{supply}) = 29{,}750\ Btu/hr \therefore T_{supply} = 55.7°F$$

$$(Q_s = 1.21 \times 0.7\ m^3/s \times [23.3°C - T_{supply}]) \ (= 8.7\ kW \therefore T_{supply} = 13.1°C)$$

Figure 14 summarizes the psychrometric analysis of this system's full-load dehumidification performance. At the peak dry-bulb condition, controlling the temperature in the space to 74°F (23.3°C) will result in a comfortable relative humidity of 52 percent.

Note: To simplify the analysis, which is detailed in Appendix A of this manual, the latent load in the classroom is limited to moisture generated by the occupants. A higher relative humidity would result if other sources of indoor moisture, such as infiltration and vapor-pressure diffusion, were considered. The cooling coil is expected to offset the "non-space" latent load that results from ventilating the classroom with outdoor air.

Dehumidifying with Constant-Volume Mixed Air

$$Q_T = 4.5 \times 1{,}500\ cfm \times (31.4 - 22.9\ Btu/lb)$$
$$= 57{,}375\ Btu/hr = 4.78\ tons$$

$$(Q_T = 1.2 \times 0.7\ m^3/s \times [73.1 - 53.1\ kJ/kg])$$
$$(= 16.8\ kW)$$

The total capacity required from the cooling coil at the peak dry-bulb condition is 4.78 tons (16.8 kW).

At full load, the cooling coil removes both sensible heat and moisture (latent heat), directly controlling space temperature and indirectly affecting space humidity.

The peak dew-point condition does not necessarily represent the worst-case condition for humidity control. It simply is an easy "test case" for analyzing part-load dehumidification performance. ■

Performance at peak dew-point (part-load) condition. As the sensible-cooling load in the space decreases, a constant-volume HVAC system allows the supply-air temperature to rise by reducing the capacity of the cooling coil. In this example system, coil capacity is reduced by modulating the water valve. Although this control action successfully maintains the desired dry-bulb temperature for the space, raising the supply-air temperature also reduces the amount of moisture that condenses on the coil; space humidity rises. In other words, making the coil surface warmer decreases the rate at which moisture condenses from the mixed air.

At the peak dew-point condition:

$$Q_s = 1.085 \times 1{,}500\ cfm \times (74°F - T_{supply})$$
$$= 17{,}850\ Btu/hr \therefore T_{supply} = 63°F$$

$$(Q_s = 1.21 \times 0.7\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 5.2\ kW \therefore T_{supply} = 17.2°C)$$

To determine whether a system will provide adequate dehumidification at part load, analyze performance at the peak dew-point condition. For our Jacksonville classroom, the peak dew-point condition is 76°F DP, 84°F DB (24.6°C DP, 28.8°C DB). The cooler outdoor dry-bulb temperature and correspondingly lower solar and conducted heat gains reduce the sensible load in the classroom to 17,850 Btu/hr (5.2 kW). Because the classroom's latent load remains unchanged at 5,250 Btu/hr (1.5 kW), however, the sensible-heat ratio (SHR) for the space drops to 0.77. Consequently, the 1,500 cfm (0.7 m^3/s) of supply air must be delivered at a higher temperature, 63°F (17.2°C) in this case, to avoid overcooling the space.

Warmer supply air, combined with the lower space SHR, raises the relative humidity in the classroom from 52 percent to 67 percent (Figure 14) — well above the 60 percent limit that ASHRAE recommends. Although the cooling coil *could* provide additional cooling (up to 4.78 tons [16.8 kW] if sized for the design dry-bulb condition), the thermostat reduces coil capacity to 3.66 tons (12.9 kW). This control action maintains the dry-bulb temperature in the classroom at setpoint, but space humidity rises. Oversizing the cooling coil will not prevent the shortfall in latent capacity if system control is based solely on the dry-bulb temperature in the space.

On a mild, rainy day:

$$Q_s = 1.085 \times 1{,}500\ cfm \times (74°F - T_{supply})$$
$$= 12{,}250\ Btu/hr \therefore T_{supply} = 66.5°F$$

$$(Q_s = 1.21 \times 0.7\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 3.6\ kW \therefore T_{supply} = 19.2°C)$$

Performance on a mild, rainy day (part-load condition). Although the peak dew-point condition is helpful for analyzing the part-load dehumidification performance of an HVAC system, do *not* assume that it represents the worst-case condition for space humidity control. Most of the time, the humidity in the space depends more on the space SHR and the system control strategy than on outdoor conditions.

Consider our example Jacksonville classroom on a mild, rainy day. At 70°F DB, 69°F WB (21.2°C DB, 20.6°C WB), the sensible load in the classroom drops further — this time to 12,250 Btu/hr (3.6 kW). Given an unchanged latent load of 5,250 Btu/hr (1.5 kW) due to occupants, the SHR in the classroom drops to 0.70. To prevent overcooling, the thermostat reduces the cooling coil capacity to 1.63 tons (5.74 kW) so that the 1,500 cfm (0.7 m³/s) of supply air is delivered to the classroom at 66.5°F (19.2°C). *How is humidity in the classroom affected?* Relative humidity climbs to 73 percent!

Application Considerations

Ventilation

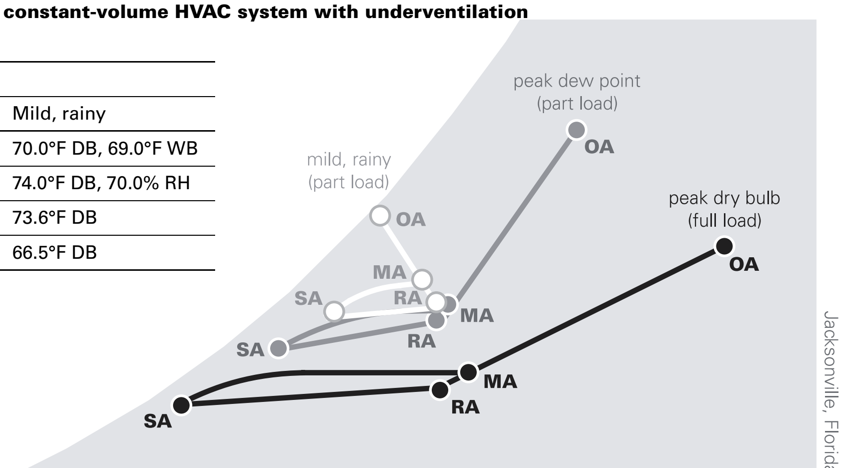
The 1989 revision of ASHRAE Standard 62 increased the required *per-person* ventilation rate from 5 cfm to 20 cfm (from 3 L/s to 10 L/s) for office buildings, and from 5 cfm to 15 cfm (from 3 L/s to 8 L/s) for schools. Bringing more outdoor air into the building to satisfy ventilation requirements significantly increases the cooling and heating loads on the HVAC system. But, does bringing more outdoor air into the building for ventilation cause moisture-related IAQ problems? Some people think so. Let's examine what happens if the example classroom receives only 150 cfm (0.07 m³/s) of outdoor air for ventilation rather than the 450 cfm (0.21 m³/s) that ASHRAE Standard 62 requires. Only the ventilation load differs from the previous examples (Figure 14, p. 20); the sensible- and latent-cooling loads for the classroom are unchanged, as are the supply-air temperatures and sensible-heat ratios (SHRs) for that space.

Figure 15 illustrates the effect of underventilating the classroom:

- *At the peak dry-bulb condition,* the relative humidity drops from 52 percent to approximately 50 percent.

Figure 15. Dehumidification performance of a basic, constant-volume HVAC system with underventilation

Design condition	Full load	Part load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96.0°F DB, 76.0°F WB	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 50.4% RH	74.0°F DB, 64.6% RH	74.0°F DB, 70.0% RH
MA	76.2°F DB	75.0°F DB	73.6°F DB
SA	55.7°F DB	63.0°F DB	66.5°F DB



Dehumidifying with Constant-Volume Mixed Air

- *At the peak dew-point condition,* the relative humidity drops from 67 percent to approximately 65 percent.
- *On a mild and rainy day,* the relative humidity drops from 73 percent to 70 percent.

In this example, a lower ventilation rate *slightly* reduces space humidity; however, the reduction may *not* remediate the humidity problems inherent to a constant-volume control strategy that is based only on the space dry-bulb temperature. More importantly, lowering the ventilation rate can create other IAQ problems.

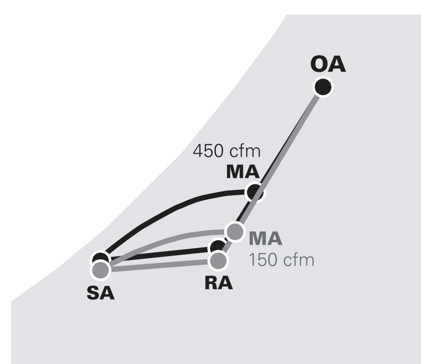
Lowering the ventilation rate does not significantly improve dehumidification performance because space humidity depends on the dew point of supply air leaving the cooling coil — *not* on the dew point of mixed air entering the coil. Lowering the ventilation rate introduces a smaller quantity (and percentage) of outdoor air, which lowers the dew point of the mixed air entering the cooling coil. But as Figure 16 shows, the dew point of the supply air leaving the coil is not much lower than when the system treats a larger quantity of outdoor air.

A lower *sensible load,* however, will result in a supply-air temperature that is too warm to induce moisture condensation from the air passing through the coil. Such conditions will also produce a considerably different supply-air dew point.

In other words, when the outdoor air is more humid than the desired indoor humidity, the extent to which ventilation (outdoor air) affects indoor humidity depends on the loads in the space and the supply-air condition.

Figure 16. Effect of ventilation on the dehumidification performance of a basic, constant-volume HVAC system

Design condition	Ventilation airflow: 450 cfm OA	Ventilation airflow: 150 cfm OA
OA (peak DP)	84.0°F DB, 76.0°F DP	84.0°F DB, 76.0°F DP
RA	74.0°F DB, 67.0% RH	74.0°F DB, 64.6% RH
MA	77.0°F DB, 67.0°F DP	75.0°F DB, 63.0°F DP
SA	63.0°F DB, 61.0°F DP	63.0°F DB, 59.5°F DP



Dehumidifying with Constant-Volume Mixed Air

Climate

The previous example demonstrated that the *quantity* of outdoor air is not necessarily the primary cause of indoor humidity problems. Can the same be said for the *condition* of the outdoor air? Table 2 shows peak dew-point and mild, rainy conditions for seven U.S. cities with differing climates. The table also shows the relative humidity that would result in the same example classroom, with its basic constant-volume HVAC system, at each condition.

Notice the similarity of the peak dew-point conditions for most locations. In these regions, the resulting space relative humidity is similar at peak dew-point conditions. In the dry climates (Denver and San Francisco), the system performs better because the outdoor air is dry enough to provide a dehumidifying effect. Although the increased frequency and duration of humid conditions is greater in hot, humid climates, the conditions capable of causing moisture-related problems occur in many regions. Ignoring system operation at part-load conditions can lead to high indoor humidity in dry climates as well as in hot, humid locales.

Table 2. Constant-volume system performance for various cities in the United States

	Peak dew-point condition		Mild, rainy condition	
Location	Outdoors	Indoor RH	Outdoors	Indoor RH
Baltimore, Maryland	75°F (23.8°C) DP, 83°F (28.1°C) DB	62%	70°F (21.2°C) DB, 69°F (20.6°C) WB	65%
Dallas, Texas	75°F (23.7°C) DP, 82°F (28.0°C) DB	66%	70°F (21.2°C) DB, 69°F (20.6°C) WB	68%
Denver, Colorado	60°F (15.6°C) DP, 69°F (20.4°C) DB	55%	63°F (17.2°C) DB, 61°F (16.1°C) WB	58%
Jacksonville, Florida	76°F (24.6°C) DP, 84°F (28.8°C) DB	67%	70°F (21.2°C) DB, 69°F (20.6°C) WB	73%
Los Angeles, California	67°F (19.4°C) DP, 75°F (23.6°C) DB	62%	63°F (17.2°C) DB, 62°F (16.7°C) WB	65%
Minneapolis, Minnesota	73°F (22.5°C) DP, 83°F (28.5°C) DB	66%	70°F (21.2°C) DB, 69°F (20.6°C) WB	70%
San Francisco, California	59°F (15.2°C) DP, 67°F (19.4°C) DB	56%	54°F (12.2°C) DB, 53°F (11.7°C) WB	56%

Packaged DX Equipment

In constant-volume applications with high ventilation requirements, packaged direct-expansion (DX) air-conditioning equipment can compound indoor humidity problems. More outdoor air, especially in humid climates, increases the required cooling and dehumidification capacity.

Dehumidifying with Constant-Volume Mixed Air

The key to designing a system with adequate dehumidification capability at *all* load conditions lies with determining the proper relationship between airflow and cooling capacity. Sensible cooling loads in the space — *not* ventilation requirements — dictate airflow for the space (unless the required ventilation exceeds the airflow needed to cool the space). The increase in outdoor air required for ventilation requires more cooling capacity. For a given space load, an increase in the ventilation load results in less airflow per cooling ton (m^3/s per kW). The flexibility of applied systems, such as chilled-water air handlers, normally lets you select equipment based on a specific airflow rate (cfm [m^3/s]) *and* a specific cooling capacity (tons [kW]). By contrast, packaged unitary systems (a direct-expansion rooftop air conditioner, for example) typically limit your selection to a cfm/ton ($m^3/s/kW$) range of application.

At the peak dry-bulb condition:

$$Q_s = 1.085 \times 1{,}750\ cfm \times (74°F - T_{supply})$$
$$= 29{,}750\ Btu/hr \therefore T_{supply} = 58.3°F$$

$$(Q_s = 1.21 \times 0.83\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 8.7\ kW \therefore T_{supply} = 14.6°C)$$

Recall that a chilled-water coil provides the air conditioning for the classroom in the preceding examples. The coil was selected to deliver 4.78 tons (16.8 kW) of cooling capacity at 1,500 cfm (0.7 m^3/s) supply airflow, resulting in a flow-to-capacity ratio of 314 cfm/ton (0.042 $m^3/s/kW$).

Most packaged DX air conditioners, however, are designed to operate between 350 and 450 cfm/ton (0.047 and 0.060 $m^3/s/kW$). The classroom in our example would require a nominal 5-ton (17.6 kW) air conditioner that delivers no less than 350 cfm/ton (0.047 $m^3/s/kW$), or 1,750 cfm (0.83 m^3/s). To assure adequate cooling capacity at full-load conditions, you must accept this higher-than-required supply airflow instead of the desired 1,500 cfm (0.7 m^3/s). Because the sensible load is unchanged, however, the 1,750 cfm (0.83 m^3/s) of supply air must be delivered at 58.3°F (14.6°C) to avoid overcooling the space.

At the peak dew-point condition:

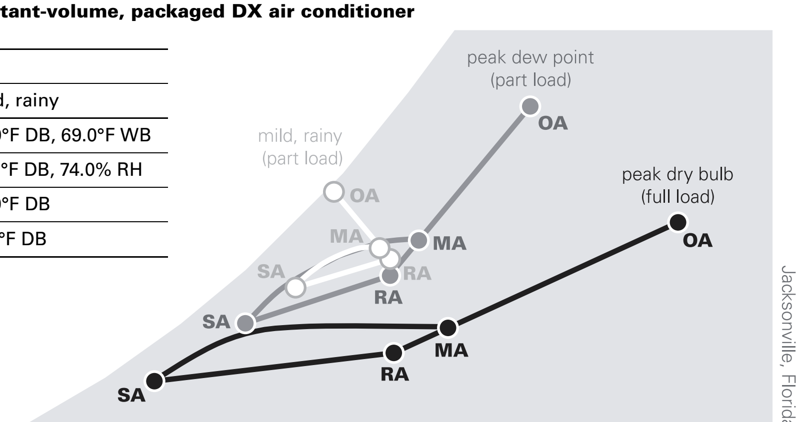
$$Q_s = 1.085 \times 1{,}750\ cfm \times (74°F - T_{supply})$$
$$= 17{,}850\ Btu/hr \therefore T_{supply} = 64.6°F$$

$$(Q_s = 1.21 \times 0.83\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 5.2\ kW \therefore T_{supply} = 18.1°C)$$

Oversized supply airflow results in warmer air leaving the cooling coil. In non-arid climates, this higher supply-air temperature reduces the dehumidification capacity of the system. At the peak dry-bulb condition, the relative humidity in the example classroom increases from 52 percent to 56 percent (Figure 17).

Figure 17. Dehumidification performance of a basic, constant-volume, packaged DX air conditioner

Design condition	Full load	Part load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96.0°F DB, 76.0°F WB	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 56.2% RH	74.0°F DB, 68.7% RH	74.0°F DB, 74.0% RH
MA	79.7°F DB	76.6°F DB	73.0°F DB
SA	58.3°F DB	64.6°F DB	67.5°F DB



Dehumidifying with Constant-Volume Mixed Air

Not surprisingly, the classroom becomes even more humid at the peak dew-point condition. With the thermostat throttling the capacity of the cooling coil to meet the smaller space sensible load, the 64.6°F (18.1°C) supply air offers even less dehumidification; relative humidity climbs to about 69 percent.

On a mild, rainy day:

$$Q_s = 1.085 \times 1{,}750\ cfm \times (74°F - T_{supply})$$
$$= 12{,}250\ Btu/hr \therefore T_{supply} = 67.5°F$$

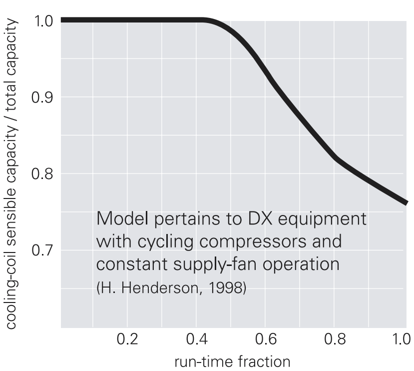
$$(Q_s = 1.21 \times 0.83\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 3.6\ kW \therefore T_{supply} = 19.7°C)$$

As for the mild and rainy day, the supply-air temperature rises to 67.5°F (19.7°C) and the humidity in the space increases to 74 percent.

Selecting larger packaged unitary equipment to provide additional cooling capacity can yield a higher supply airflow, correspondingly warmer supply air, and an elevated indoor humidity.

Note: Excess supply airflow and the increased humidity that accompanies it also can result from a conservative estimate of the space sensible load. When selecting cooling equipment (whether chilled water or DX) for constant-volume applications, exercise particular care to avoid oversizing the supply airflow.

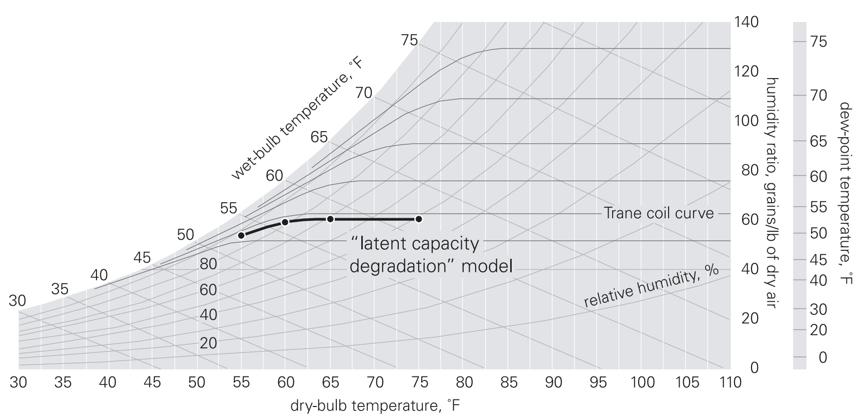
Figure 18. Henderson's "latent capacity degradation" model



Compressor cycling in DX equipment further complicates humidity problems. When the compressors turn off, condensate on the cooling coil re-evaporates and the supply fan "pushes" the moisture downstream to the occupied space. Recent research (Henderson, 1998) led to the development of a "latent capacity degradation model" for DX equipment in which the compressors cycle and the supply fan runs constantly. This model (Figure 18) predicts the latent cooling (dehumidification) capacity of the equipment as a function of the run-time fraction, which represents how long the compressor operates during an hour.

Plotting the latent capacity degradation model on the psychrometric chart (Figure 19) reveals that, over time, there is little difference in performance for cycling DX systems versus chilled-water coils with modulating valves.[2] In other words, given the same supply airflow, the resulting relative humidity indoors will be essentially the same regardless of which type of system (DX or applied chilled water) is used.

Figure 19. Comparison of "latent capacity degradation" model to a Trane coil curve



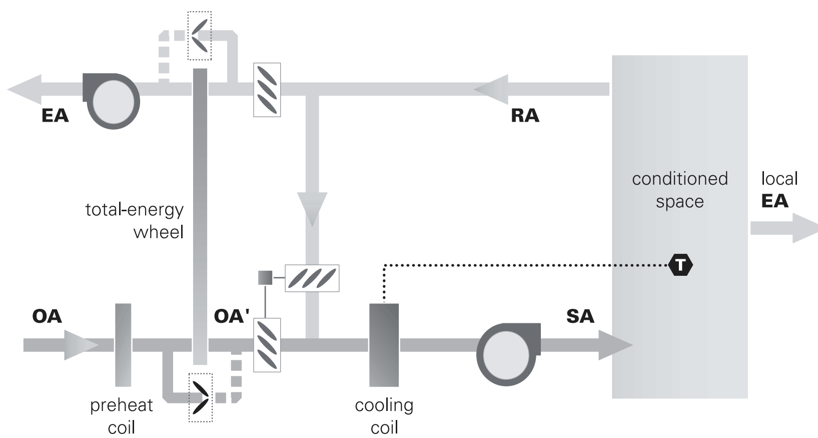
Dehumidifying with Constant-Volume Mixed Air

Total-Energy Recovery

Refer to *Air-to-Air Energy Recovery in HVAC Systems,* Trane applications engineering manual SYS-APM003-EN, for more information about total-energy wheels and other types of air-to-air energy recovery. ■

Some design engineers believe that passive energy-recovery devices provide adequate space dehumidification. A passive, *total*-energy-recovery device, such as a total-energy wheel (Figure 20), revolves through the parallel outdoor- and exhaust-air streams, preconditioning the outdoor air and reducing the capacity required from the cooling and heating coils. During the cooling season, the desiccant-coated wheel removes both sensible heat and moisture from the outdoor air and rejects it to the exhaust air. During the heating season, the sensible heat and moisture that the wheel collects from the exhaust air preheats and prehumidifies the entering outdoor air. Transferring energy between the air streams provides two benefits: downsized equipment for cooling, heating, and humidification; and reduced operating costs.

Figure 20. Constant-volume system with total-energy recovery



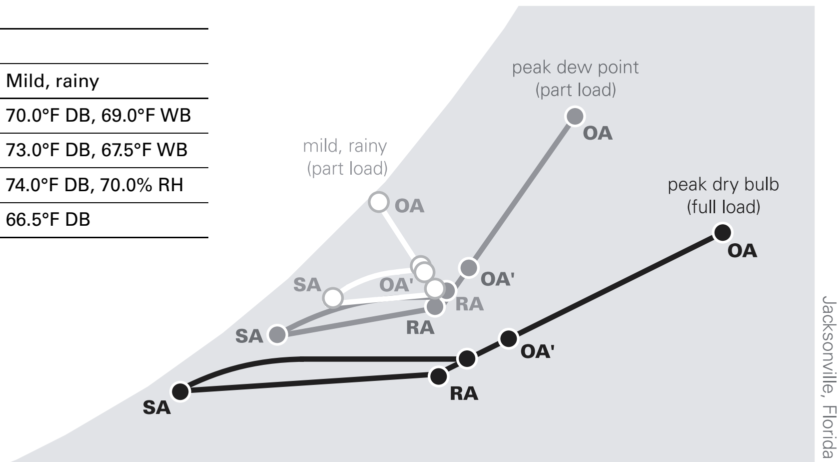
Chilled water applications. Figure 21 (p. 28) illustrates the effect of adding a total-energy wheel, which has an effectiveness rating of 70 percent, to the basic chilled water, constant-volume system in previous examples. *At the peak dry-bulb condition,* the total-energy wheel preconditions the outdoor air to 81°F DB, 66.8°F WB (27.2°C DB, 19.3°C WB). The resulting mixed-air condition reduces the total load on the cooling coil from 4.66 tons (16.4 kW) to 3.5 tons

2 H. Henderson. "The Impact of Part-Load Air-Conditioner Operation on Dehumidification Performance: Validating a Latent Capacity Degradation Model," Conference Proceedings from *IAQ & Energy 1998: Using ASHRAE Standards 62 and 90.1,* (Atlanta, GA: American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., 1998).

Dehumidifying with Constant-Volume Mixed Air

Figure 21. Dehumidification performance of a constant-volume system with total-energy recovery

Design condition	Full load	Part load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96.0°F DB, 76.0°F WB	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
OA'	81.0°F DB, 66.8°F WB	77.0°F DB, 70.0°F WB	73.0°F DB, 67.5°F WB
RA	74.0°F DB, 50.4% RH	74.0°F DB, 64.6% RH	74.0°F DB, 70.0% RH
SA	55.7°F DB	63.0°F DB	66.5°F DB



(12.3 kW); however, the wheel does not affect the cooling loads in the space, so the supply-air temperature and supply airflow remain unchanged. The resulting relative humidity in the classroom drops from 52 percent to 50 percent at full load.

With a smaller sensible load in the space *at the peak dew-point condition,* the thermostat reduces coil capacity from 3.66 tons to 2.47 tons (from 12.9 kW to 8.7 kW) to deliver 63°F (17.2°C) supply air. The resulting relative humidity is 65 percent, compared to 67 percent without the wheel. *On a mild, rainy day,* the resulting relative humidity is 70 percent, compared to 73 percent without the wheel.

Preconditioning the outdoor air with a total-energy wheel significantly reduces mechanical cooling requirements at both full-load and part-load conditions, but it does little to lower indoor humidity.

Even if the total-energy wheel could be 100 percent effective, when it is humid outside, the air passing through the supply side of the wheel would be only as dry as — but never drier than — the air passing through the exhaust side of the wheel. Because the exhaust air stream *originates* in the space, the air leaving the supply side of the wheel will not be drier than the space.

Packaged DX applications. Adding a total-energy wheel to a constant-volume, packaged DX air conditioner reduces the cooling capacity required from the mechanical cooling system and, therefore, increases the required cfm/ton (m³/s/kW). Consequently, the wheel may make it possible to select a smaller DX unit with an airflow that more closely matches space requirements (within the constraints of the flow-to-capacity ratio) in lieu of a larger unit with too much airflow.

Dehumidifying with Constant-Volume Mixed Air

At the peak dry-bulb condition in our example, preconditioning the outdoor air with a 70 percent-effective, total-energy wheel reduces the coil load from 4.66 tons (16.4 kW) to 3.5 tons (12.3 kW). Based on the desired supply airflow of 1,500 cfm (0.7 m^3/s), the new airflow-to-capacity ratio of 428 cfm/ton (0.057 $m^3/s/kW$) now falls within the design operating range for packaged DX equipment. Basing the selection on 1,500 cfm (0.7 m^3/s) rather than 1,750 cfm (0.83 m^3/s) lowers the supply-air temperature from 58.3°F (14.6°C) to 55.7°F (13.1°C) and provides more dehumidification. The resulting relative humidity in the classroom is 50 percent, rather than the 56 percent that resulted from the oversized packaged DX unit without a wheel. *At the part-load, peak dew-point condition,* the wheel results in a relative humidity of 65 percent, compared to 69 percent without the wheel. *On a mild and rainy day,* the resulting space relative humidity is 70 percent, compared to 74 percent without the wheel.

So, the total-energy wheel improves the dehumidification performance of constant-volume, packaged DX equipment by avoiding oversizing the airflow, but it does not eliminate the problem of high indoor humidity.

Cold Supply Air

Lowering the leaving-air temperature of the cooling coil removes more moisture from the air and requires less airflow to offset the sensible-cooling load in the space. *At the peak dry-bulb condition,* delivering supply air to the classroom at 50°F (10°C) rather than 55.7°F (13.1°C) reduces the required supply airflow from 1,500 cfm (0.7 m^3/s) to 1,142 cfm (0.54 m^3/s). Supplying colder, drier air also reduces the relative humidity from 52 percent to 47 percent (Figure 22, p. 30). The total cooling capacity required from the coil increases to 5.1 tons (17.9 kW).

Peak dry-bulb condition:

$$Q_s = 1.085 \times V_{sa} \times (74 - 50°F)$$
$$= 29{,}750\ Btu/hr \therefore V_{sa} = 1{,}142\ cfm$$

$$(Q_s = 1.21 \times V_{sa} \times [23.3 - 10°C])$$
$$(= 8.7\ kW \therefore V_{sa} = 0.54\ m^3/s)$$

The classroom requires warmer supply air *at the peak dew-point condition* because the sensible-cooling load is less. But the supply air is still significantly cooler — 59.6°F (15.3°C) versus 63°F (17.2°C) — than when the system delivers a higher supply airflow. The resulting relative humidity also improves slightly (62 percent versus 67 percent), and the required cooling capacity is 3.8 tons (13.4 kW). *For a mild, rainy day,* the supply-air temperature increases to 64.1°F (17.8°C) and requires a cooling capacity of 1.8 tons (6.3 kW). The resulting relative humidity in the classroom reaches 69 percent.

Peak dew-point condition:

$$Q_s = 1.085 \times 1{,}142\ cfm \times (74°F - T_{supply})$$
$$= 17{,}850\ Btu/hr \therefore T_{supply} = 59.6°F$$

$$(Q_s = 1.21 \times 0.54\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 5.2\ kW \therefore T_{supply} = 15.3°C)$$

Mild, rainy day:

$$Q_s = 1.085 \times 1{,}142\ cfm \times (74°F - T_{supply})$$
$$= 12{,}250\ Btu/hr \therefore T_{supply} = 64.1°F$$

$$(Q_s = 1.21 \times 0.54\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 3.6\ kW \therefore T_{supply} = 17.8°C)$$

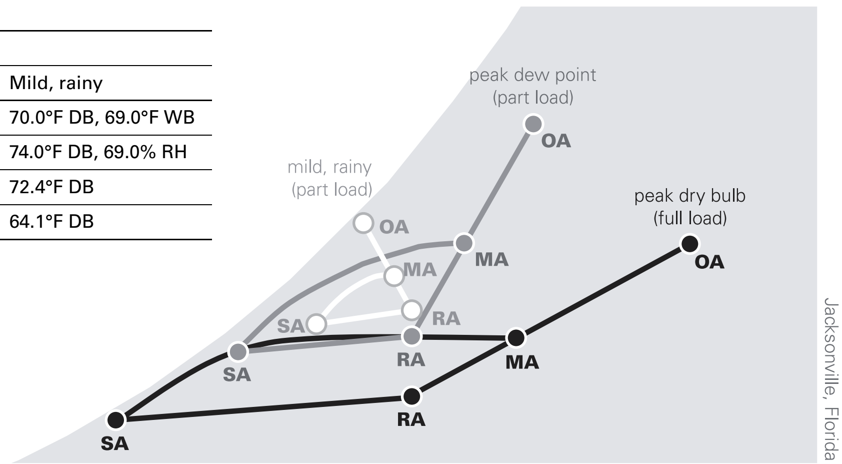
Delivering "cold" supply air can improve the dehumidification performance of a constant-volume system, but it will *not* solve the problem of high indoor humidity.

Note: Applied chilled water systems typically work best for "cold air" distribution because the designer can match the design requirements for airflow and cooling capacity. Packaged DX systems defer many design

Dehumidifying with Constant-Volume Mixed Air

Figure 22. Dehumidification performance of a constant-volume system that delivers "cold" supply air

Design condition	Full load	Part load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96.0°F DB, 76.0°F WB	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 47.0% RH	74.0°F DB, 62.0% RH	74.0°F DB, 69.0% RH
MA	82.7°F DB	77.9°F DB	72.4°F DB
SA	50.0°F DB	59.6°F DB	64.1°F DB



decisions to the manufacturer, which reduces initial cost; however, the limitations of a fixed design may make it difficult to achieve the desired "cold coil" temperature.

Humidity Control during Unoccupied Periods

Constant-volume systems that base control solely on the space dry-bulb temperature (*indirect* dehumidification) are unlikely to remove much moisture at night or during periods of low occupancy when space sensible loads are likely to be very low. Constant-volume systems that include a means to *directly* control space humidity — "dual-path" air handlers (p. 47) or supply-air tempering (p. 50), for example — will require an after-hours source of reheat energy. Dedicated outdoor-air systems, which are discussed in a later chapter (pp. 75–99), may be best suited to address dehumidification needs during unoccupied periods.

Building Pressurization

Consult *Building Pressurization Control,* Trane applications engineering manual AM-CON-17, for information about regulating building pressure through design and control of the HVAC system. ■

Maintaining an appropriate indoor–outdoor pressure difference is generally straightforward during mechanical cooling operation because the airflows in a constant-volume system do not change. Most problems occur at night (when exhaust systems are left on, but the ventilation system is off) or during economizer cooling.

Systems with airside economizers may require some method of building-pressure control to avoid overpressurization.

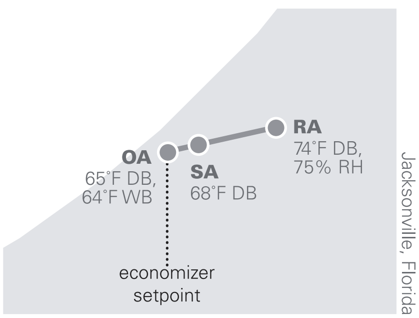
Dehumidifying with Constant-Volume Mixed Air

Airside Economizing

In constant-volume systems, the economizer cycle (p. 18) is often controlled by monitoring the outdoor-air dry-bulb temperature and comparing it with a fixed, predetermined limit. The following example illustrates the potential humidity problems associated with this control strategy.

Suppose that the outdoor-air condition for our Jacksonville school is 65°F DB, 64°F WB (18.3°C DB, 17.8°C WB). The constant-volume system responds to the less-than-design, sensible-cooling load in the space by reducing cooling capacity, which raises the supply-air temperature to 68°F (20°C). If the economizer setpoint is 65°F (18.3°C), the outdoor- and return-air dampers modulate to mix 1,000 cfm (0.47 m³/s) of outdoor air with recirculated return air to maintain the space temperature at setpoint. Although this method of economizer control allows the cooling coil to shut off, the high moisture content of the outdoor air increases the indoor humidity to 75 percent (Figure 23).

Figure 23. Effect of airside economizer control on space humidity



Application considerations

- When using the “fixed dry bulb” method of economizer control, pick a limit that is low enough to avoid bringing *moisture-laden* outdoor air indoors.
- When designing a constant-volume system that requires an economizer to comply with Standard 90.1, investigate “fixed enthalpy” and “electronic enthalpy” control (which are allowed by the standard). Alternatively, consider selecting cooling equipment with an efficiency that is high enough to exempt the system from the economizer requirement.
- To determine the appropriate economizer control, consider the climate, hours of occupancy, and potential operating-cost savings.
- In most climates, avoid using a “differential (comparative) enthalpy” strategy to control the economizer in a constant-volume system. If you *do* opt to use this strategy, install a humidity sensor in the space to disable the economizer whenever the indoor relative humidity exceeds 60 percent.

Implications of ANSI/ASHRAE/IESNA Standard 90.1. Standard 90.1–2001 (Section 6.3.1) contains requirements for economizers in HVAC systems, including when they are required and how they should be controlled. When the cooling capacity of the constant-volume air handler is less than either 65,000 Btu/hr (19 kW) or 135,000 Btu/hr (38 kW), depending on the climate, an economizer is not required. If you choose to use one anyway, requirements related to the control of that economizer no longer apply (because the standard did not require the economizer in the first place). Although compliance with Section 6.3.1 should minimize energy use, *it may not acceptably control indoor humidity at all operating conditions in all climates.*

Section 6.3.1 defines high-limit-shutoff requirements for airside economizers. These requirements are based on climate and control method (fixed dry bulb, differential enthalpy, and so on). “Fixed dry bulb” control of economizers is

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allowed in any climate, with this stipulation: When used in a humid climate, the economizer can only operate when the outdoor dry-bulb temperature is less than or equal to 65°F (18.3°C). (The preceding example shows what can happen when the "fixed dry bulb" economizer setpoint is too high.) Of course, the effect of this control method for a particular installation will depend on the number of hours that the system operates in the "economizer" mode.

Improving Coincidental Dehumidification

The design of a basic constant-volume HVAC system can be altered to improve coincidental dehumidification performance *without* directly controlling space humidity. Table 3 compares the effect of these modifications (described on pp. 32–43), when applied to the example classroom in Jacksonville, Florida.

Table 3. Comparison of coincidental ("indirect") dehumidification[1]

		Resulting relative humidity, %		
Design of constant-volume system		Peak dry bulb	Peak dew point	Mild, rainy day
Basic (unaltered)		52	67	73
Basic plus…	adjustable fan speed (p. 32)	52	60	68
	mixed-air bypass (p. 34)	52	65	68
	mixed-air bypass *and* adjustable fan speed (p. 36)	52	58	65
	return-air bypass with *full* coil face at part load (p. 37)	52	55	60
	return-air bypass with *reduced* coil face at part load (p. 39)	52	64	66

[1] Comparison of dehumidification performance is based on a classroom in Jacksonville, Florida. See "Analysis of Dehumidification Performance" (p. 19) for a description of the room and the constant-volume HVAC system serving it.

Figure 24. Classroom unit ventilator

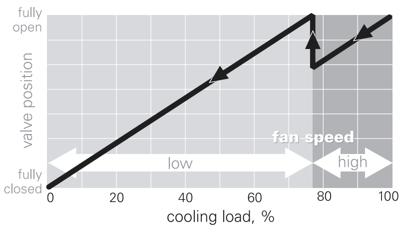
Adjustable Fan Speed

Many room terminals, such as fan–coils and classroom unit ventilators (Figure 24), include fans that can run at different speeds. Depending on the equipment, fan speed is controlled manually by a switch or automatically by a unit controller.

Slowing the fan speed improves the coincidental dehumidification provided by constant-volume room terminals; it is also the first step to reduce cooling capacity. Let's use the example classroom to demonstrate the effect of less airflow. Assume that the HVAC system provides 1,500 cfm (0.7 m³/s) of supply airflow when the fan operates at its highest speed. As the sensible-cooling load in the space decreases (Figure 25), the system initially responds by switching to

Dehumidifying with Constant-Volume Mixed Air

Figure 25. Example of automatic fan-speed adjustment



Peak dew-point condition:

$$Q_s = 1.085 \times 1{,}025\ cfm \times (74°F - T_{supply})$$
$$= 17{,}850\ Btu/hr \therefore T_{supply} = 57.9°F$$

$$(Q_s = 1.21 \times 0.48\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 5.2\ kW \therefore T_{supply} = 14.3°C)$$

Mild, rainy day:

$$Q_s = 1.085 \times 1{,}025\ cfm \times (74°F - T_{supply})$$
$$= 12{,}250\ Btu/hr \therefore T_{supply} = 63.0°F$$

$$(Q_s = 1.21 \times 0.48\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 3.6\ kW \therefore T_{supply} = 17.2°C)$$

low-speed fan operation, which reduces the supply airflow to 1,025 cfm (0.48 m^3/s). As the space load decreases further, the control valve modulates the chilled water flow through the coil to appropriately reduce cooling capacity.

Because the fan operates at high speed when the peak dry-bulb condition exists, dehumidification performance matches that of the basic constant-volume system described at the beginning of our analysis (p. 20). That is, the system supplies 55.7°F (13.1°C) air to satisfy the sensible-cooling load in the space and maintain the 74°F DB (23.3°C) target; the resulting humidity is 52 percent.

At the part-load, peak dew-point condition (Figure 26), the reduced supply airflow results in a lower supply-air temperature, 57.9°F (14.3°C) versus 63°F (17.2°C). Reducing the airflow allows the coil to remove more moisture, improving the dehumidification performance of the system. At this condition, the relative humidity in the space improves from 67 percent to 60 percent; the required cooling capacity is 3.9 tons (13.7 kW).

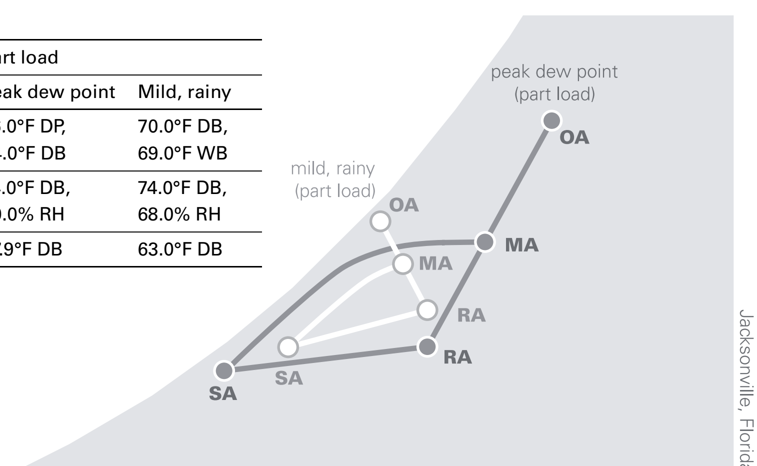
On the mild and rainy day, the supply-air temperature rises to 63°F (17.2°C) and the resulting space relative humidity climbs to 68 percent. The required cooling capacity is 1.7 tons (6.0 kW).

Application considerations

- Adjusting the fan speed offers an important acoustical benefit, particularly for room terminals located within the occupied space: Fans operate quieter at low speed.
- Unit controls should automatically adjust the position of the outdoor-air damper whenever the fan speed changes, thereby assuring that the space continues to receive the proper amount of outdoor air. As part of the system

Figure 26. Constant-volume dehumidification performance at low fan speed

Design condition	Part load	
	Peak dew point	Mild, rainy
OA	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 60.0% RH	74.0°F DB, 68.0% RH
SA	57.9°F DB	63.0°F DB



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air-balancing procedure and with the exhaust fan operating, determine the appropriate damper position for each fan speed.

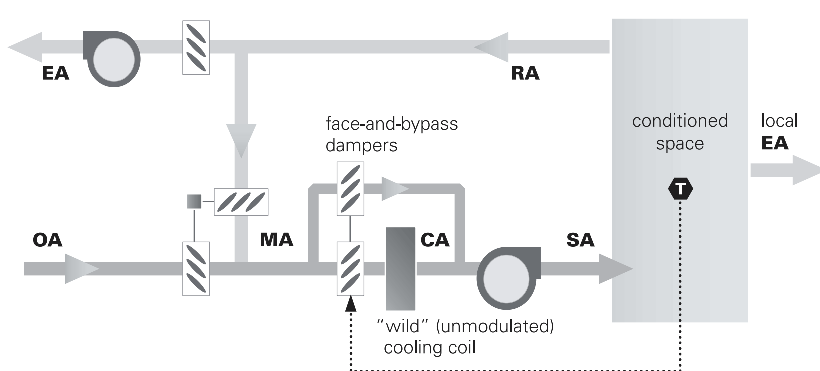
Mixed-Air Bypass

Face-and-bypass dampers, arranged to allow mixed air to bypass the cooling coil, are often used to improve the indirect dehumidification performance of a constant-volume system. Simple and inexpensive, mixed-air bypass blends cold, dry air leaving the cooling coil with bypassed mixed air. The space thermostat controls cooling capacity by adjusting the positions of the linked face-and-bypass dampers, regulating airflow through and around the coil to achieve the proper supply-air temperature (Figure 27); chilled water flow through the coil remains constant. This control method is sometimes described as letting the cooling coil "run wild."

At the peak dry-bulb condition, the face damper is wide open and the bypass damper is closed. All of the mixed air passes through the cooling coil, so dehumidification performance is identical to that of the basic constant-volume system without mixed-air bypass (p. 20).

At the part-load, peak dew-point condition, the face damper modulates closed and the linked bypass damper modulates open to satisfy the space-thermostat setpoint. The entering water temperature and water-flow rate through the coil are unchanged. Diverting some of the mixed air *around* the coil slows the velocity of the air passing *through* the coil; more of the entrained moisture condenses, so the conditioned air (CA) leaves the coil drier and colder — that is, at 52°F (11.1°C) for our example classroom in Jacksonville (Figure 28), as determined with the help of a coil-performance program. The conditioned air then blends with the bypassed, mixed air to achieve the desired supply-air temperature of 63°F (17.2°C).

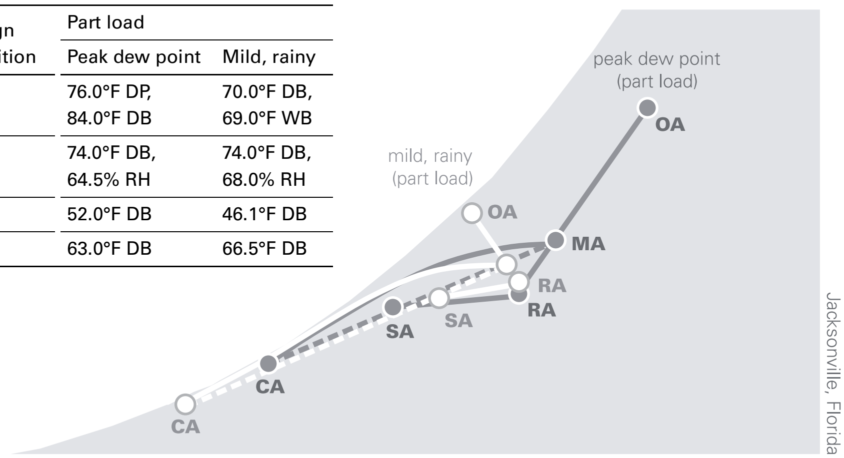
Figure 27. Basic constant-volume HVAC system with mixed-air bypass



Dehumidifying with Constant-Volume Mixed Air

Figure 28. Constant-volume dehumidification performance with mixed-air bypass

Design condition	Part load: Peak dew point	Part load: Mild, rainy
OA	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 64.5% RH	74.0°F DB, 68.0% RH
CA	52.0°F DB	46.1°F DB
SA	63.0°F DB	66.5°F DB



Do not assume that the cold, dry air leaving the coil will adequately dehumidify the space. At the peak dew-point condition, moisture in the bypassed air prevents more than a slight decrease in relative humidity — from 67 percent to 65 percent, in this case. The total coil load increases from 3.68 tons (12.9 kW) to 3.98 tons (14.0 kW).

On the mild and rainy day, the air leaves the cooling coil at 46.1°F (7.8°C) and blends with the bypassed, mixed air to achieve the required supply-air temperature of 66.5°F (19.2°C). The resulting relative humidity rises to 68 percent, and the required cooling capacity is 1.8 tons (6.3 kW). Again, mixed-air bypass improves the indirect (coincidental) dehumidification of this basic constant-volume system … but only slightly.

Application considerations

- Resetting the chilled water temperature or varying the water flow through the cooling coil will degrade the dehumidification performance of a constant-volume system equipped with mixed-air bypass.
- Mixed-air bypass works with either blow-through or draw-through supply fans. It also requires less space than true (full coil face active at part load) return-air bypass, making it advantageous for terminal-unit applications.
- Mixed-air bypass dehumidifies more effectively than return-air bypass in arid climates, where *outdoor air* is usually drier than return air. For climates where *return air* is usually drier than outdoor air, however, return-air bypass (p. 37) dehumidifies more effectively than mixed-air bypass.
- To prevent water vapor from condensing on the interior and/or exterior of the air-handler casing, close the chilled-water control valve when the fan is off, or when the face damper is nearly closed.

Dehumidifying with Constant-Volume Mixed Air

Mixed-air bypass plus adjustable fan speed. Indirect dehumidification provided by a constant-volume system with mixed-air bypass improves when combined with low-speed fan operation as the first step of capacity control. Recall from an earlier example (p. 32) that the system first responds to a decrease in the space sensible-cooling load by switching the fan to low speed, which reduces supply airflow to 1,025 cfm (0.48 m³/s).

Peak dew-point condition:

$$Q_s = 1.085 \times 1{,}025\, cfm \times (74°F - T_{supply})$$
$$= 17{,}850\, Btu/hr \therefore T_{supply} = 57.9°F$$

$$(Q_s = 1.21 \times 0.48\, m^3/s \times [23.3°C - T_{supply}])$$
$$(= 5.2\, kW \therefore T_{supply} = 14.3°C)$$

Mild, rainy day:

$$Q_s = 1.085 \times 1{,}025\, cfm \times (74°F - T_{supply})$$
$$= 12{,}250\, Btu/hr \therefore T_{supply} = 63.0°F$$

$$(Q_s = 1.21 \times 0.48\, m^3/s \times [23.3°C - T_{supply}])$$
$$(= 3.6\, kW \therefore T_{supply} = 17.2°C)$$

When the space load decreases further, the face-and-bypass dampers vary the cooling capacity accordingly by diverting some of the mixed air around the coil. At the part-load, peak dew-point condition (Figure 29), less airflow lowers the supply-air temperature from 63°F (17.2°C) to 57.9°F (14.3°C). Reduced supply airflow also means that more mixed air passes through the coil before blending with the bypassed mixed air. The air that passes through the cooling coil is cooled to 51.5°F DB (10.8°C DB). At this condition, the relative humidity in the space improves to 58 percent, compared to 65 percent with mixed-air bypass only. The required cooling capacity is 4.1 tons (14.4 kW).

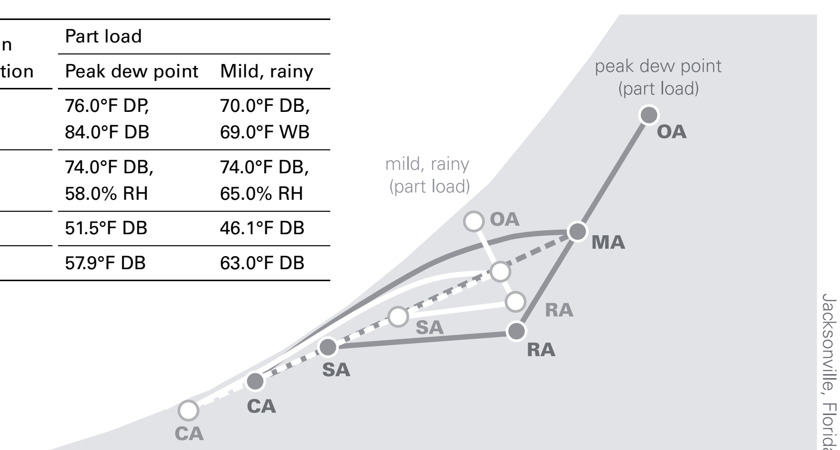
On the mild and rainy day, the air leaves the cooling coil at 46.1°F (7.8°C) and blends with the bypassed, mixed air to achieve the required supply-air temperature of 63°F (17.2°C). The resulting relative humidity is 65 percent, compared to 68 percent with mixed-air bypass only. The required cooling capacity is 1.8 tons (6.3 kW).

Application considerations

- Adjusting the fan speed offers an important acoustical benefit, particularly for in-room terminals: Fans operate quieter at low speed.
- Unit controls should automatically adjust the position of the outdoor-air damper whenever the fan speed changes, thereby assuring that the space continues to receive the proper amount of outdoor air. As part of the system

Figure 29. Constant-volume dehumidification performance with mixed-air bypass and low-speed fan operation

Design condition	Part load: Peak dew point	Part load: Mild, rainy
OA	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 58.0% RH	74.0°F DB, 65.0% RH
CA	51.5°F DB	46.1°F DB
SA	57.9°F DB	63.0°F DB



air-balancing procedure and with the exhaust fan operating, determine the appropriate damper position for each fan speed.

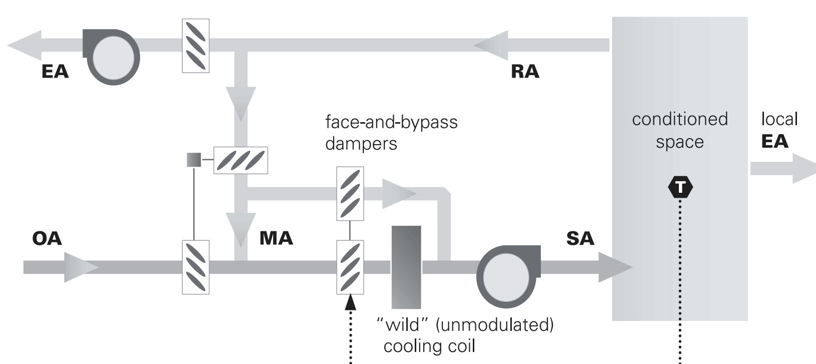
Return-Air Bypass

Full coil face area. Face-and-bypass dampers also can be arranged to bypass only *return* air (Figure 30) instead of a mixture of outdoor and return air. Return-air bypass requires additional space and ductwork, which may increase initial cost; however, it also limits humidity better than other methods of coincidental dehumidification, making return-air bypass a cost-effective enhancement for constant-volume systems.

As with mixed-air bypass, the space thermostat controls cooling capacity by adjusting the position of the linked face-and-bypass dampers, regulating airflow through and around the cooling coil. Because the coil "runs wild," its surface can be very cold at part-load conditions, enhancing the system's ability to dehumidify the space without directly controlling humidity. What makes return-air bypass more effective than mixed-air bypass at most loads, however, is that it directs *all* of the moist outdoor air through the cooling coil. The cold, dry air leaving the cooling coil then blends with relatively dry return air (rather than a mixture of humid outdoor and return air).

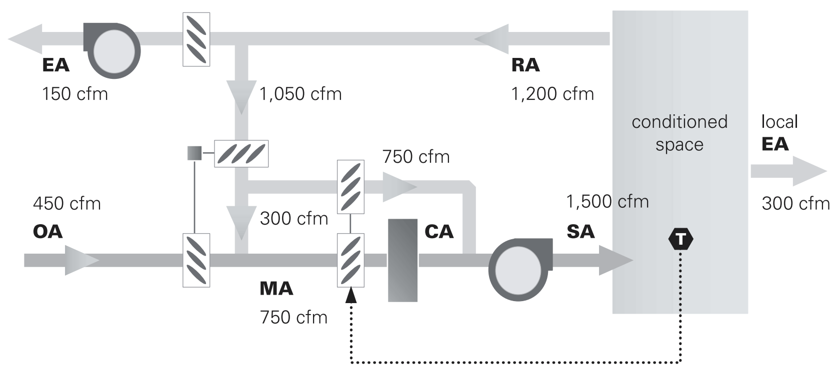
At the peak dry-bulb condition, the face damper is wide open and the bypass damper is closed; therefore, all of the mixed air passes through the cooling coil. At this condition, dehumidification performance is identical to that of the basic constant-volume system in our example Jacksonville classroom (p. 20). As the sensible-cooling load in the space decreases, the face damper modulates toward closed and the linked bypass damper modulates open. Some of the recirculated return air bypasses the cooling coil, while the rest mixes with the outdoor air and passes through the cooling coil. The bypassed return air and conditioned air mix downstream of the coil (Figure 31, p. 38). The entering water temperature and water flow rate through the coil remain unchanged, so

Figure 30. Basic constant-volume HVAC system with return-air bypass



Dehumidifying with Constant-Volume Mixed Air

Figure 31. Effect of return-air bypass on part-load airflows (peak dew-point condition)



It is difficult to accurately model the part-load performance of an HVAC system with return-air bypass. Both the airflow through the coil and the temperature of the mixed air as it enters the coil change as the face-and-bypass dampers modulate. Analyzing part-load performance is a trial-and-error process that is best accomplished using coil modeling software and a psychrometric chart. ■

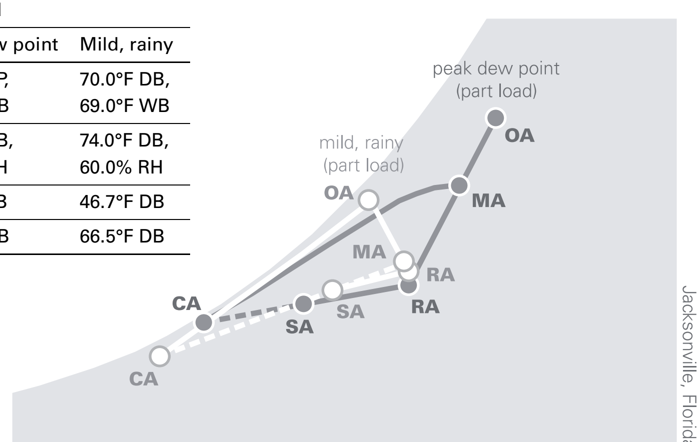
with less airflow through the coil, the air leaving the coil is colder than at the peak dry-bulb condition.

At the part-load, peak dew-point condition (Figure 31, Figure 32), all 450 cfm (0.21 m³/s) of the outdoor air brought into the classroom for ventilation passes through the cooling coil, as does 300 cfm (0.14 m³/s) of the recirculated return air; the resulting coil load is 4.2 tons (14.7 kW). The rest of the recirculated return air, 750 cfm (0.35 m³/s), bypasses the coil and mixes with the 51.8°F (11.0°C) conditioned air leaving the coil. When the blended, 63°F (17.2°C) supply air reaches the classroom, the resulting relative humidity is 55 percent.

On the mild and rainy day (Figure 32, Figure 33), only 400 cfm (0.19 m³/s) of the outdoor air passes through the cooling coil, which results in a cooling load of 2.2 tons (7.7 kW). The remaining 50 cfm (0.02 m³/s) of outdoor air mixes with

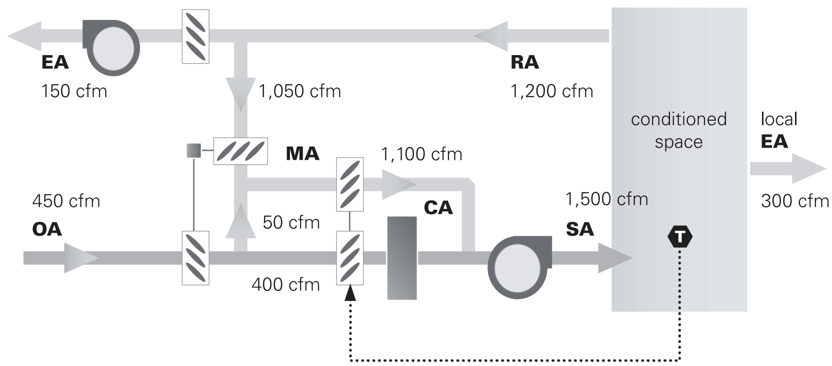
Figure 32. Constant-volume dehumidification performance with return-air bypass and entire coil surface

Design condition	Part load	
	Peak dew point	Mild, rainy
OA	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 55.2% RH	74.0°F DB, 60.0% RH
CA	51.8°F DB	46.7°F DB
SA	63.0°F DB	66.5°F DB



Dehumidifying with Constant-Volume Mixed Air

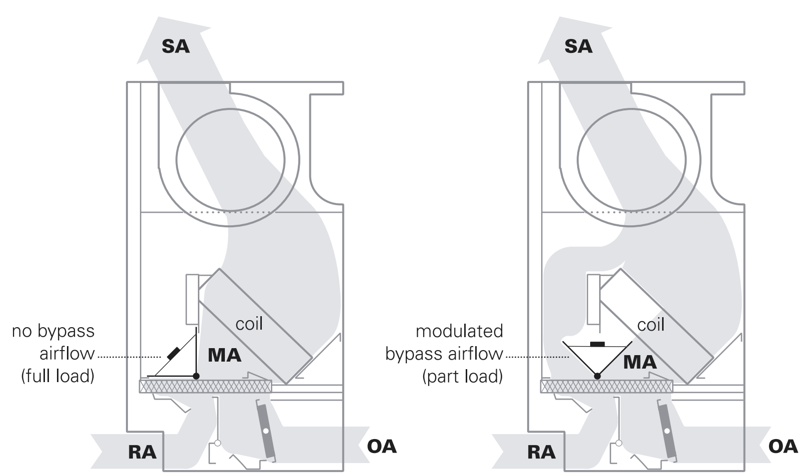
Figure 33. Effect of return-air bypass on part-load airflows (mild, rainy condition)



1,050 cfm (0.5 m³/s) of recirculated return air and bypasses the coil. In the classroom, the blended 66.5°F (19.2°C) supply air results in a relative humidity of 60 percent.

Reduced coil face area. There are various ways to implement return-air bypass in HVAC equipment. Because of the limited space within terminals, such as unit ventilators, operating the face-and-bypass dampers often reduces the usable face area of the coil. In other words, as the face damper opens (in response to a diminishing sensible-cooling load), it actually blocks part of the cooling-coil surface (Figure 34). Less dehumidification occurs because less of the coil is available for cooling and because the velocity of the air passing through the coil is essentially unchanged.

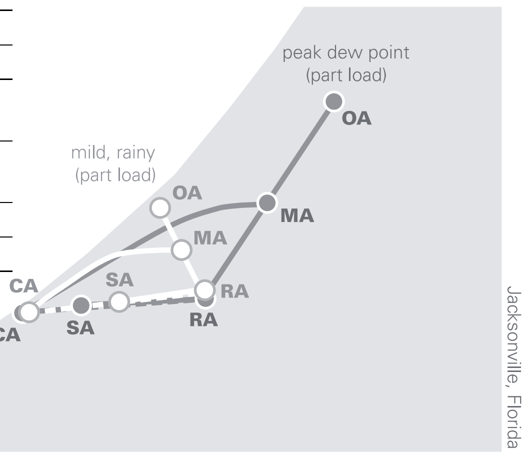
Figure 34. Operation of face-and-bypass dampers that reduce available coil surface



Dehumidifying with Constant-Volume Mixed Air

Figure 35. Constant-volume dehumidification performance with return-air bypass and reduced coil surface

Design condition	Part load	
	Peak dew point	Mild, rainy
OA	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 64.0% RH	74.0°F DB, 66.0% RH
CA	59.6°F DB	59.8°F DB
SA	63.0°F DB	66.5°F DB

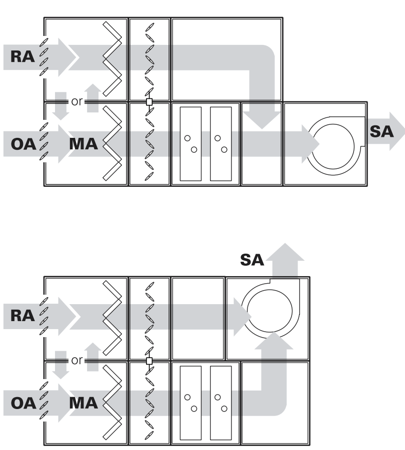


Our example classroom helps demonstrate the effect of less coil surface during return-air-bypass operation (Figure 35). If the face area of the coil decreases as the face damper closes, the relative humidity in the classroom climbs to 64 percent at the part-load, peak dew-point condition, as compared to 55 percent when the entire coil face is available. The required cooling capacity is 3.9 tons (13.7 kW). Mild and rainy conditions require 1.9 tons (6.7 kW) of cooling capacity, and the resulting relative humidity increases to 66 percent.

Application considerations

- Resetting the chilled water temperature or varying the water flow through the cooling coil degrades the dehumidification performance of a constant-volume system equipped with return-air bypass.
- Return-air bypass requires a draw-through supply-fan arrangement and more space, typically, than mixed-air bypass. For proper dehumidification, the air-handler configuration *must* mix the bypassed return air with conditioned air *downstream* of the cooling coil (Figure 36).
- Because it usually directs all incoming outdoor air through the cooling coil, return-air bypass dehumidifies better than mixed-air bypass when the *outdoor* air contains *more* moisture than the return air. For arid climates where outdoor air is usually drier than return air, however, mixed-air bypass (p. 34) works best.
- At very low loads, the airflow through the coil may be less than the outdoor airflow. In this case, the remaining portion of the outdoor air mixes with the recirculated return air to bypass the cooling coil. Even though some of the outdoor air bypasses the cooling coil, the percentage of outdoor air in the bypass path generally is very low when compared to mixed-air bypass.

Figure 36. Examples of air handlers with properly configured return-air bypass

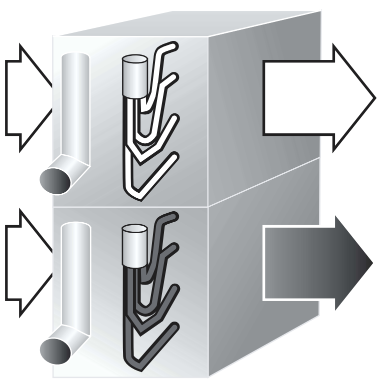


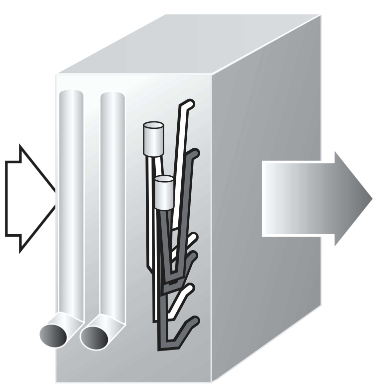
Dehumidifying with Constant-Volume Mixed Air

- When considering equipment that uses return-air bypass, determine whether the face of the cooling coil is blocked at part-load conditions. Reducing the available face area significantly impairs dehumidification performance.
- To prevent water vapor from condensing on the interior and/or exterior of the air-handler casing, close the chilled-water control valve when the fan is off, or when the face damper is nearly closed.

DX Coil Circuiting

Figure 37. Common DX coil configurations





When more than one liquid-refrigerant distributor serves a direct-expansion (DX) cooling coil, the coil is divided into sections (Figure 37). Each section is independently controlled by its own expansion device. The most common configurations for divided, finned-tube evaporator coils are:

- **Face-split**, which divides the coil into parallel sections. Both sections are active when the cooling load is high, providing a uniform leaving-air temperature. At low loads, only one coil section is active to cool and dehumidify the air passing through it. Air passing through the inactive (top) section remains unconditioned. The two air streams mix downstream of the coil, producing average temperature and humidity conditions.
- **Intertwined**, which divides the coil by feeding alternate tubes of the coil via two distributors. At high loads, both distributors feed liquid refrigerant to all of the tubes. At low loads, only one distributor operates, and the coil performs as though its fin surface area were substantially greater. Therefore, the coil surface can be warmer at part load (reducing the risk of frost), and still provide a uniform leaving-air temperature. This performance characteristic makes intertwined coils well-suited for VAV applications.

Which of these DX coil-circuiting arrangements, face-split or intertwined, provides the best *constant-volume* dehumidification? To find out, we used a computer model to simulate performance at the part-load, peak dew-point condition for our example Jacksonville classroom. The HVAC system in this analysis contained two equally sized compressors and a coil served by two solenoid valves. Whenever the thermostat turned off a compressor, the corresponding solenoid valve stopped refrigerant flow through half of the coil. Table 4 compares the results of the analysis.

At the peak dew-point condition, one compressor operates continuously while the other cycles on and off. With both compressors operating and the entire coil surface active, the leaving-air condition — 58.7°F DB, 58.4°F DP (14.8°C DB, 14.7°C DP) — is identical for both coil circuiting arrangements. Offsetting the space sensible-cooling load and maintaining the thermostat setpoint, however, requires 63°F DB (17.2°C DB) supply air. Therefore, operating both compressors

Dehumidifying with Constant-Volume Mixed Air

eventually overcools the space. The thermostat then stops the second compressor, and one of the solenoid valves prevents refrigerant flow through half of the coil.

Table 4. Performance comparison of DX coil-circuiting options at peak dew point[1]

Operating condition		Face-split coil	Intertwined coil
Coil leaving-air temperature with: ■ Both compressors operating ■ Full coil surface active		58.7°F (14.8°C) DB, 58.4°F (14.7°C) DP	58.7°F (14.8°C) DB, 58.4°F (14.7°C) DP
Coil leaving-air temperature with: ■ One compressor operating ■ Half of coil surface active	active coil section only	57.8°F (14.3°C) DB, 57.5°F (14.2°C) DP	not applicable
	mixed air downstream of coil	67.4°F (19.7°C) DB, 62.7°F (17.1°C) DP	64.0°F (17.8°C) DB, 63.5°F (17.5°C) DP
Averaged supply-air condition to match sensible-cooling load		63.0°F (17.2°C) DB, 60.6°F (15.9°C) DP	63.0°F (17.2°C) DB, 62.6°F (17.0°C) DP
Resulting condition in space		74.0°F (23.3°C) DB, 68% RH	74.0°F (23.3°C) DB, 71% RH

[1] Coil performance is based on the peak dew-point condition for Jacksonville, Florida: 76°F DP, 84°F DB (24.6°C DP, 28.8°C DB).

Face-split coil. With only one compressor operating, air passing through the active (bottom) half of the face-split coil is cooled and dehumidified to 57.8°F DB, 57.5°F DP (14.3°C DB, 14.2°C DP). When this conditioned air mixes with the unconditioned air that passed through the inactive (top) half of the coil, the resulting mixed air downstream of the coil is 67.4°F DB, 62.7°F DP (19.7°C DB, 17.1°C DP). The supply air eventually undercools the space because it is too warm to offset the sensible-cooling load. At this point, the thermostat restarts the second compressor and the entire coil surface is active again.

With the second compressor cycling on and off, the average supply-air condition during an hour of HVAC operation is 63.0°F DB, 60.6°F DP (17.2°C DB, 15.9°C DP).

Intertwined coil. Unlike face-split circuiting, every other row in an intertwined coil still receives refrigerant after the second compressor stops. Air passing through the coil is cooled and dehumidified to 64.0°F DB, 63.5°F DP (17.8°C DB, 17.5°C DP). Because the supply air is too warm to offset the sensible-cooling load, the second compressor cycles as necessary to maintain the thermostat setpoint. However, because the supply-air condition is close to the desired condition of 63°F (17.2°C), the system with the intertwined coil operates the second compressor much less throughout the hour than the system with the face-split coil.

Dehumidifying with Constant-Volume Mixed Air

With the second compressor cycling on and off, the average supply-air condition for an hour of operation yields 63.0°F DB, 62.6°F DP (17.2°C DB, 17.0°C DP).

Conclusion. At the peak dew-point condition, with one compressor operating continuously and the other cycling on and off, the constant-volume performance of face-split and intertwined coils is similar. The average supply-air dew point for the face-split coil is only 2°F (1.1°C) less than that of the intertwined coil, and it results in a space relative humidity that is only 3 percent less. To represent a broader range of loads, performance was also compared for mild, rainy conditions (Table 5) and yielded similar results. Again, the face-split coil provided a slightly lower space relative humidity, that is, 71 percent versus 73 percent for the intertwined coil.

None of the performance models accounted for the transient effect of moisture re-evaporating from the inactive half of the face-split coil; so, the already small difference in performance is likely to be even less.

Application consideration

- For constant-volume DX applications in which humidity control is important, use supply-air tempering (pp. 50–52), or separately condition the incoming outdoor air (pp. 44–50), to improve dehumidification performance.

Table 5. Performance comparison of DX coil-circuiting options on a mild, rainy day[1]

Operating condition		Face-split coil	Intertwined coil
Coil leaving-air temperature with: ■ One compressor operating ■ Half of coil surface active	active coil section only	55.0°F (12.8°C) DB, 54.8°F (12.7°C) DP	not applicable
	mixed air downstream of coil	63.9°F (17.7°C) DB, 61.0°F (16.1°C) DP	60.5°F (15.8°C) DB, 60.3°F (15.7°C) DP
Coil entering- and leaving-air temperatures with: ■ No compressors operating ■ Entire coil surface inactive		72.8°F (22.7°C) DB, 66.1°F (18.9°C) DP	72.8°F (22.7°C) DB, 66.1°F (18.9°C) DP
Averaged supply-air condition to match sensible-cooling load		66.5°F (19.2°C) DB, 62.5°F (16.9°C) DP	66.5°F (19.2°C) DB, 63.3°F (17.4°C) DP
Resulting condition in space		74.0°F (23.3°C) DB, 71% RH	74.0°F (23.3°C) DB, 73% RH

[1] Coil performance is based on a mild, rainy condition for Jacksonville, Florida: 70°F DB, 69°F WB (21.2°C DB, 20.6°C WB).

Dehumidifying with Constant-Volume Mixed Air

"Direct" Control of Humidity

Coincidental (indirect) dehumidification enhancements may work well for comfort-cooling applications in certain indoor environments and in certain climates. When latent loads and sensible loads vary significantly, however, or when it is necessary to maintain a low humidity in the occupied space, it may be necessary to *directly* control both dry-bulb temperature *and* humidity. This is usually accomplished either by separately conditioning the outdoor air and return air *or* by overcooling and tempering the supply air.

Note: For the analyses discussed in this section, we assumed that the HVAC system directly controls the indoor relative humidity, keeping it below 60 percent.

Separate Air Paths

Providing separate treatment paths for the outdoor air entering the building and for return air from the space can enable direct humidity control. These paths may reside in individual air handlers, as in a "dedicated outdoor-air system," or within the same air-handler casing, as in a "dual-path air handler."

Dedicated outdoor-air system

Dedicated outdoor-air systems can be arranged in several ways. These configurations are discussed in more detail in "Dehumidifying with Dedicated Outdoor Air," pp. 75–99. ■

As its name implies, a dedicated outdoor-air (OA) system devotes one air handler exclusively to cooling and dehumidifying all outdoor air so that it is drier than the air in the space. The conditioned outdoor air is then delivered either directly to the space or to other air handlers. Common names for an air handler that serves this purpose include "dedicated outdoor-air unit," "100 percent-outdoor-air unit," "fresh-air unit," and "makeup-air unit."

Figure 38. Dedicated outdoor-air system

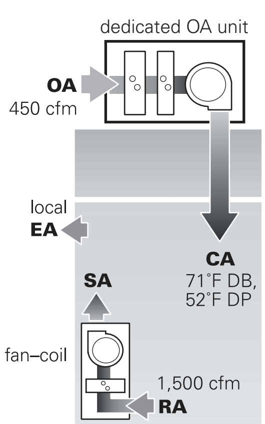


Figure 38 shows a dedicated outdoor-air system that delivers 450 cfm (0.21 m³/s) of dry, neutral-temperature outdoor air directly to our example Jacksonville classroom. The dedicated OA unit dehumidifies the entering outdoor air to a low dew point and then reheats it to the approximate dry-bulb-temperature target for the space. In this example, the dedicated OA system dehumidifies the outdoor air to 52°F DP (11.1°C DP) and then reheats it to 71°F DB (21.7°C DB).

The dedicated OA unit modulates cooling-coil capacity to maintain the desired leaving dew-point temperature. This dew point is determined during the design process to assure that the conditioned outdoor air is dry enough to properly dehumidify the space at all load conditions. With the addition of a space humidity sensor, the leaving dew-point temperature can be reset in response to actual space conditions. Meanwhile, a fan–coil in the classroom cools 1,500 cfm (0.7 m³/s) of recirculated air from the space to offset the local cooling loads.

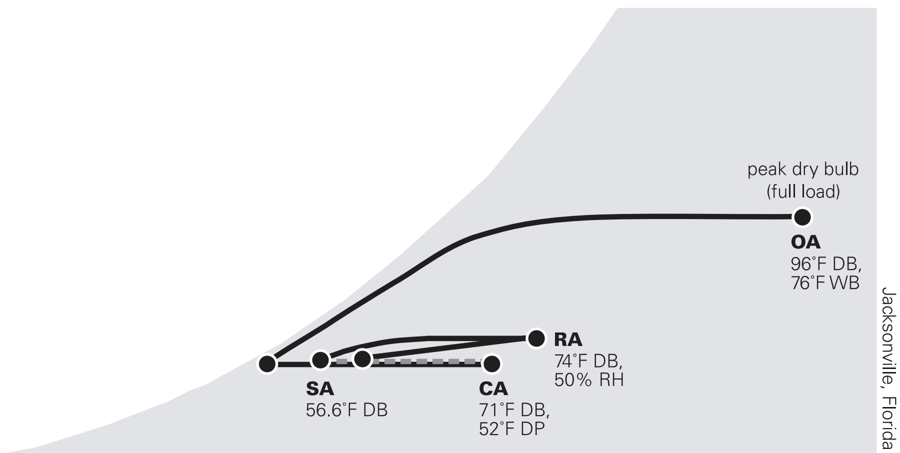
Dehumidifying with Constant-Volume Mixed Air

Peak dry-bulb condition:

$$Q_{s,fc} = Q_{s,space} - Q_{s,doa}$$
$$= 29{,}750\ Btu/hr - [1.085 \times 450 cfm \times (74 - 71°F)]$$
$$= 28{,}285\ Btu/hr$$
$$= 1.085 \times 1{,}500\ cfm \times (74°F - T_{supply})$$
$$\therefore T_{supply} = 56.6°F$$

$$(Q_{s,fc} = Q_{s,space} - Q_{s,doa})$$
$$(= 8.7\ kW - [1.21 \times 0.21\ m^3/s \times (23.3 - 21.7°C)])$$
$$(= 8.29\ kW)$$
$$(= 1.21 \times 0.7\ m^3/s \times [23.3°C - T_{supply}])$$
$$(\therefore T_{supply} = 13.5°C)$$

Figure 39. Dehumidification performance of a dedicated outdoor-air system at peak dry-bulb condition



At the peak dry-bulb condition (Figure 39), the fan–coil cools 100 percent-recirculated air to 56.6°F DB (13.5°C DB); the resulting relative humidity is 50 percent. (This supply-air temperature is slightly warmer than in a basic constant-volume system because the 71°F DB [21.7°C DB] air, supplied by the dedicated outdoor-air unit, offsets part of the sensible-cooling load in the space.) Together, the dedicated outdoor-air unit and the fan–coil — which provide 3.0 tons (10.6 kW) and 2.8 tons (9.8 kW), respectively — offset the total cooling load of 5.8 tons (20.4 kW).

At the part-load, peak dew-point condition (Figure 40, p. 46), the dedicated outdoor-air unit delivers the outdoor air at the same conditions, 71°F DB (21.7°C DB) and 52°F DP (11.1°C DP). Fan–coil capacity modulates to match the lower sensible-cooling load in the space, which raises that unit's supply-air temperature to 63.9°F (17.7°C) and results in a relative humidity of 56 percent. Again, the total cooling load of 4.8 tons (16.9 kW) is divided between the dedicated outdoor-air unit and the fan–coil, which handle 3.4 tons (12.0 kW) and 1.4 tons (4.9 kW), respectively.

Appendix B in this manual describes the design process that we used to select the dedicated outdoor-air unit and the local terminal(s). For this example, we sized the dedicated OA unit to maintain the relative humidity below 60 percent in the classroom at full- and part-load conditions. ■

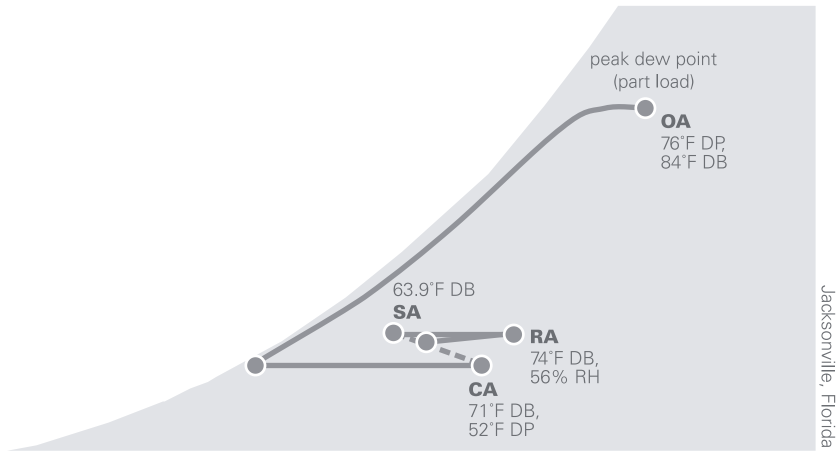
Dehumidifying with Constant-Volume Mixed Air

Peak dew-point condition:

$$
\begin{aligned}
Q_{s,fc} &= Q_{s,space} - Q_{s,doa} \\
&= 17{,}850\ Btu/hr \\
&\quad -[1.085 \times 450 cfm \times (74 - 71°F)] \\
&= 16{,}385\ Btu/hr \\
&= 1.085 \times 1{,}500\ cfm \times (74°F - T_{supply}) \\
&\therefore\ T_{supply} = 63.9°F
\end{aligned}
$$

$$
\begin{aligned}
(Q_{s,fc} &= Q_{s,space} - Q_{s,doa}) \\
(&= 5.2 \\
&\quad -[1.21 \times 0.21\ m^3/s \times (23.3 - 21.7°C)]) \\
(&= 4.8\ kW) \\
(&= 1.21 \times 0.7\ m^3/s \times [23.3°C - T_{supply}]) \\
(&\therefore\ T_{supply} = 17.7°C)
\end{aligned}
$$

Figure 40. Dehumidification performance of a dedicated outdoor-air system at peak dew-point condition



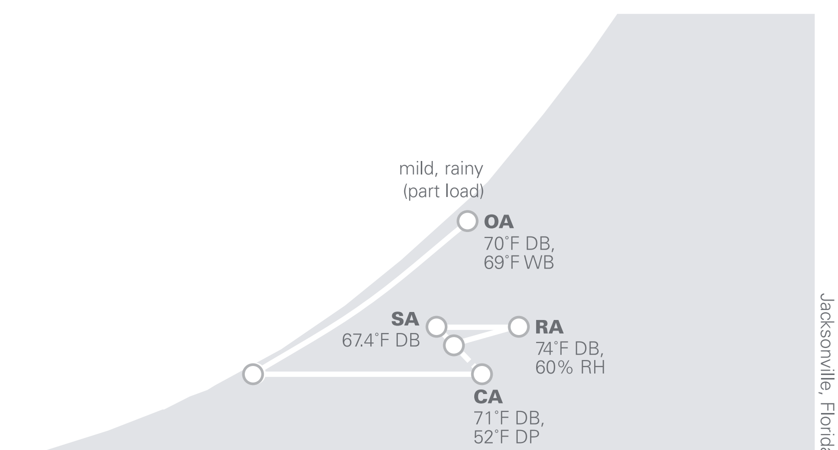
The sensible-cooling load in the space is even lower *on the mild and rainy day,* so the supply-air temperature from the fan–coil increases to 67.4°F (19.5°C), while the conditioned air from the dedicated outdoor-air unit remains unchanged. The resulting relative humidity in the space is 60 percent. Of the total cooling load, which is 2.9 tons (10.2 kW), the dedicated outdoor-air unit handles 2.0 tons (7.0 kW) and the fan–coil handles 0.9 tons (3.2 kW).

Mild, rainy day:

$$
\begin{aligned}
Q_{s,fc} &= Q_{s,space} - Q_{s,doa} \\
&= 12{,}250\ Btu/hr \\
&\quad -[1.085 \times 450 cfm \times (74 - 71°F)] \\
&= 10{,}785\ Btu/hr \\
&= 1.085 \times 1{,}500\ cfm \times (74°F - T_{supply}) \\
&\therefore\ T_{supply} = 67.4°F
\end{aligned}
$$

$$
\begin{aligned}
(Q_{s,fc} &= Q_{s,space} - Q_{s,doa}) \\
(&= 3.6\ kW \\
&\quad -[1.21 \times 0.21\ m^3/s \times (23.3 - 21.7°C)]) \\
(&= 3.2\ kW) \\
(&= 1.21 \times 0.7\ m^3/s \times [23.3°C - T_{supply}]) \\
(&\therefore\ T_{supply} = 19.5°C)
\end{aligned}
$$

Figure 41. Dehumidification performance of a dedicated outdoor-air system on a mild, rainy day



Dehumidifying with Constant-Volume Mixed Air

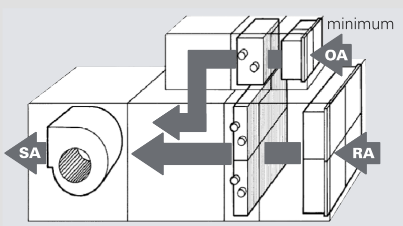
"Dual-path" air handler

A dual-path air handler, which separately conditions both return air and outdoor air, offers an alternative to one dedicated outdoor-air unit for the system and a room terminal in each space. Each air path includes a dedicated cooling coil, but the same constant-volume fan serves both paths (Figure 42).

Working together, the cooling coils in the return-air (RA) and outdoor-air (OA) paths maintain the dry-bulb temperature and humidity in the space:

- The OA cooling coil prevents the humidity in the space from exceeding a predefined limit, dehumidifying the outdoor air enough to offset the latent load. A humidity sensor in the space directly controls coil capacity.
- The RA cooling coil provides the additional cooling needed to offset the sensible load. A thermostat in the space directly controls coil capacity to maintain the space dry-bulb temperature at setpoint.

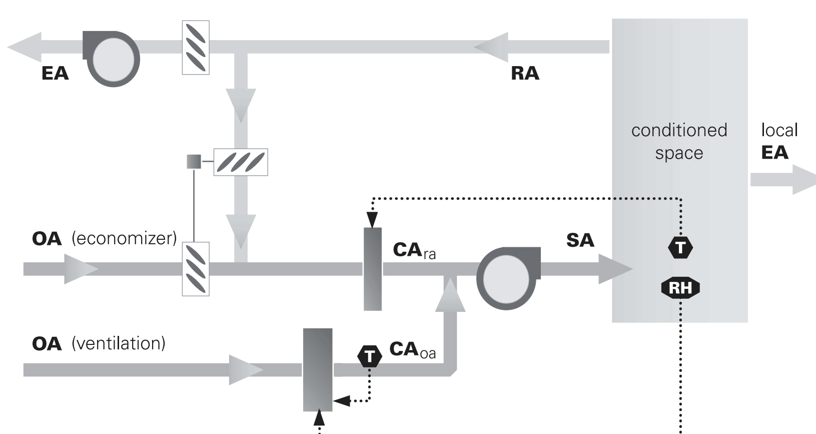
Arranging the air paths in a stacked configuration (below) reduces the footprint of the "dual-path" air handler.



At the example classroom's *peak dry-bulb condition* (Figure 43, p. 48), each coil conditions the air to 55.7°F (13.1°C). When the combined airflows are supplied to the space, the resulting relative humidity is about the same as that achieved by the basic, single-coil HVAC system. The total load of 4.81 tons (16.9 kW) is split between the outdoor-air coil, which handles 2.15 tons (7.6 kW), and the return-air coil, which handles 2.66 tons (9.4 kW).

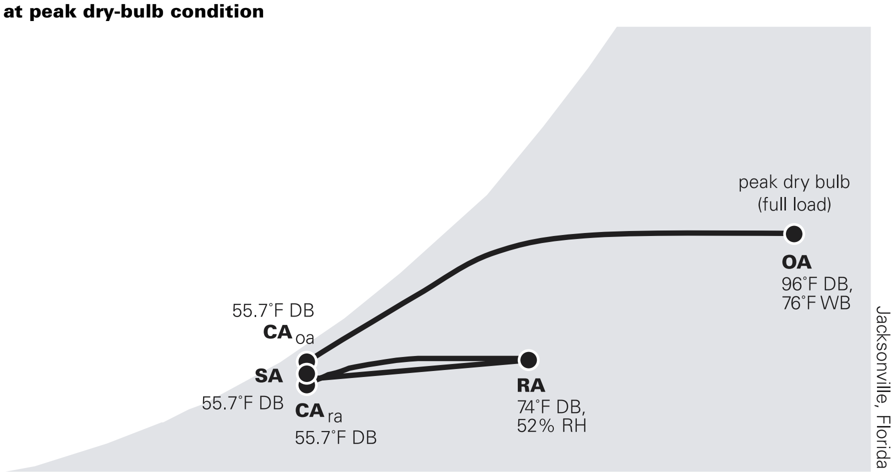
At the part-load, peak dew-point condition (Figure 44, p. 48), the humidity sensor modulates the capacity of the OA cooling coil to maintain the relative humidity at the desired upper limit — 60 percent in this case. This is done by opening the chilled water valve and reducing the leaving-coil temperature to 52°F (11.1°C). The thermostat reduces the capacity of the RA cooling coil to match the diminished sensible load; the resulting temperature of the blended

Figure 42. Dual-path air handler



Dehumidifying with Constant-Volume Mixed Air

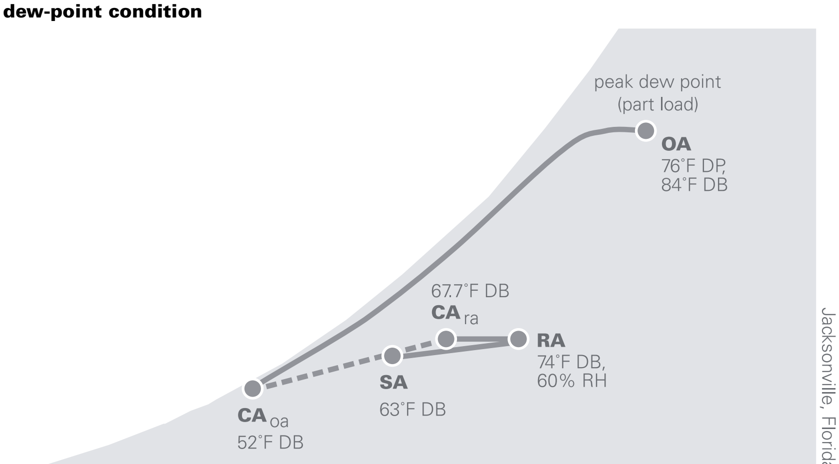
Figure 43. Dehumidification performance of a dual-path air handler at peak dry-bulb condition



supply air is 63°F (17.2°C). Again, the total cooling load of 4.07 tons (14.3 kW) is divided between the two coils: the OA coil handles 3.44 tons (12.1 kW) and the RA coil handles 0.63 tons (2.2 kW). The system directly controls the relative humidity in the space to the desired upper limit of 60 percent.

Maintaining 60 percent-relative humidity when it is *mild and rainy outside* (Figure 45) requires a leaving-air temperature of 51°F (10.6°C) from the OA coil. To achieve a blended supply-air temperature of 66.5°F (19.2°C) and avoid overcooling the space, the RA coil cools the recirculated return air from 74°F (23.3°C) to 73.1°F (22.8°C). The OA coil load is 2.09 tons (7.4 kW), and the RA coil load is 0.09 tons (0.3 kW).

Figure 44. Dehumidification performance of a dual-path air handler at peak dew-point condition



Dehumidifying with Constant-Volume Mixed Air

Figure 45. Dehumidification performance of a dual-path air handler on a mild, rainy day

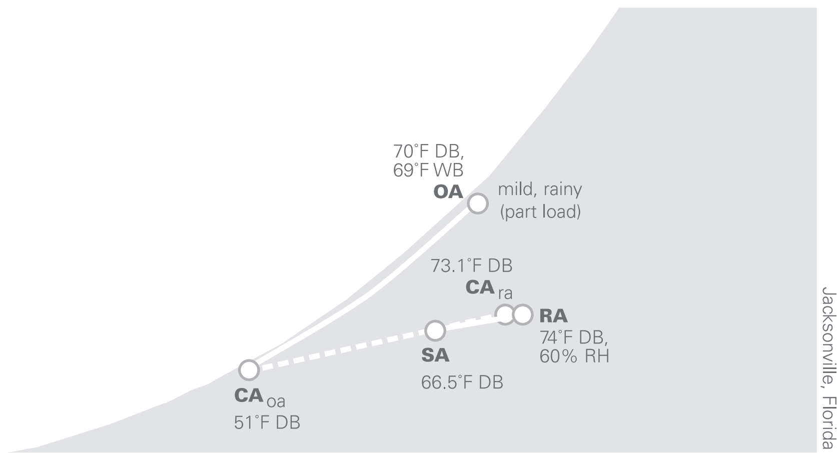


Table 6. Summary of cooling-coil loads for example dual-path air handler[1]

	Cooling load, tons (kW)		
	OA coil	RA coil	Block
Peak DB	2.15 (7.6)	2.66 (9.4)	4.81 (16.9)
Peak DP	3.44 (12.1)	0.63 (2.2)	4.07 (14.3)
Mild, rainy	2.09 (7.4)	0.09 (0.3)	2.18 (7.7)

[1] Outdoor conditions refer to the classroom examples set in Jacksonville, Florida:

Peak dry bulb	96°F (35.7°C) DB,	76°F (24.5°C) WB
Peak dew point	76°F (24.6°C) DP,	84°F (28.8°C) DB
Mild, rainy	70°F (21.2°C) DB,	69°F (20.6°C) WB

Application considerations

Table 6 summarizes the cooling-coil loads at the peak dry-bulb, peak dew-point, and mild, rainy conditions. In this case, the highest load on the RA cooling coil occurs at the peak dry-bulb condition, while the highest load on the OA cooling coil occurs at the peak dew-point condition. However, neither of these conditions necessarily represents the worst-case combined load for these coils.

- Size each cooling coil for its individual *peak* load.
- Because the peak loads on the two coils occur at different times, size the cooling equipment (a central chilled water plant, for example) based on the *block* load rather than the sum of the peak loads.
- If the risk of below-freezing outdoor temperatures exists during occupied periods, consider protecting the chilled water coil in the outdoor air path by: installing a preheat coil upstream of the chilled water coil; using an air-to-air, energy-recovery device to recover heat from the exhaust air; or adding glycol to the chilled water system.
- When the sensible load is low enough, the RA cooling coil may actually turn off while the OA cooling coil continues to produce cold air. A heating coil is generally required to heat the recirculated return air and avoid overcooling the space.

  Section 6.3.2.3 of ASHRAE Standard 90.1–2001 (see p. 8 in this manual) restricts the mixing of hot and cold air. If the dual-path air handler adds heat to the return-air path at certain part-load conditions, *and if it meets one of the following criteria,* it may be exempt from this restriction:

  - *Per Exception B,* the dual-path air handler's design cooling capacity is 6.67 tons (23 kW) or less, *and* the combined load on both coils is less than 50 percent of design capacity when heat is added to the return-air path.

Dehumidifying with Constant-Volume Mixed Air

- *Per Exception D,* the dual-path air handler serves a space that requires specific humidity levels to satisfy process needs; examples include computer rooms, museums, surgical suites, supermarkets, refrigerated warehouses, and ice arenas.
- *Per Exception E,* at least 75 percent of the heat added to the return-air path originates from an on-site source of recovered (condenser heat, for example) or solar energy.

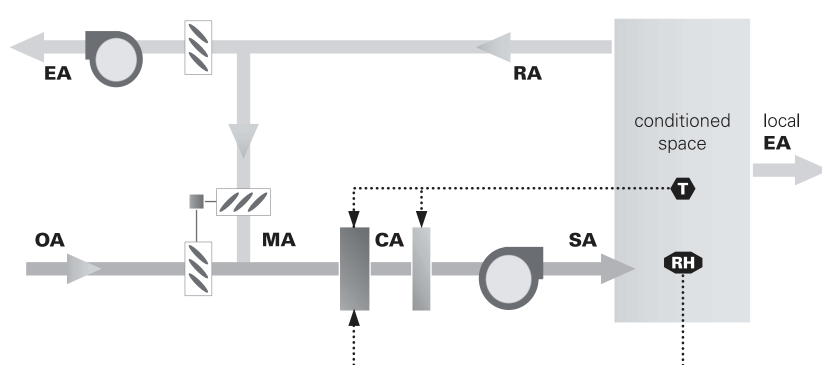
- For cold-weather climates, size the capacity of the RA heating coil to offset the heating loads in the space.
- Size the OA cooling coil for the minimum outdoor airflow required for ventilation. Provide an additional outdoor air path if the system includes an airside economizer.
- For proper system control, install a humidity sensor in the space and assure that the air-handler controls can independently modulate the capacity of each cooling coil.

Supply-Air Tempering

Using a single cooling coil in series with a source of heat for tempering (Figure 46) provides an alternative means of directly controlling indoor humidity. The cooling coil dehumidifies the air to a dew point that is dry enough to maintain the space at an acceptable humidity level. The downstream heating coil "tempers" (raises) the supply-air dry-bulb temperature just enough to maintain the thermostat setpoint and avoid overcooling the space.

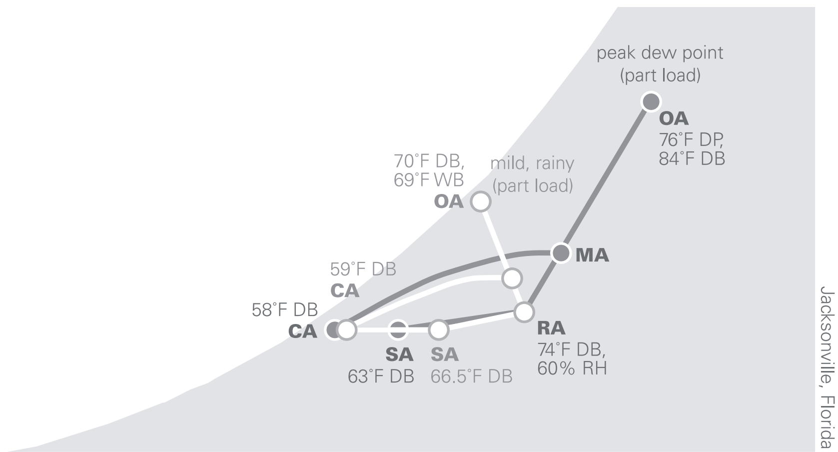
Note: In this manual, we use the term "tempering" instead of "reheat" because the heating coil only moderates the cooling effect of the dry supply air. The space still requires cooling, but not as much as the dehumidification process provides.

Figure 46. Constant-volume HVAC system with supply-air tempering



Dehumidifying with Constant-Volume Mixed Air

Figure 47. Dehumidification performance of supply-air tempering at part-load conditions



At the peak dry-bulb condition for our Jacksonville classroom, a constant-volume system with supply-air tempering performs identically to the basic system in our original example (Figure 14, p. 20). Both systems also respond identically when the sensible-cooling load in the space decreases; that is, they raise the supply-air temperature by reducing the cooling capacity of the coil. If the space humidity rises above the maximum limit, however, the system with supply-air tempering overcools the supply air by modulating the cooling-coil capacity to enforce the maximum humidity limit. Another common control strategy operates the cooling coil at full capacity when the humidity in the space exceeds the specified maximum limit, which lowers the relative humidity more quickly so that the system can return to the normal cooling mode.

In either case, the overcooled supply air then passes through the heating coil, which adds a small amount of heat to temper the air and avoid overcooling the space.

Recall that the basic, constant-volume system achieved a relative humidity of 67 percent at the *peak dew-point condition*. In the system with supply-air tempering (Figure 47), however, the humidity sensor increases the capacity of the cooling coil to avoid violating the 60 percent maximum limit for relative humidity; the resulting leaving-coil temperature is 58°F DB (14.4°C DB). The thermostat prevents overcooling by increasing the capacity of the heating coil, which tempers the supply air to 63°F (17.2°C). The total load on the cooling coil at this condition is 4.74 tons (16.7 kW); there is also a 8.14 MBh (2.4 kW) load on the heating coil.

On the mild and rainy day, the air leaves the cooling coil at 59°F (15°C), and is then tempered to 66.5°F (19.2°C) to enforce the 60-percent-humidity limit in the space. At this condition, the total cooling-coil load is 3.15 tons (11.1 kW) and the heating coil-load is 12.2 MBh (3.6 kW).

Dehumidifying with Constant-Volume Mixed Air

Application considerations. Table 7 and Figure 48 summarize the coil loads at the peak dry-bulb, peak dew-point, and mild, rainy conditions. In this example, the largest cooling burden occurs at the peak dry-bulb condition. Although the enthalpy of the outdoor air is higher at the peak dew-point condition, the humidity sensor in the space permits the relative humidity (RH) to reach the maximum limit; therefore, the enthalpy of the air leaving the coil is higher, too. Lowering the maximum RH limit may cause the highest cooling load to occur at a condition other than the peak dry bulb.

- When designing a constant-volume system that includes supply-air tempering, size the cooling coil and central plant (in the case of a chilled water system) to handle the largest cooling load. Remember that the largest load may occur at full-load *or* part-load conditions, depending on the desired humidity limit and the load characteristics of the space.
- Adding a reheat coil to the supply-air path increases the fan power requirement.
- Supply-air tempering using recovered heat can reduce system operating costs by avoiding the use of new energy for heat. Furthermore, it may allow the system to meet the requirements of energy standards (ASHRAE Standard 90.1, for example) and codes.
- For proper system control, install a humidity sensor in the space. Also, assure that the air-handler controls can determine when to switch between the "standard cooling" and "dehumidification" modes, as well as modulate the capacity of each coil independently.

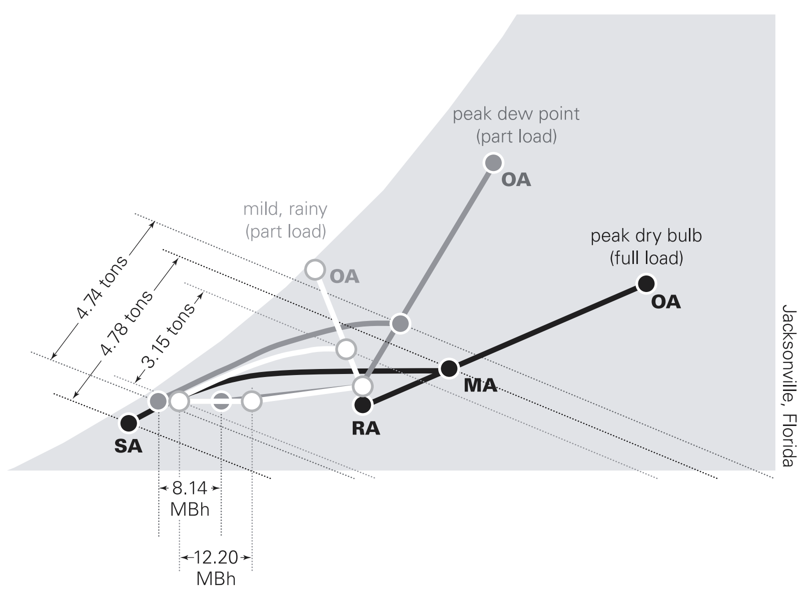
Table 7. Summary of coil loads for supply-air-tempering example[1]

	Coil load (kW)	
	Cooling, tons	Heating, MBh
Peak DB	4.78 (16.8)	0.00 (0.0)
Peak DP	4.74 (16.7)	8.14 (2.4)
Mild, rainy	3.15 (11.1)	12.20 (3.6)

[1] Outdoor conditions refer to the classroom examples set in Jacksonville, Florida:

Peak dry bulb	96°F (35.7°C) DB,	76°F (24.5°C) WB
Peak dew point	76°F (24.6°C) DP,	84°F (28.8°C) DB
Mild, rainy	70°F (21.2°C) DB,	69°F (20.6°C) WB

Figure 48. Comparison of coil loads for supply-air tempering at various conditions



Dehumidifying with Constant-Volume Mixed Air

Recovered heat

Tempering supply air requires a source of heat, but ASHRAE Standard 90.1–2001 *prohibits* the use of "new" energy for tempering or reheat in constant-volume systems, doesn't it? *Not necessarily*. Section 6.3.2.3 (see p. 8 in this manual) defines several *exceptions* for which new-energy reheat is permitted — smaller terminal equipment, midsize equipment that is capable of unloading to 50 percent capacity before reheat is used, and systems that serve certain types of spaces (museums, surgical suites, and supermarkets, for example). Furthermore, tempering is always permissible if at least 75 percent of the energy required for reheat is recovered on-site.

Restricted use of new energy for tempering will probably result in HVAC system designs that temper supply air by recovering heat from the cooling process, particularly in dehumidification applications. Recovering sensible heat from another part of the HVAC system reduces operating costs by avoiding the use of new energy (electricity, hot water, steam, gas) for that purpose. Sources of recoverable heat include:

- Condenser water in a water-cooled, chilled water system
- Hot refrigerant in a refrigeration system
- Another air stream or another location in the same air stream (using an air-to-air heat exchanger)

For more information on recovering heat from a water-cooled chiller, including system configurations that optimize this arrangement, refer to the following manuals:

- *Waterside Heat Recovery in HVAC Systems,* Trane applications engineering manual SYS-APM005-EN
- *Multiple-Chiller-System Design and Control,* Trane applications engineering manual SYS-APM001-EN
- *Application Guide: Chiller Heat Recovery,* ASHRAE publication ISBN 1-8883413-74-5 ■

Condenser-water heat recovery. In a water-cooled-chiller application of the vapor-compression refrigeration cycle, the compressor discharges hot refrigerant vapor into a shell-and-tube heat exchanger or "condenser." Heat transfers from the hot, high-pressure refrigerant vapor inside the condenser shell to the relatively cool water flowing through the tubes; the loss of heat causes the refrigerant to condense into a liquid. The warm condenser water is then pumped to a cooling tower, where it is cooled by the outdoor air. Instead of rejecting the condenser heat to the cooling tower, it can be recovered and used to temper supply air.

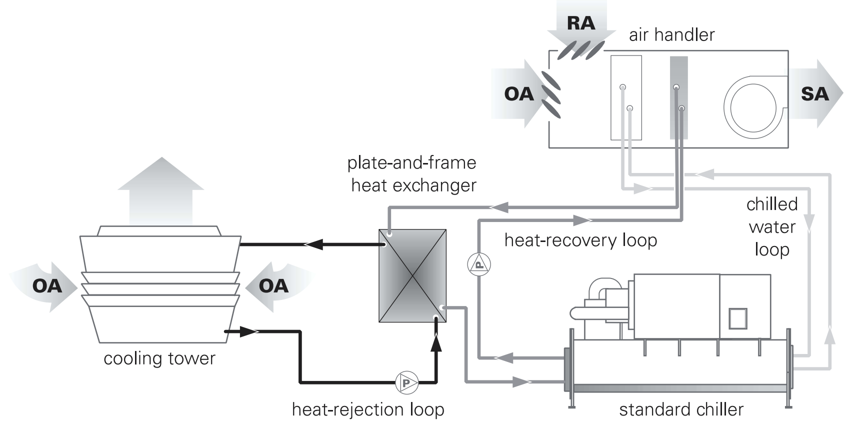
Condenser-water heat recovery is especially cost-effective in supply-air-tempering applications. It not only provides sufficient heat for tempering, but allows the primary heating equipment (boilers) to be turned off during the summer. Of course, recovered heat is only available while the chiller operates.

Any water-cooled chiller can provide sensible heat for supply-air tempering.

Chiller with a standard condenser
The leaving-air temperatures in supply-air-tempering applications typically range from 55°F (13°C) to 75°F (24°C), so the water used for tempering need not need be hot. Most standard water-cooled chillers can provide suitable condenser-water temperatures if operated at a slightly elevated refrigerant-condensing temperature.

Dehumidifying with Constant-Volume Mixed Air

Figure 49. Condenser-water heat recovery using a plate-and-frame heat exchanger



Recirculating the same condenser water through an air handler's hot water coil and an open cooling tower increases the potential for tube fouling. Adding a second water loop, another pump, and a plate-and-frame heat exchanger can eliminate this risk. One loop circulates water through the chiller condenser, the plate-and-frame heat exchanger, and the hot water coil; the other loop circulates water through the plate-and-frame heat exchanger and the open cooling tower (Figure 49).

This arrangement also accommodates closed-circuit cooling towers or other types of evaporative fluid coolers. However, it is less efficient than other methods of condenser-water heat recovery because it adds an intermediate heat-transfer step and uses more pump energy.

"Heating" chiller in sidestream position

For systems with multiple chillers, using one of the chillers as a "heater" offers a more efficient alternative to an intermediate heat exchanger. The "heating" chiller does not require an additional condenser; instead, it is installed and controlled for the condenser heat that it rejects. The evaporator is connected to the chilled water loop, typically in the sidestream configuration (Figure 50); it provides only the cooling needed to satisfy the heating load on the condenser. The temperature of the water leaving the evaporator is a by-product, letting the more efficient cooling-only chillers meet the rest of the cooling load. In effect, the evaporator of the "heating" chiller precools the system return water, reducing the load on the downstream chillers.

This arrangement is well-suited for the year-round heating loads associated with supply-air tempering to control humidity…especially for buildings that require significantly less heating than cooling (buildings in hot, humid climates, for example). The "heating" chiller operates more efficiently than other heat-producing devices, and it is less expensive than a chiller equipped with a second, heat-recovery condenser.

Dehumidifying with Constant-Volume Mixed Air

Figure 50. Condenser-water heat recovery using a sidestream “heating” chiller

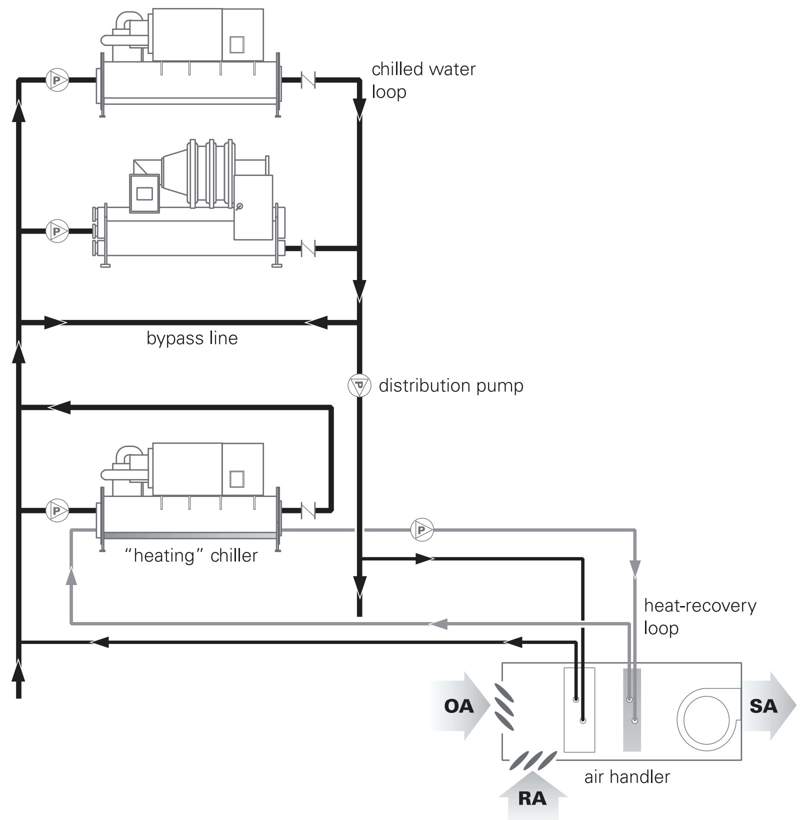
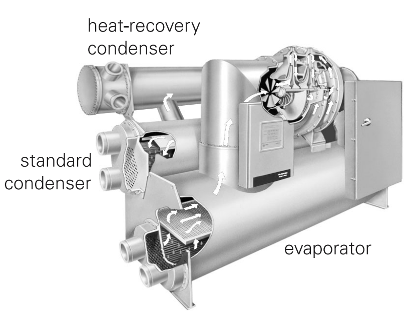


Figure 51. Centrifugal chiller with heat-recovery condenser



Chiller with heat-recovery condenser

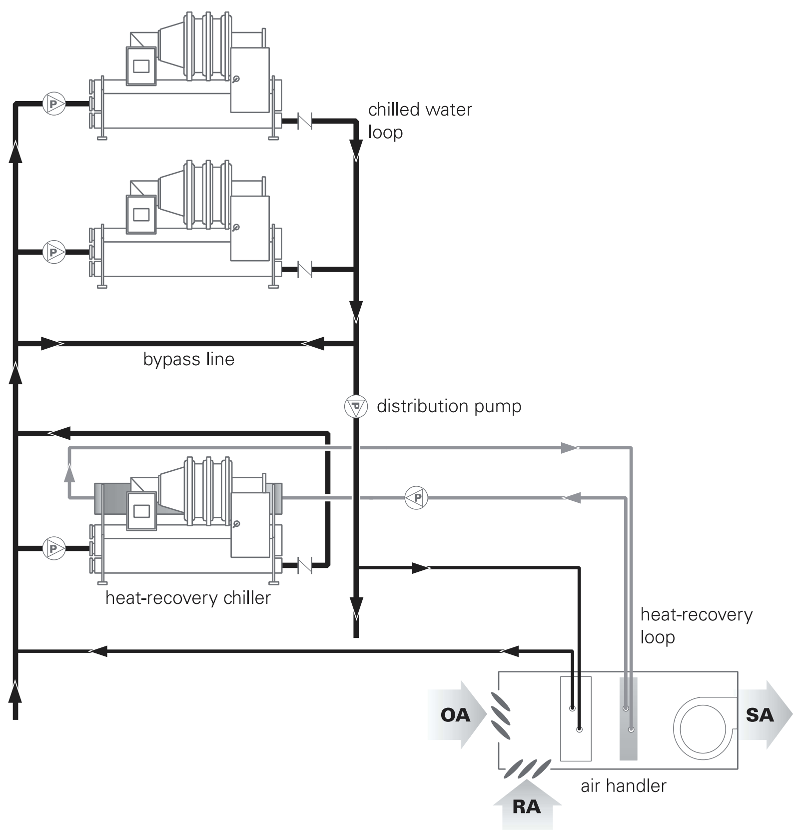
Another means of condenser-water heat recovery requires a chiller that is equipped with either two separate condensers (Figure 51) *or* a single condenser containing two separate tube bundles. Both scenarios require two condenser-water loops: one loop circulates water through a cooling tower to reject heat from the standard condenser, and the other carries water from the dedicated heat-recovery condenser to the hot-water coil for supply-air tempering.

The hot refrigerant vapor discharged by the compressor migrates to the condenser with the lowest pressure. Condenser pressure is a function of that condenser’s leaving water temperature. Raising the leaving water temperature of the *standard* condenser — reducing the flow rate or increasing the entering water temperature — increases the heat available from the heat-recovery condenser. This approach eliminates the intermediate heat exchanger, but the additional condenser increases the initial cost of the chiller.

In a multiple-chiller system, installing the heat-recovery chiller in the sidestream configuration (Figure 52) provides two notable benefits. First, the

Dehumidifying with Constant-Volume Mixed Air

Figure 52. Condenser-water heat recovery using a sidestream, heat-recovery chiller



heat-recovery chiller can supply warmer water than the other operating chillers. It provides only the cooling needed to offset the supply-air-tempering load on the heat-recovery condenser, effectively precooling the returning system water and letting the more efficient cooling-only chillers meet the rest of the cooling load.

Second, if piped in a primary–secondary (“decoupled”) configuration, the heat-recovery chiller receives the warmest system water. This arrangement maximizes recoverable heat as well as the efficiency of this chiller. Because it is positioned *upstream* of the bypass line, the heat-recovery chiller is not affected by excess flow from the supply side of the system (which would otherwise lower that chiller’s entering-water temperature).

Dehumidifying with Constant-Volume Mixed Air

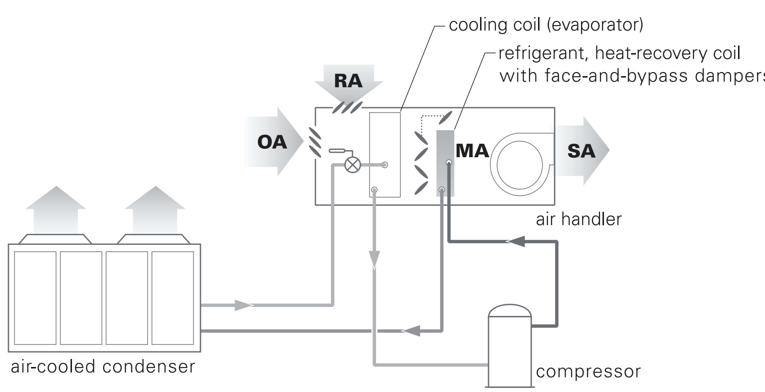
For more information about recovering heat from a DX refrigeration system, refer to *Refrigerant Heat Recovery,* Trane applications engineering manual SYS-AM-5. ■

Refrigerant heat recovery. Heat generated by the vapor-compression refrigeration cycle is also recoverable from direct-expansion (DX) refrigerating equipment and air-cooled chillers. This is typically accomplished by piping a heat-recovering refrigerant coil downstream of the compressor, either in series with, or parallel to, the standard condenser coil. Sometimes described as a "hot-gas reheat coil" or "desuperheater," the heat-recovery coil collects sensible heat from the hot refrigerant vapor and transfers it to the supply air stream. When the humidity indoors exceeds the desired upper limit, the evaporator (DX cooling) coil dehumidifies the supply air; the heat-recovery coil then tempers the cold, dry supply air to avoid overcooling the space.

Figure 53 depicts a refrigerant, heat-recovery coil that is piped in *series* with a standard air-cooled condenser. *If the humidity exceeds the maximum limit,* cooling capacity is increased in response to indoor humidity. The linked face-and-bypass dampers modulate the capacity of the heat-recovery coil to maintain the space temperature at setpoint. *If the humidity does not exceed the maximum limit,* then the cooling coil is controlled to maintain the space temperature at setpoint. Because tempering is unnecessary, the face dampers modulate closed and the bypass dampers modulate open. Directing the air stream around the heat-recovery coil lowers the pressure drop.

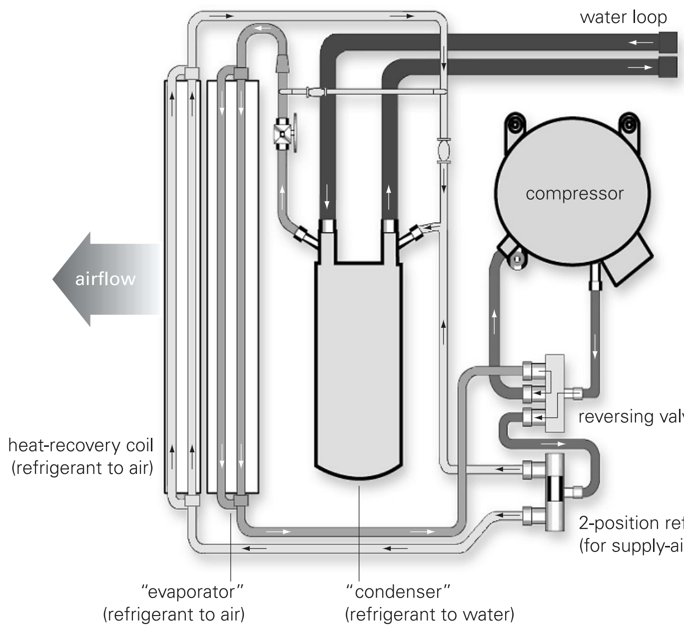
Figure 54 (p. 58) shows a refrigerant heat-recovery coil piped in *parallel* with the condenser in a water-source heat pump. *If the humidity exceeds the maximum limit,* the compressor operates to lower the indoor humidity. A two-position refrigerant valve opens to divert the hot, high-pressure refrigerant vapor from the compressor to the refrigerant heat-recovery coil, which is located downstream of the refrigerant-to-air heat exchanger. (The heat exchanger acts as an "evaporator" during the cooling mode.) This refrigerant valve cycles open and closed to maintain the space temperature at setpoint.

Figure 53. Direct-expansion HVAC system with series-piped condenser and refrigerant, heat-recovery coil



Dehumidifying with Constant-Volume Mixed Air

Figure 54. Water-source heat pump with parallel-piped condenser and heat-recovery coil



Application considerations

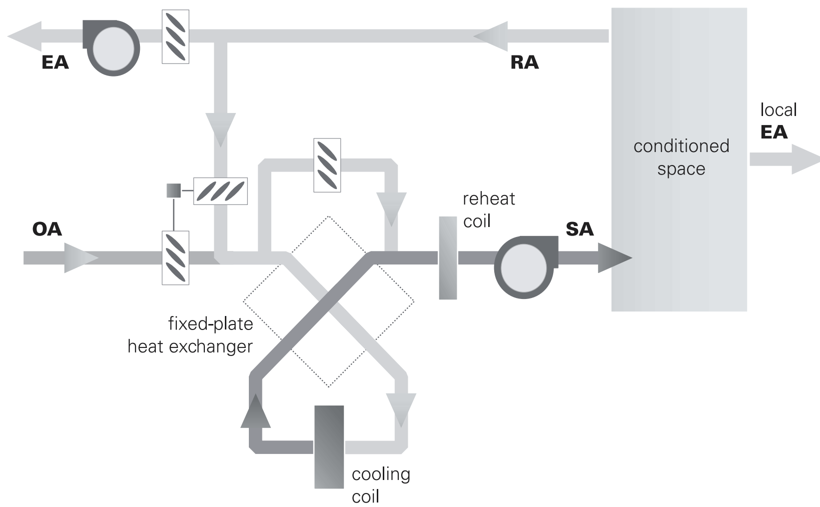
- Refrigerant heat recovery is readily packaged within an air handler and is not susceptible to freezing, which makes it a convenient and relatively inexpensive means to temper supply air.
- If not factory-engineered and -installed, use care when selecting the heat-recovery coil and installing the refrigerant piping and controls. Pipe the heat-recovery coil in *series* with the condenser, and use face-and-bypass dampers to control capacity. Doing so simplifies the refrigeration circuit and facilitates proper compressor lubrication.
- Heat for tempering is only available while the compressor operates.
- The additional coil increases the airside pressure drop and associated fan-energy consumption.

Air-to-air heat recovery. Sensible heat for supply-air tempering can be recovered from another air stream (or another location in the same air stream) by using an air-to-air heat exchanger. The air-to-air heat exchanger can be a coil loop, a fixed-plate heat exchanger, a heat pipe, or a sensible rotary heat exchanger (heat wheel). There are two configurations for using a sensible air-to-air heat exchanger for supply-air tempering: series and parallel.

Air-to-Air Energy Recovery in HVAC Systems (Trane applications engineering manual SYS-APM003-EN) discusses the use of air-to-air heat exchangers for tempering supply air. ■

Dehumidifying with Constant-Volume Mixed Air

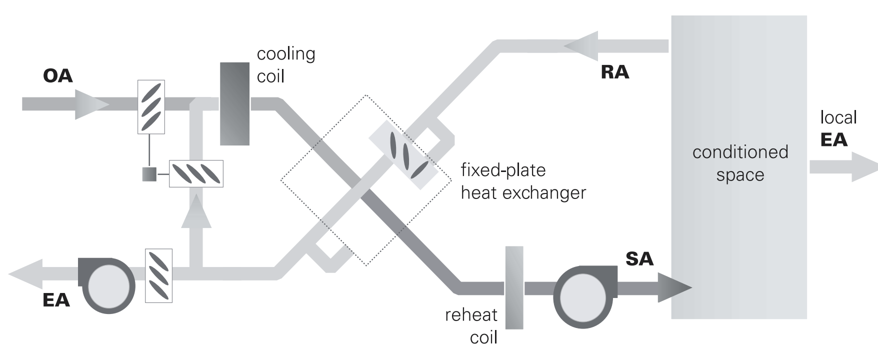
Figure 55. Air-to-air heat recovery applied in a series configuration



Series configuration
Figure 55 depicts a constant-volume, mixed-air system. The air-to-air, sensible-energy-recovery device is applied in a series (or "wrap-around") configuration. To temper the supply air, the device removes sensible heat from the air upstream of the dehumidifying/cooling coil and releases it downstream of the coil. Technically, this arrangement transfers heat from one location to another within the same air stream, rather than "recovering" it from elsewhere in the system.

Parallel configuration
Figure 56 shows the same constant-volume, mixed-air system; this time, the sensible-energy-recovery device is applied in a "parallel" configuration. The device collects sensible heat from the return air stream and releases it downstream of the dehumidifying/cooling coil, warming the supply air.

Figure 56. Air-to-air heat recovery applied in a parallel configuration



Dehumidifying with Constant-Volume Mixed Air

Series or parallel?
In constant-volume, mixed-air systems, both series and parallel configurations reduce the heating energy required for tempering. When comparing the amount of recoverable heat, however, the return air stream (parallel configuration) is a more constant source of heat than the outdoor air (series configuration). If recovered heat is needed when the outdoor air is warmer than the return air, the series configuration transfers more heat; but if recovered heat is needed when the outdoor air is cooler than the return air, the parallel configuration transfers more heat.

Although both configurations "precool" the entering air, which saves cooling energy when tempering, neither configuration permits downsizing of the cooling and heating plants. At the full-load, peak dry-bulb condition, the air is supplied to the space at the design (cold) supply-air temperature. Tempering is unnecessary, so no precooling occurs. Therefore, the cooling coil and cooling plant must be sized to handle the total design-cooling load.

Finally, either configuration may require supplemental heat at certain conditions; also, both require a method for modulating the capacity of the air-to-air heat exchanger to avoid overheating the space during dehumidification. The right choice for a given project depends on the balance of initial cost, energy savings for cooling and heating, and increased fan energy resulting from the additional static-pressure loss through the heat exchanger. ■

Dehumidifying with Variable-Volume Mixed Air

Mixed-air systems use an air handler to condition a combination of outdoor air and recirculated return air before delivering the mixed air to each space. The *variable-air-volume (VAV)* version of a mixed-air system (Figure 57) consists of a central air handler and multiple VAV terminals, each of which is controlled by a space thermostat. Unlike a constant-volume system, which delivers a constant amount of air at varying temperatures, a VAV system delivers varying amounts of constant-temperature air, typically 45°F to 55°F DB (7°C to 13°C DB).

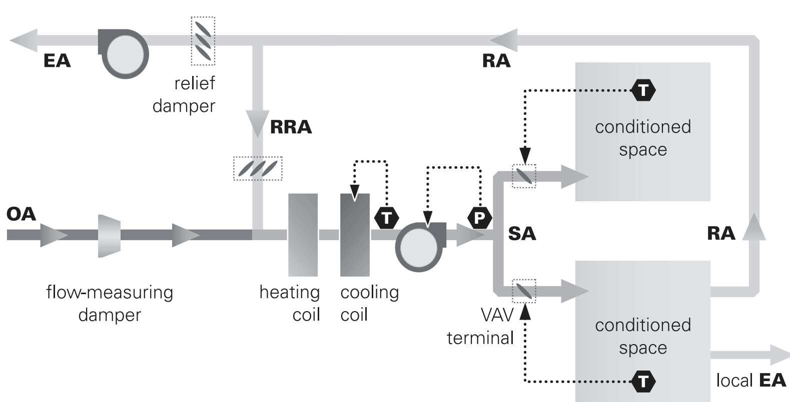
A thermostat in each space compares the dry-bulb temperature to a setpoint, and the VAV terminal responds by modulating the volume of supply air to match the changing sensible-cooling load in the space. Meanwhile, the central supply fan modulates to maintain the static-pressure setpoint in the duct system, and the capacity of the central cooling coil modulates to maintain a constant supply-air dry-bulb temperature.

VAV systems typically provide effective, *coincidental* dehumidification over a wide range of indoor load conditions (sensible-heat ratios). If supply-air-temperature reset is not used, and as long as any space needs cooling, the VAV air handler will supply dry (low-dew-point) air to all VAV terminals. Moisture generated within the space is absorbed by the dry supply air — offsetting the latent load — then removed from the space by the return air stream.

Analysis of Dehumidification Performance

Accurate predictions of coincidental dehumidification require an analysis of system operation at both full-load *and* part-load conditions. The following examples are based on the 10,000 ft^3 (283 m^3), 30-occupant classroom in Jacksonville, Florida. Unlike the previous constant-volume examples, which featured a single-space system, the classroom is served by a multiple-space system. To provide thermal comfort, the target condition is 74°F DB (23.3°C DB)

Figure 57. Basic, variable-air-volume HVAC system



Dehumidifying with Variable-Volume Mixed Air

and 50 percent-relative humidity, with a design supply airflow of nine air changes or 1,500 cfm (0.7 m³/s) per hour.

Performance at peak dry-bulb (full-load) condition. At the peak dry-bulb condition, the space sensible-cooling load and supply-air temperature are the same as for a constant-volume system. Given the supply airflow of 1,500 cfm (0.7 m³/s), the system must deliver 55.7°F (13.1°C) supply air to offset the sensible cooling load in the space and satisfy the thermostat setpoint of 74°F DB (23.3°C DB).

At the peak dry-bulb condition:

$$Q_s = 1.085 \times 1{,}500\ cfm \times (74°F - T_{supply})$$
$$= 29{,}750\ Btu/hr \therefore T_{supply} = 55.7°F$$

$$(Q_s = 1.21 \times 0.7\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 8.7\ kW \therefore T_{supply} = 13.1°C)$$

Psychrometric analysis (Figure 58) reveals that the cooling coil removes both sensible heat and moisture from the air, directly controlling space temperature and coincidentally affecting space humidity. Maintaining the temperature in the space at 74°F (23.3°C) requires a total capacity of 4.78 tons (16.8 kW) from the cooling coil and results in a comfortable relative humidity of 52 percent.

As the sensible-cooling load decreases, the VAV system supplies less air to the space while maintaining a constant supply-air temperature.

Performance at peak dew-point (part-load) condition. Lower solar- and conducted-heat gains and cooler outdoor air reduce the sensible-cooling load in the classroom. With no change in the occupant-generated latent load, the sensible-heat ratio for the space drops to 0.77. The supply-air temperature remains constant at 55.7°F (13.1°C), so supply airflow is reduced to 899 cfm (0.42 m³/s). The required cooling capacity is 4.0 tons (14.2 kW).

At the peak dew-point condition:

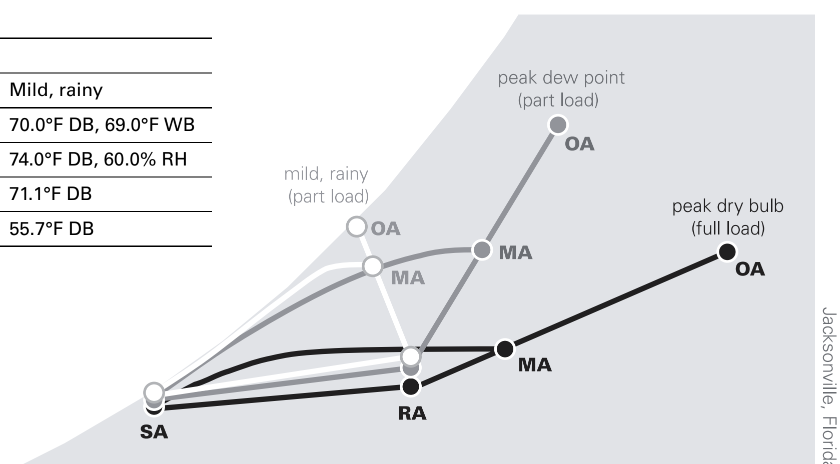
$$Q_s = 1.085 \times V_{sa} \times (74 - 55.7°F)$$
$$= 17{,}850\ Btu/hr$$
$$\therefore V_{sa} = 899\ cfm$$

$$(Q_s = 1.21 \times V_{sa} \times [23.3 - 13.1°C])$$
$$(= 5.2\ kW)$$
$$(\therefore V_{sa} = 0.42\ m^3/s)$$

Because the supply air is still cool and dry, the relative humidity in the classroom only rises to 57 percent (Figure 58). By contrast, the relative humidity reaches 67 percent when the classroom is served by a basic constant-volume system operating at the same condition (p. 21).

Figure 58. Dehumidification performance of a variable-air-volume HVAC system at various outdoor conditions

Design condition	Full load	Part load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96.0°F DB, 76.0°F WB	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 52.4% RH	74.0°F DB, 57.0% RH	74.0°F DB, 60.0% RH
MA	80.6°F DB	79.0°F DB	71.1°F DB
SA	55.7°F DB	55.7°F DB	55.7°F DB



Dehumidifying with Variable-Volume Mixed Air

Performance on a mild, rainy day (part-load condition). Although the peak dew-point condition is helpful for analyzing the part-load dehumidification performance of an HVAC system, *do not* assume that it represents worst-case conditions for humidity control. Typically, indoor humidity depends as much on the sensible and latent cooling loads in the space, the type of HVAC system, and the method of controlling that system, as it does on outdoor conditions.

Mild, rainy day:

$$Q_s = 1.085 \times V_{sa} \times (74 - 55.7\,°F)$$
$$= 12{,}250\ Btu/hr$$
$$\therefore V_{sa} = 617\ cfm$$

$$(Q_s = 1.21 \times V_{sa} \times [23.3 - 13.1\,°C])$$
$$(= 3.6\ kW)$$
$$(\therefore V_{sa} = 0.29\ m^3/s)$$

Consider our example Jacksonville classroom on a mild, rainy day (Figure 58) — 70°F DB, 69°F WB (21.2°C DB, 20.6°C WB). With no change in the occupant-generated latent load, the space sensible load drops to 12,250 Btu/hr (3.6 kW) and the space sensible-heat ratio drops to 0.70. Only 617 cfm (0.29 m^3/s) of constant-temperature supply air is required to offset the sensible load without overcooling the space. At this airflow, the relative humidity increases to 60 percent while the required capacity from the cooling coil decreases to 2.1 tons (7.4 kW).

Application Considerations

Minimum Airflow Settings

In most applications, each VAV terminal has a minimum airflow setting that usually represents either the ventilation requirement for the space or the performance limits of the diffusers or VAV terminal. Providing less than the required minimum airflow may:

- Underventilate the space and degrade indoor air quality.
- "Dump" cold supply air into the space, making occupants uncomfortable. (Most diffusers require a minimum discharge velocity to properly mix the air within the space.)
- Cause erroneous airflow readings that interfere with proper control. (The accuracy of the flow sensor in the VAV terminal is based upon a specific airflow range.)

The minimum airflow setting for the VAV terminal that serves our example classroom is 550 cfm (0.26 m^3/s).

Mild, rainy day and minimum airflow:

$$Q_s = 1.085 \times 700\ cfm \times (T_{space} - 55.7\,°F)$$
$$= 12{,}250\ Btu/hr \therefore T_{space} = 71.8\,°F$$

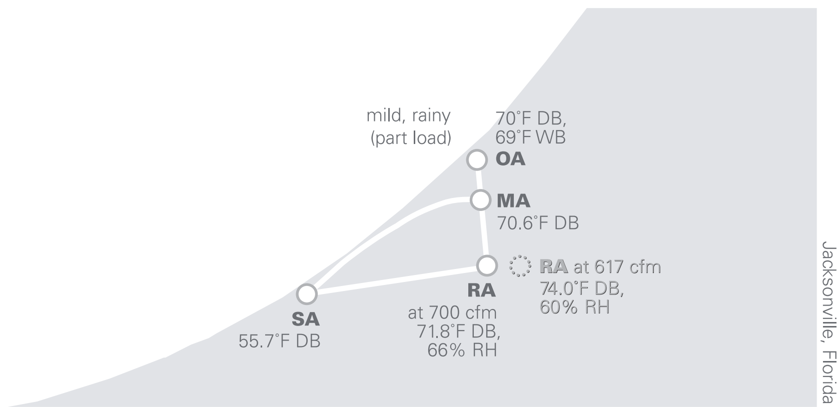
$$(Q_s = 1.21 \times 0.33\ m^3/s \times [T_{space} - 13.1\,°C])$$
$$(= 3.6\ kW \therefore = T_{space} = 22.1\,°C)$$

Eventually, the sensible-cooling load in the space becomes small enough that the required supply (primary) airflow is less than the minimum airflow setting of the VAV terminal. If we assume that the minimum airflow setting for the example classroom is 700 cfm (0.33 m^3/s), then this situation occurs on a mild, rainy day. If the supply-air temperature is held constant at 55.7°F (13.1°C), the VAV system will overcool the space to 71.8°F (22.1°C).

A psychrometric analysis (Figure 59, p. 64) reveals that the relative humidity will climb to 66 percent because of the decreased dry-bulb temperature. As a

Dehumidifying with Variable-Volume Mixed Air

Figure 59. Overcooling results when minimum airflow exceeds required airflow



result, the classroom will *feel* cool and damp even though the actual moisture content of the air is unchanged from the previous example in which only 617 cfm (0.29 m³/s) is supplied to the classroom.

Supply-Air-Temperature Reset

Mild, rainy day and SA temperature reset:

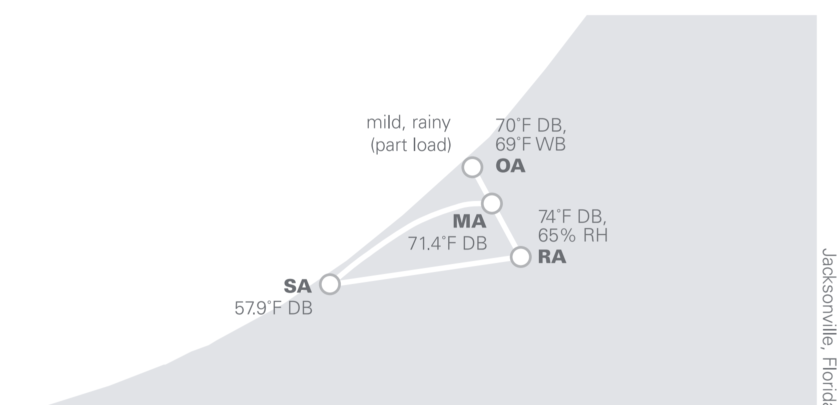
$$Q_s = 1.085 \times 700\ cfm \times (74°F - T_{supply})$$
$$= 12{,}250\ Btu/hr \therefore T_{supply} = 57.9°F$$

$$(Q_s = 1.21 \times 0.33\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 3.6\ kW \therefore T_{supply} = 14.3°C)$$

One way to prevent overcooling is to reset the supply-air temperature upward at low-load conditions. If the minimum airflow setting is 700 cfm (0.33 m³/s), raising the supply-air temperature to 57.9°F (14.3°C), for example, avoids overcooling the classroom on a mild, rainy day…but the cooling coil also removes less moisture from the supply air. As a result, the relative humidity in the space increases to 65 percent (Figure 60).

Resetting the supply-air temperature reduces the energy consumed by the mechanical cooling equipment—from 2.1 tons (7.2 kW) to 1.9 tons (6.7 kW) in

Figure 60. Effect of supply-air-temperature reset on dehumidification performance

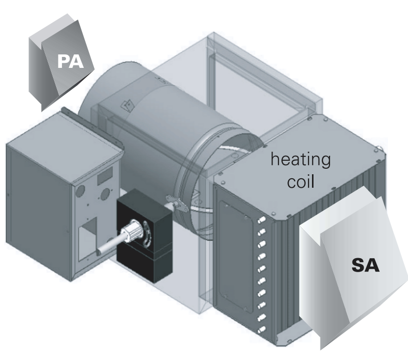


this example. All spaces receive warmer air. Therefore, the spaces not only become more humid, but also require *more* air to offset the sensible-cooling loads. The fans therefore consume more energy.

Note: In VAV applications, supply-air-temperature reset is typically used to avoid excessive reheat during cold weather. Avoid using supply-air-temperature reset during the cooling season unless the system analysis indicates that the savings in mechanical cooling and reheat energy will outweigh the increase in supply-fan energy and space humidity.

Supply-Air Tempering at VAV Terminals

Figure 61. VAV terminal with heating coil



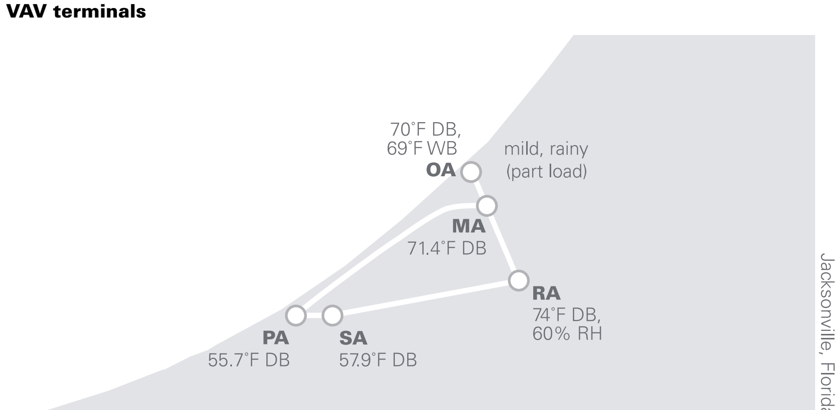
Overcooling and increased humidity can be avoided by tempering the supply air when it diminishes to the minimum airflow setting of the VAV terminal. "Tempering" moderates the cooling effect by adding sensible heat to the supply air, either at the VAV terminal (Figure 61) or within the space.

Figure 62 illustrates the effect of adding supply-air tempering to the VAV system that serves our example classroom. When the supply airflow is reduced to the minimum airflow setting of 700 cfm (0.33 m³/s), a heating coil in the VAV terminal warms 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 classroom and, on a mild and rainy day, results in a relative humidity of 60 percent. The total load on the cooling coil is 2.1 tons (7.4 kW), while the load on the heating coil is 1.7 MBh (0.49 kW).

Application considerations

- Certain zones in a VAV system typically require tempering, even when high sensible-cooling loads exist elsewhere. To curb operating costs, consider on-site recovered heat (discussed on p. 66) for supply-air tempering.
- If the VAV control strategy includes supply-air-temperature reset, provide a humidity sensor to regulate the humidity in the space. Raising the supply-air

Figure 62. Dehumidification performance of a VAV system with supply-air tempering at VAV terminals



temperature not only increases supply airflow *but also the energy consumption of the fans*. The increase in fan energy often exceeds the cooling and tempering energy saved as a result of resetting the supply-air temperature.

- When using electric heating coils, comply with the manufacturer's guidelines for minimum airflow limits across the heating elements to assure safe operation.

Options for supply-air tempering include radiant heaters in the space, heating coils mounted on the VAV terminals, fan-powered VAV terminals, and dual-duct air distribution.

Heating coils at VAV terminals

Doesn't ASHRAE Standard 90.1–2001 prohibit *the use of new-energy "reheat" in VAV terminals?*

Not necessarily. Section 6.3.2.3 (see p. 8 in this manual) defines several *exceptions* for which new-energy reheat is permitted. Exception A in the standard permits the use of new energy for reheat after the supply airflow is reduced to a defined limit.

The minimum airflow setting for most zones in a VAV system is less than the limits defined by this section of Standard 90.1. ■

It is common to think of electricity, hot water, or steam as a source of heat for supply-air tempering performed by a heating coil at the VAV terminal. However, recovering sensible heat from elsewhere in the HVAC system reduces operating costs. For example, the sensible heat collected from the condenser of a water-cooled chiller is easily distributed to VAV terminals throughout the building.

Condenser-water heat recovery is particularly well-suited for supply-air tempering applications: It provides the relatively small amount of heat needed for tempering and allows the primary heating equipment (boilers, for example) to be turned off during the summer. Any water-cooled chiller can be used to provide sensible heat for supply-air tempering. Examples of common system configurations were discussed in the previous chapter; see pp. 53–56.

Fan-powered VAV terminals

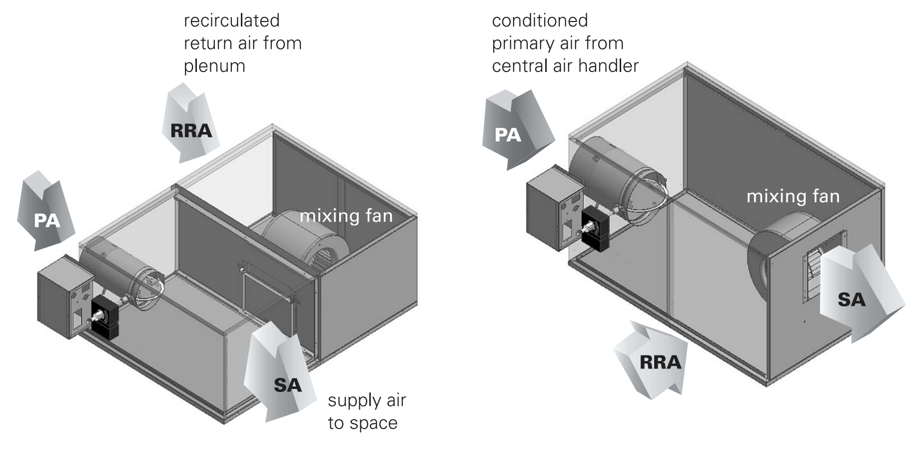
When return air from the space passes through an open ceiling plenum, it collects heat from the lights and roof. A fan-powered VAV terminal mixes this warm plenum air with cold primary air to provide local tempering at low-load conditions. Depending on the application, the plenum air may be warm enough to reduce or eliminate the need for a supplemental heating coil.

There are two types of fan-powered VAV terminals: "parallel" and "series." These classifications describe the arrangement of the fan in the VAV terminal relative to the primary-air fan in the central air handler (Figure 63).

The small, constant-volume fan in a **parallel, fan-powered VAV terminal** is situated in the local recirculated-return-air path, parallel to the primary-air fan. As the cooling load in the space decreases, the central air handler delivers less primary air to the VAV terminal. The small fan in the VAV terminal only operates when the primary airflow drops to the minimum airflow setting. Mixing recirculated return air from the plenum with the cool primary air increases the total airflow delivered to the space and raises the supply-air temperature.

Dehumidifying with Variable-Volume Mixed Air

Figure 63. Fan-powered VAV terminals



parallel, fan-powered VAV terminal

series, fan-powered VAV terminal

The slightly larger, constant-volume fan in a **series, fan-powered VAV terminal** is positioned in the local supply-air path so that it is in series with the primary-air fan. Unlike the parallel configuration, the fan in a series, fan-powered VAV terminal operates continuously when the space is occupied. The fan draws air from both the primary air stream and the plenum to supply the space with a constant volume of air at all times. As the cooling load in the space decreases, the central air handler delivers less primary air to the VAV terminal. To maintain a constant supply airflow, the VAV terminal draws in more recirculated return air from the plenum. A heating coil can provide supplemental supply-air tempering if the cooling load drops below the VAV terminal's minimum primary-airflow setting.

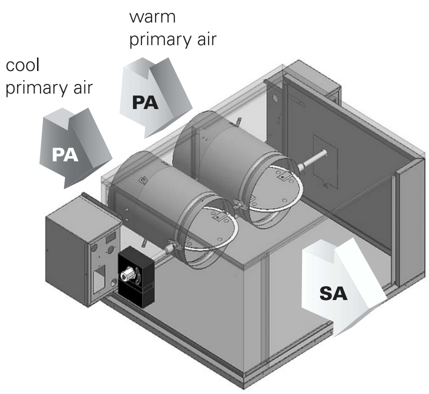
A system that uses series, fan-powered VAV terminals does a better job of dehumidifying the space at part-load conditions than systems equipped with other types of VAV terminals. Series, fan-powered VAV terminals require more primary airflow (PA) to offset the warm return air (RRA), which is recirculated to provide the space with a constant supply airflow. More of the dry primary air results in lower space humidity.

Dual-duct air distribution

VAV systems that are designed for dual-duct air distribution also temper supply air at part-load conditions by mixing warm air with cold primary air. Instead of using recirculated return air from the plenum, the VAV terminal receives warm primary air through separate ductwork. Two modulating devices, one for each air stream, control the amount of cool and/or warm primary air that enters the

Dehumidifying with Variable-Volume Mixed Air

Figure 64. Dual-duct VAV terminal

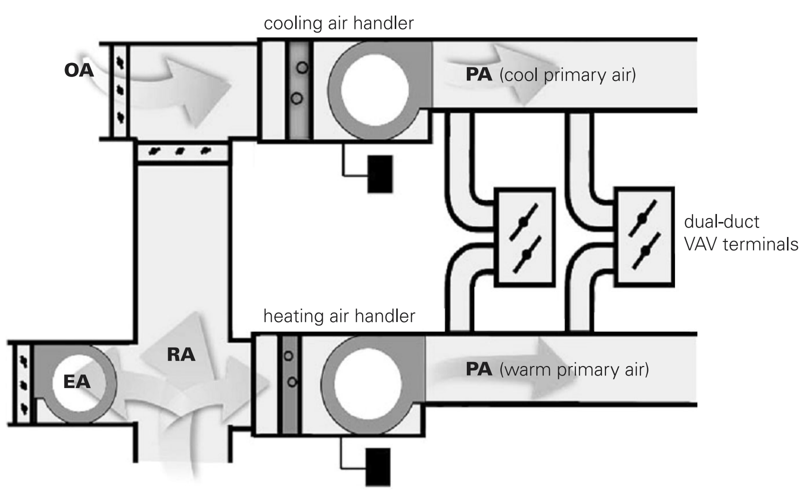


VAV terminal (Figure 64). The primary air streams mix inside the VAV terminal and are then delivered to the space (Figure 65).

As the cooling load in the space decreases, the modulation device that controls the cool primary air modulates toward its minimum-open position. When the cooling load drops to the point where the required amount of cool primary air is less than the VAV terminal's minimum airflow setting, the second modulation device begins to open. This allows warm primary air to mix with and temper the cool primary air before it is supplied to the space.

With further decreases in the cooling load, the space will eventually require heating. To offset an increasing heating load, the VAV terminal mixes the minimum amount of cool primary air with ever-increasing amounts of warm primary air. When the heating load becomes large enough, the recirculated return air is heated before it is delivered to the VAV terminals as warm primary air.

Figure 65. VAV system with dual-duct air distribution



Humidity Control during Unoccupied Periods

Buildings that are served by VAV systems may require an after-hours source of "reheat" energy — for example, heating coils at the VAV terminals or warm plenum air if the VAV terminals are fan-powered. If the VAV system includes an air handler that exclusively conditions the outdoor air, then the dedicated outdoor-air handler also can provide after-hours dehumidification. (The next chapter discusses dedicated outdoor-air systems in detail.)

Dehumidifying with Variable-Volume Mixed Air

Building Pressurization

Consult *Building Pressurization Control,* Trane applications engineering manual AM-CON-17, for information about regulating building pressure through design and control of the HVAC system. ■

For most buildings, the difference between indoor and outdoor static pressures results directly from the combined effect of continuously changing conditions: weather ("stack effect"), wind, and operation of the mechanical ventilation system (local exhaust fans, airside economizer). Maintaining the desired pressure difference, even during normal daytime operation, is particularly challenging in VAV applications because the supply airflow also changes. Preventing both infiltration and economizer-induced overpressurization requires a control strategy that directly controls building pressure. Such strategies monitor building pressure and then modulate relief airflow accordingly — by either adjusting the capacity of the relief fan or the position of the relief damper — to maintain the desired pressure difference across the building envelope.

Airside Economizing

Climate, hours of occupancy, and potential savings in operating cost usually influence the choice between methods of economizer control. In most VAV applications, however, the *differential (comparative) enthalpy* economizer best balances dehumidification performance and operating-cost savings. As its name implies, comparative enthalpy control compares the enthalpy of the outdoor air to the enthalpy of the recirculated return air. When the outdoor air has a lower enthalpy than the return air, the outdoor-air damper fully opens. This strategy reduces cooling-energy consumption. Because the air that passes through the cooling coil in a VAV application is always dehumidified to the same low dew point, it avoids introducing unwanted moisture into the space.

Note: If the VAV system includes both an airside economizer and supply-air-temperature reset, make sure that the control scheme will not introduce humid outdoor air into the space.

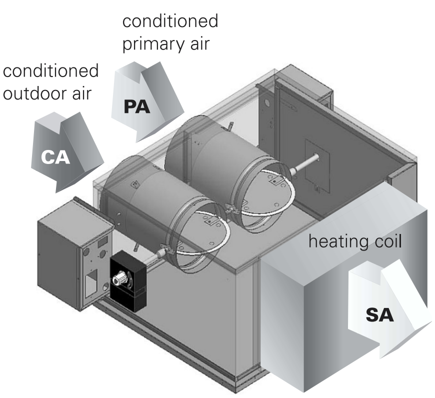
Dehumidifying with Variable-Volume Mixed Air

Improving Dehumidification Performance

VAV systems can provide effective *coincidental* dehumidification over a wide range of indoor load conditions; but the basic design of the system can be altered to enhance dehumidification performance.

Condition Outdoor Air Separately

Figure 66. Dual-duct VAV terminal used with dedicated outdoor air



One way to improve indoor humidity control is to separately treat the outdoor air before mixing it with recirculated return air. A dedicated air handler cools and dehumidifies all of the outdoor air to a dew point that is drier (lower) than the space. The conditioned outdoor air then is delivered to one or more VAV air handlers, or directly to the individual, dual-duct VAV (Figure 66) terminals that serve each zone. The "ventilation" damper in the dual-duct VAV terminal maintains the required quantity of outdoor air from the dedicated outdoor-air unit, while the "primary-air" damper regulates the 100 percent-recirculated return air from the VAV air handler.

To demonstrate how separately conditioning the outdoor air affects dehumidification performance, let's revisit the example classroom. Assume that the dedicated outdoor-air handler supplies 450 cfm (0.21 m^3/s) of outdoor air to the dual-duct VAV terminal serving the classroom. The outdoor air is cooled and dehumidified to 52°F DP (11.1°C DP) and delivered — without tempering or reheat — to the "ventilation" damper in the dual-duct terminal.

At the full-load, peak dry-bulb condition (Figure 67), the VAV air handler delivers 1,050 cfm (0.5 m^3/s) of 57.3°F (14.1°C), primary air to the second damper in the dual-duct VAV terminal. Inside the VAV terminal, primary air (PA) mixes with conditioned outdoor air (CA); the resulting supply air yields a 50-percent relative humidity in the classroom. The cooling-coil load is 3.1 tons (10.8 kW)

Figure 67. Dehumidification performance of a VAV system with separately conditioned outdoor air at peak dry-bulb condition

Space sensible-cooling load offset by dedicated outdoor-air handler:

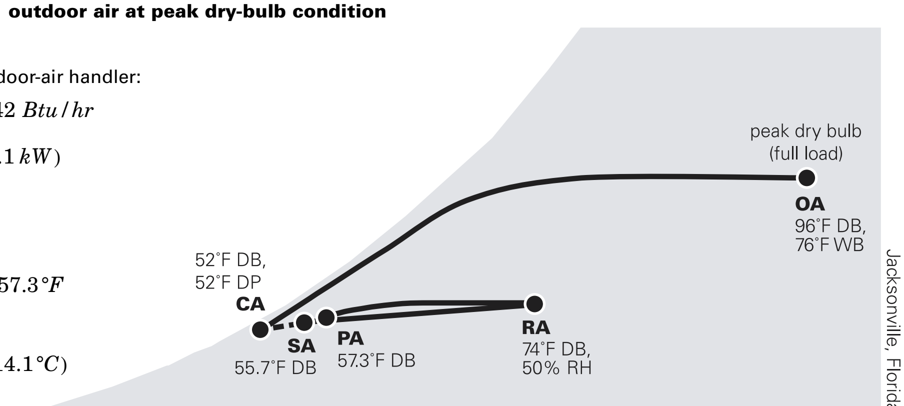
$$Q_{s,doa} = 1.085 \times 450\ cfm \times (74 - 52\,°F) = 10,742\ Btu/hr$$

$$(Q_{s,doa} = 1.21 \times 0.21\ m^3/s \times [23.3 - 11.1\,°C] = 3.1\ kW)$$

Recirculating air handler at peak dry-bulb condition:

$$Q_s = 29,750 - 10,742\ Btu/hr$$
$$= 1.085 \times 1,050\ cfm \times (74\,°F - T_{pa}) \therefore T_{pa} = 57.3\,°F$$

$$(Q_s = 8.7 - 3.1\ kW)$$
$$(= 1.21 \times 0.5\ m^3/s \times [23.3\,°C - T_{pa}] \therefore T_{pa} = 14.1\,°C)$$



Dehumidifying with Variable-Volume Mixed Air

Figure 68. Dehumidification performance of a VAV system with separately conditioned outdoor air at peak dew-point condition

Space sensible-cooling load offset by dedicated outdoor-air handler:

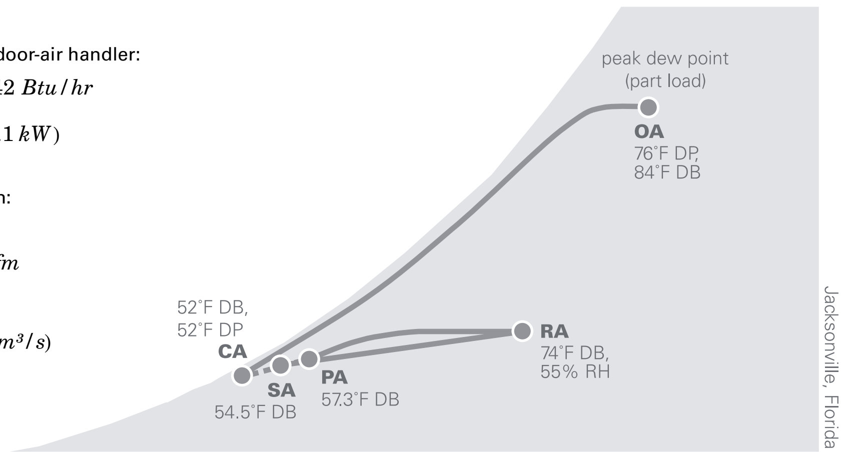
$$Q_{s,doa} = 1.085 \times 450\, cfm \times (74 - 52\,°F) = 10,742\, Btu/hr$$

$$(Q_{s,doa} = 1.21 \times 0.21\, m^3/s \times [23.3 - 11.1\,°C] = 3.1\, kW)$$

Recirculating air handler at peak dew-point condition:

$$Q_s = 18,750 - 10,742\, Btu/hr$$
$$= 1.085 \times V_{pa} \times (74 - 57.3\,°F) \therefore V_{pa} = 392\, cfm$$

$$(Q_s = 5.2 - 3.1\, kW)$$
$$(= 1.21 \times V_{pa} \times [23.3 - 14.1\,°C] \therefore V_{pa} = 0.18\, m^3/s)$$



for the dedicated outdoor-air handler and 1.8 tons (6.2 kW) for the VAV air handler.

Less sensible cooling is required at the *part-load, peak dew-point condition* (Figure 68). Therefore, while the dual-duct VAV terminal receives the same amount of 52°F DP (11.1°C DP), conditioned outdoor air, it receives only 392 cfm (0.18 m³/s) of primary air from the VAV air handler. The combined airflow, 842 cfm (0.40 m³/s), yields a relative humidity of 55 percent in the classroom. This time, the cooling-coil load is 3.4 tons (12.1 kW) for the dedicated outdoor-air handler and 0.7 tons (2.6 kW) for the VAV air handler.

On a mild, rainy day (Figure 69, p. 72), the VAV terminal mixes 450 cfm (0.21 m³/s) of conditioned outdoor air with only 83 cfm (0.04 m³/s) of primary air, which results in a relative humidity of 58 percent. In this application, the minimum airflow for the VAV terminal is only 450 cfm (0.21 m³/s), which equals the ventilation requirement for the classroom.

Cooling-coil loads are 2.0 tons (7.2 kW) for the dedicated outdoor-air handler and 0.2 tons (0.7 kW) for the VAV air handler.

Dehumidifying with Variable-Volume Mixed Air

Figure 69. Dehumidification performance of a VAV system with separately conditioned outdoor air on a mild, rainy day

Space sensible-cooling load offset by dedicated outdoor-air handler:

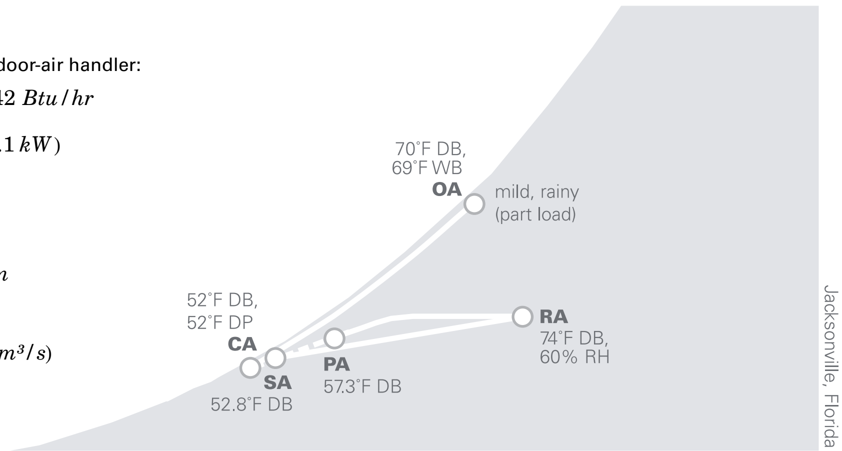
$$Q_{s,doa} = 1.085 \times 450\ cfm \times (74 - 52\,°F) = 10,742\ Btu/hr$$

$$(Q_{s,doa} = 1.21 \times 0.21\ m^3/s \times [23.3 - 11.1\,°C] = 3.1\ kW)$$

Recirculating air handler on a mild, rainy day:

$$Q_s = 12,250 - 10,742\ Btu/hr$$
$$= 1.085 \times V_{pa} \times (74 - 57.3\,°F) \therefore V_{pa} = 83\ cfm$$

$$(Q_s = 3.6 - 3.1\ kW)$$
$$(= 1.21 \times V_{pa} \times [23.3 - 14.1\,°C] \therefore V_{pa} = 0.04\ m^3/s)$$



For more information about the "multiple-space" equation from ASHRAE Standard 62 and how it applies to traditional, mixed-air VAV systems, see "The Threefold Challenge of Ventilating Single-Duct VAV Systems" in Trane *Engineers Newsletter* ENEWS-27/1. It is archived in the "newsletters" section of www.trane.com/commercial. ■

Application considerations

- In a traditional, mixed-air VAV system, the multiple-space equation (Equation 6–1) from ASHRAE Standard 62–2001 requires that the VAV air handler bring in more outdoor air than the sum of the space ventilation requirements. A dedicated-outdoor-air design delivers conditioned outdoor air directly to individual VAV terminals (or spaces) and, therefore, is not considered to be a multiple-space, recirculating ventilation system. For this reason, a dedicated outdoor-air system requires less total outdoor airflow than a traditional VAV design.
- Using a dual-duct VAV terminal in this manner typically requires the addition of a reheat coil for tempering at low cooling loads. (Eventually, the space sensible-cooling load decreases to the point that the primary-air damper closes. The cold, conditioned outdoor air then must be tempered to avoid overcooling the space.)
- As an alternative to separately conditioning the outdoor air in a VAV system, modify the fan-powered VAV terminals by adding a second "ventilation" damper. The modified VAV terminals operate similarly to dual-duct VAV terminals, but the "free" heat from the plenum can be used to temper the conditioned outdoor air when the space cooling load is low.

The next chapter, "Dehumidifying with Dedicated Outdoor Air," discusses system configurations, design procedures, and application considerations for air distribution systems that separately treat outdoor air.

Dehumidifying with Variable-Volume Mixed Air

Deliver Colder Supply Air

Peak dry-bulb condition:

$$Q_s = 1.085 \times V_{sa} \times (74 - 50°F)$$
$$= 29{,}750\ Btu/hr \therefore V_{sa} = 1{,}142\ cfm$$

$$(Q_s = 1.21 \times V_{sa} \times [23.3 - 10°C])$$
$$(= 8.7\ kW \therefore V_{sa} = 0.54\ m^3/s)$$

Peak dew-point condition:

$$Q_s = 1.085 \times V_{sa} \times (74 - 50°F)$$
$$= 17{,}850\ Btu/hr \therefore V_{sa} = 685\ cfm$$

$$(Q_s = 1.21 \times V_{sa} \times [23.3 - 10°C])$$
$$(= 5.2\ kW \therefore V_{sa} = 0.32\ m^3/s)$$

Mild, rainy day:

$$Q_s = 1.085 \times 550\ cfm \times (74°F - T_{supply})$$
$$= 12{,}250\ Btu/hr \therefore T_{supply} = 53.5°F$$

$$(Q_s = 1.21 \times 0.26\ m^3/s \times [23.3°C - T_{supply}])$$
$$(= 3.6\ kW \therefore T_{supply} = 11.9°C)$$

Lowering the leaving-air temperature for the central cooling coil condenses more moisture from the supply air and requires less airflow to offset the sensible-cooling load in the space.

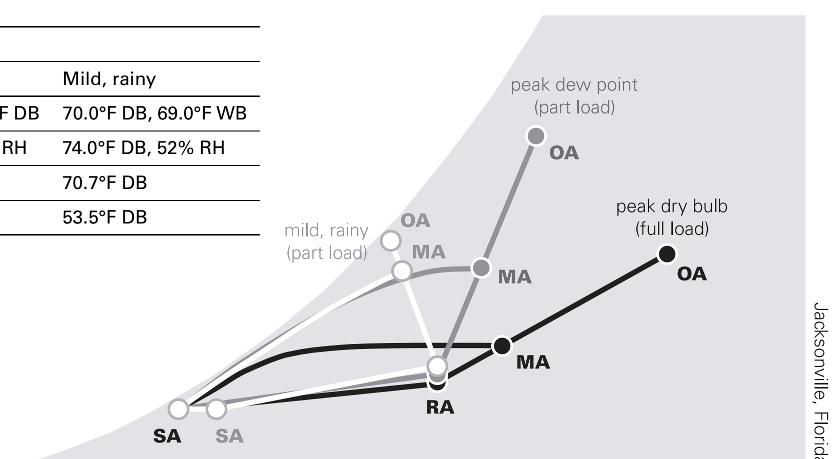
At the peak dry-bulb condition (Figure 70), delivering supply air to the example classroom at 50°F (10°C) instead of 55.7°F (13.1°C) reduces the required airflow from 1,500 cfm (0.7 m^3/s) to 1,142 cfm (0.54 m^3/s). Supplying the classroom with colder, drier supply air also reduces the relative humidity from 52 percent to 47 percent. The required cooling-coil capacity is 5.1 tons (17.8 kW).

The classroom needs less of the constant-temperature supply air *at the peak dew-point condition* because the sensible-cooling load is smaller. To match the load in the space, the VAV system reduces the supply airflow to 685 cfm (0.32 m^3/s). The resulting relative humidity is 50 percent, and the required cooling capacity is 4.3 tons (15.1 kW).

On a mild, rainy day, the amount of supply air needed to offset the classroom's sensible-cooling load is less than the VAV terminal's minimum airflow setting; local tempering is required to avoid overcooling. As in the example at the beginning of this chapter, the minimum airflow setting is 550 cfm (0.26 m^3/s). The VAV terminal must temper the air to 53.5°F (11.9°C) before delivering it to the space. The resulting relative humidity in the classroom is 52 percent. The cooling-coil load is 2.5 tons (8.8 kW), while the heating-coil at the VAV terminal is 2.1 MBh (0.6 kW).

Figure 70. Effect of "cold" supply air on dehumidification performance of a VAV system

Design condition	Full load	Part load	
	Peak dry bulb	Peak dew point	Mild, rainy
OA	96.0°F DB, 76.0°F WB	76.0°F DP, 84.0°F DB	70.0°F DB, 69.0°F WB
RA	74.0°F DB, 47% RH	74.0°F DB, 50% RH	74.0°F DB, 52% RH
MA	82.7°F DB	80.6°F DB	70.7°F DB
SA	50.0°F DB	50.0°F DB	53.5°F DB



Dehumidifying with Variable-Volume Mixed Air

A previous issue of the Trane *Engineers Newsletter,* titled "Cold Air Makes Good $ense" (ENEWS-29/2), discusses the benefits and design considerations associated with cold-air VAV systems. To read it, visit the "newsletters" section of www.trane.com/commercial.

The *Cold Air Distribution System Design Guide* (ISBN 1-883413-37-0), by Allan T. Kirkpatrick and James S. Elleson, is another useful reference. It is available from ASHRAE's online bookstore at www.ashrae.org. ■

Application considerations

- In addition to drier spaces, other benefits of cold-air systems include smaller air handlers, VAV terminals, and ducts, as well as less supply-fan energy.
- Increased reheat energy and fewer hours of airside economizer operation partially offset the supply-fan energy savings. Intelligent system control is crucial to fully realize the potential savings.
- Some designers raise the space thermostat setpoint to further reduce the supply airflow in cold air systems. This decision increases the indoor dew point, which may negate the benefit of otherwise drier spaces.
- Applied, chilled water systems typically work best for "cold air" distribution because the designer can match the design requirements for airflow and cooling capacity. Packaged, direct-expansion (DX) systems defer many design decisions to the manufacturer, which reduces the initial cost; however, the limitations of a fixed design may make it difficult to achieve the desired "cold coil" temperature. ■