Dehumidifying with Dedicated Outdoor Air

System Configurations

Figure 71. Configurations for dedicated outdoor-air systems
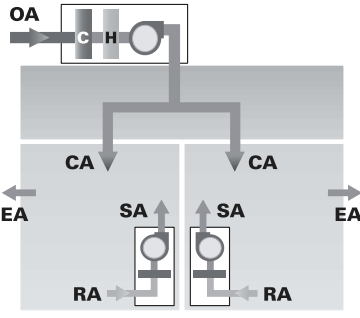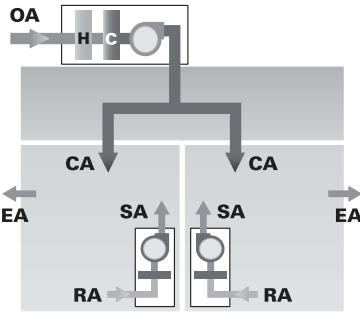
dry-bulb temperature

destination	neutral **CA** to space	cold **CA** to space
	neutral **CA** to (or near) other equipment	cold **CA** to other equipment

A dedicated outdoor-air handler separately filters, cools, dehumidifies, heats, and/or humidifies all ventilation (outdoor) air. It also can enable the direct control of indoor humidity. There are several ways to lay out a constant- or variable-air-volume HVAC system for separate treatment of ventilation air (Figure 71). These configurations are based on two factors:

■ *Where* the conditioned outdoor air is delivered — directly to occupied spaces, to air terminals, or to other types of HVAC equipment

■ The *dry-bulb temperature* of the conditioned air (CA) — *neutral,* which approximates room air, or *cold,* which usually approximates conventional, 55°F (13°C) supply air.

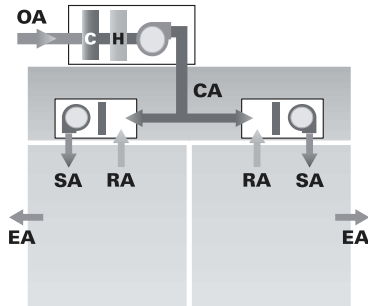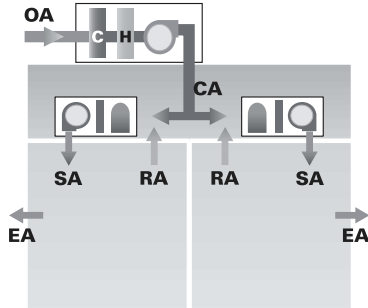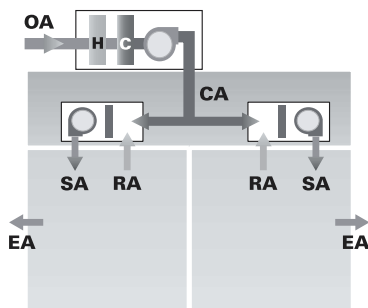Table 8 summarizes the traits of each configuration.

Table 8. Configurations for HVAC systems with dedicated outdoor-air treatment

System schematic	Typical characteristics	Application considerations
Neutral CA to space OA C H CA CA EA SA SA EA RA RA	Dedicated outdoor-air handler: ■ Offsets latent- and sensible-cooling loads resulting from ventilation (outdoor air) ■ Cools and dehumidifies the outdoor air so that its dew point is lower than that of room air ■ Reheats conditioned outdoor air to a neutral dry-bulb temperature ■ Delivers conditioned outdoor air (via ductwork) directly to the space Local HVAC units: ■ Are installed within the occupied space ■ Offset local sensible-cooling and heating loads	■ Provides design flexibility by accommodating any type of terminal device, plus horizontal and vertical discharge arrangements ■ Well-suited for applications with coincidental heating and cooling needs because it has little effect on space loads ■ Commonly applied with in-room HVAC equipment, such as fan–coils, through-the-wall air conditioners, and classroom unit ventilators
Cold CA to space OA H C CA CA EA SA SA EA RA RA	Dedicated outdoor-air handler: ■ Offsets latent- and sensible-cooling loads resulting from ventilation (outdoor air) ■ Cools and dehumidifies the outdoor air so that dry-bulb and dew-point temperatures are lower than room air ■ Does not reheat the conditioned outdoor air ■ Delivers conditioned outdoor air (via ductwork) directly to the space Local HVAC units: ■ Are installed within the occupied space ■ Offset local sensible-cooling and heating loads	■ Typically applied with in-room HVAC equipment; also used with radiant ceiling panels and other non-traditional devices ■ Requires less cooling capacity from the terminal devices, which permits downsizing, but also may create heating loads in spaces that require little or no cooling ■ To prevent uncomfortable drafts, select high-performance diffusers that induce room air to mix with (and warm) the cold conditioned air before it reaches occupants ■ Adding even a small amount of heat to the conditioned air after it leaves the cooling coil will increase the cooling capacity needed from the terminal devices

continued on next page

Dehumidifying with Dedicated Outdoor Air

Table 8. Configurations for HVAC systems with dedicated outdoor-air treatment (continued)

System schematic	Typical characteristics	Application considerations
Neutral CA to other equipment…	Dedicated outdoor-air handler: ■ Offsets latent- and sensible-cooling loads resulting from ventilation (outdoor air) ■ Cools and dehumidifies the outdoor air ■ Reheats conditioned outdoor air to a neutral dry-bulb temperature ■ Delivers conditioned outdoor air (via ductwork) to outdoor-air inlets of local units Local HVAC units: ■ Are installed outside the occupied space in ceiling plenums, closets, or equipment rooms ■ Mix recirculated return air with conditioned outdoor air inside the unit casing ■ Offset local sensible-cooling and heating loads	■ Commonly applied with central air handlers (chilled water or DX) or local HVAC terminals, such as blower–coils, packaged rooftop air conditioners, vertical self-contained air conditioners, and dual-duct VAV terminals ■ Well-suited for applications with coincidental heating and cooling needs because it has little effect on space loads
…or to plenum (near other equipment)	Dedicated outdoor-air handler: ■ Offsets latent- and sensible-cooling loads resulting from ventilation (outdoor air) ■ Cools and dehumidifies the outdoor air ■ Reheats conditioned outdoor air to a neutral dry-bulb temperature ■ Delivers conditioned outdoor air (via ductwork) near return-air inlets of local units Local HVAC units: ■ Are installed in the ceiling plenum ■ Offset local sensible-cooling and heating loads ■ Are without mixing boxes, so conditioned outdoor air and recirculated return air mix in the ceiling plenum	■ Typically applied with water-source heat pumps, horizontal fan–coils, and other local HVAC units without mixing boxes ■ Requires ducts to deliver conditioned outdoor air within 5 ft (1.5 m) of local HVAC units ■ Requires a local or powered central exhaust system ■ To prevent condensation on beams, ducts, and other surfaces, assure that the CA dry-bulb temperature exceeds the plenum-air dew point
Cold CA to other equipment	Dedicated outdoor-air handler: ■ Offsets latent- and sensible-cooling loads resulting from ventilation (outdoor air) ■ Cools and dehumidifies outdoor air ■ Does *not* reheat the conditioned outdoor air ■ Delivers conditioned outdoor air (via ductwork) to outdoor-air inlets of local units Local HVAC units: ■ Are installed outside the occupied space ■ Mix recirculated return air with conditioned outdoor air inside the unit casing ■ Offset sensible-cooling and heating loads in the space	■ Typically applied with blower–coils, packaged rooftop air conditioners, central air handlers, self-contained air conditioners, and dual-duct VAV terminals ■ Spaces that need little or no cooling may require local reheat ■ Requires less cooling capacity from terminal devices, which permits downsizing ■ Avoids reheat at the dedicated outdoor-air handler

Dehumidifying with Dedicated Outdoor Air

Design Objectives for Conditioned Outdoor Air

Moisture Content

Appendix B in this manual describes how to determine the conditioned-air dew point that will allow the dedicated outdoor-air handler to offset the local latent cooling loads. ∎

Regardless of where the conditioned outdoor air is delivered, the dedicated outdoor-air handler should make the incoming outdoor air drier than the air in the space. Delivering drier (low-dew-point) air eliminates the sensible- and latent-cooling loads associated with ventilation. If the dew-point temperature of the conditioned outdoor air is lower than the dew point in the space, it will also offset the local latent-cooling loads. Consequently, the HVAC terminals only need to offset the local sensible-cooling loads.

Delivering low-dew-point, conditioned air can greatly improve the dehumidification performance of constant-volume systems, particularly at part-load conditions. (See " 'Direct' Control of Humidity," pp. 44–46.) It also allows the HVAC terminals to "run dry" under most conditions: With the dedicated outdoor-air handler offsetting the local latent loads, the space dew point can be kept below the dew point of the air supplied by the local terminals. Little or no moisture condenses on local cooling coils and collects in drain pans, perhaps eliminating one source of moisture within the space and the HVAC system.

Why not condition the outdoor air to a neutral dry-bulb temperature without overcooling it to obtain a low dew point? To demonstrate the effect of this design choice, let's revisit the 30-occupant classroom in Jacksonville, Florida, which was introduced on p. 19. For this example, the HVAC system includes a dedicated outdoor-air handler that cools the outdoor air to a neutral dry-bulb temperature of 74°F (23.3°C) *without overcooling it;* the conditioned air is then delivered directly to the classroom, where a fan–coil provides the local air conditioning.

Although the outdoor-air handler eliminates the *sensible*-cooling load associated with ventilation, it only offsets part of the latent-cooling load because the dew point of the conditioned outdoor air is still much higher than the dew point in the space. The remaining moisture in the conditioned outdoor air must be removed from the classroom by the fan–coil. The fan–coil also must offset the entire *space* cooling load, both sensible and latent, because the "neutral" outdoor air does not provide a cooling effect.

At the full-load, peak dry-bulb condition, the fan–coil must supply air at 55.7°F DB (13.1°C DB) to maintain the classroom at 74°F DB (23.3°C DB). The psychrometric analysis in Figure 72 (p. 78) illustrates that delivering the outdoor air at a neutral dry-bulb temperature, *but humid with respect to the*

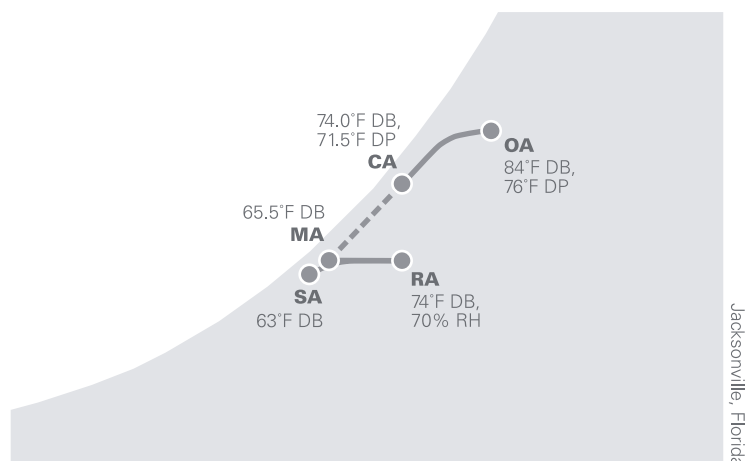# Dehumidifying with Dedicated Outdoor Air

Figure 72. Dehumidification performance of neutral-temperature conditioned air, without overcooling, at peak dry-bulb condition
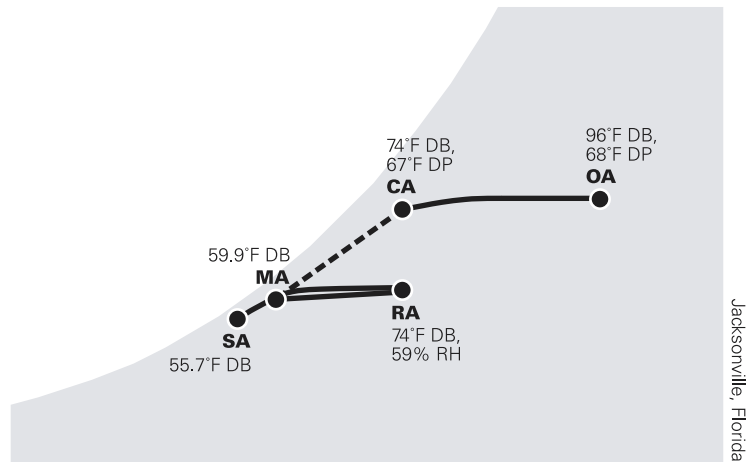
space, yields a relative humidity of 59 percent. Contrast this performance with the 52 percent relative humidity that the basic constant-volume system provides *without* a dedicated outdoor-air handler (p. 20).

At the part-load, peak dew-point condition, the dedicated outdoor-air handler again cools the outdoor air to 74°F DB (23.3°C DB). Due to the smaller sensible-cooling load, the fan–coil supplies the classroom with warmer air, 63°F (17.2°C), to avoid overcooling. Although the dry-bulb target is maintained, the relative humidity increases to 70 percent (Figure 73).

Figure 73. Dehumidification performance of neutral-temperature conditioned air, without overcooling, at part-load, peak dew-point condition

Dehumidifying with Dedicated Outdoor Air

Dedicated outdoor-air handler:

$$Q_s = 1.085 \times 450 \; cfm \times (74 - 70\,°F)$$
$$= 1,953 \; Btu/hr$$

$$(Q_s = 1.21 \times 0.21 \; m^3/s \times [23.3 - 21.2\,°C])$$
$$(= 0.53 \; kW)$$

Fan–coil:

$$Q_s = 12,250 - 1,953 \; Btu/hr$$
$$= 1.085 \times 1,500 \; cfm \times (74\,°F - T_{supply})$$
$$\therefore T_{supply} = 67.7\,°F$$

$$(Q_s = 3.6 - 0.53 \; kW)$$
$$(= 1.21 \times 0.7 \; m^3/s \times [23.3\,°C - T_{supply}])$$
$$(\therefore T_{supply} = 19.8\,°C)$$

Figure 74. Dehumidification performance of neutral-temperature conditioned air, without overcooling, on a mild and rainy day
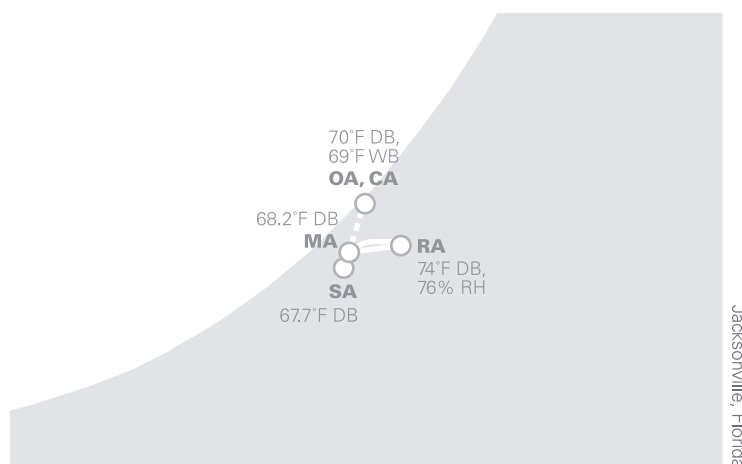
Dehumidification performance is even worse *on a mild, rainy day*. The cooling coil in the dedicated outdoor-air handler shuts off because the 70°F DB (21.2°C DB) outdoor air is cooler than the thermostat setpoint. The outdoor air offsets a small portion of the classroom's sensible-cooling load, so the fan–coil only cools the recirculated return air to 67.7°F DB (19.8°C DB). As a result, the relative humidity in the classroom increases even more, to 76 percent (Figure 74).

The preceding examples demonstrate that supplying conditioned air at a neutral dry-bulb temperature, without subcooling it to reduce its moisture content, provides less dehumidification than a system without a dedicated outdoor-air handler. In addition to lowering the sensible-heat ratio in the space, eliminating only the sensible-cooling load from the outdoor air makes it difficult for the local units to provide adequate coincidental dehumidification. Avoid designing dedicated outdoor-air systems that operate in this manner if the outdoor dew point is greater than the desired dew point indoors for any significant amount of time.

Dehumidifying with Dedicated Outdoor Air

Dry-Bulb Temperature

Whether the dedicated outdoor-air handler delivers conditioned, *dry* air directly to the space, to HVAC terminals, or to other air handlers, it can be designed to condition the outdoor air to a neutral or cold dry-bulb temperature.

Dry, neutral-temperature air. A dedicated outdoor-air handler that supplies dry, neutral-temperature air (Figure 75) initially cools and dehumidifies the outdoor air; the resulting dew point usually ranges from 40°F (4°C) to 60°F (16°C), depending on the latent load in the space. The same unit then reheats the dehumidified outdoor air to approximately room temperature, usually between 70°F DB and 75°F DB (21°C DB and 24°C DB), before discharging it.

The appropriate dry-bulb temperature for a particular application best balances the following concerns:
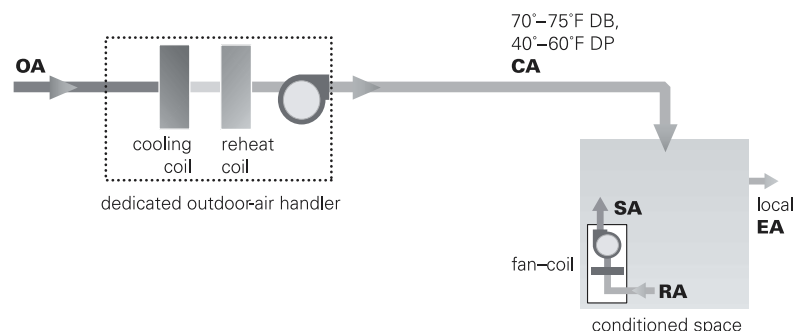
■ Energy required to reheat the conditioned outdoor air

■ Cooling effect of the conditioned air on the sizing and operation of local HVAC terminals

■ Heating energy used by the local HVAC terminals at part-load conditions

Figure 76 shows the *peak dry-bulb conditions* that exist when the classroom receives 450 cfm (0.21 m³/s) of conditioned outdoor air. In this case, the dedicated outdoor-air handler cools and dehumidifies the outdoor air to 52°F DP (11.1°C DP), which offsets the latent cooling load in the classroom and imposes a load of 3.0 tons (10.6 kW) on the cooling coil. This unit then reheats the air to 71°F DB (21.7°C DB) before discharging it directly into the classroom; the reheat load is 9.3 MBh (2.7 kW).

Because the dry-bulb temperature of the conditioned outdoor air approximates that of the classroom — 74°F (23.3°C) in this example — the cooling effect is minimal. To offset the classroom's remaining sensible-cooling load, the fan–coil

Appendix B in this manual describes how to determine the conditioned-air dew point that will allow the dedicated outdoor-air handler to offset the local latent cooling loads. This appendix also explains how to size the local HVAC terminals. ■

Figure 75. Dedicated outdoor-air handler that supplies dry, neutral-temperature air

Dehumidifying with Dedicated Outdoor Air

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

$$Q_s = 1.085 \times 450 \; cfm \times (74 - 71\,°F)$$

$$= 1,465 \; Btu/hr$$

$$(Q_s = 1.21 \times 0.21 \; m^3/s \times [23.3 - 21.7\,°C])$$

$$(= 0.41 \; kW)$$

Fan–coil at peak dry-bulb condition:

$$Q_s = 29,750 - 1,465 \; Btu/hr$$

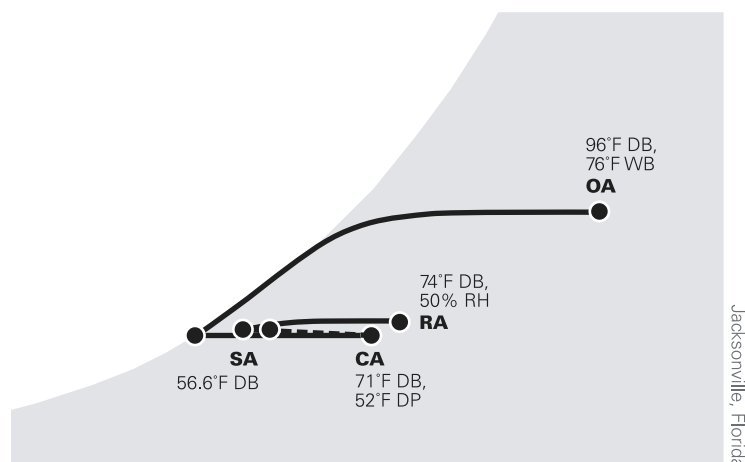$$= 1.085 \times 1,500 \; cfm \times (74\,°F - T_{supply})$$

$$\therefore T_{supply} = 56.6\,°F$$

$$(Q_s = 8.7 - 0.41 \; kW)$$

$$(= 1.21 \times 0.7 \; m^3/s \times [23.3\,°C - T_{supply}])$$

$$(\therefore T_{supply} = 13.5\,°C)$$

Figure 76. Dehumidification performance of dry, neutral-temperature, conditioned outdoor air at the peak dry-bulb condition

cools 1,500 cfm (0.7 m³/s) of recirculated return air to 56.6°F DB (13.5°C DB); this requires 2.8 tons (9.8 kW) of capacity and yields a 50 percent-relative humidity.

At the part-load, peak dew-point condition (Figure 77), the dedicated outdoor-air handler again delivers 71°F DB, 52°F DP (21.7°C DB, 11.1°C DP) air. Although the reheat load remains 9.3 MBh (2.7 kW), the cooling load increases to 3.4 tons (12.0 kW). Meanwhile, fan–coil capacity modulates to raise the supply-air temperature to 63.9°F DB (17.7°C DB) and maintain the thermostat setpoint. The resulting relative humidity is 56 percent, and the cooling load on the fan–coil is 1.4 tons (4.9 kW).

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

$$Q_s = 1.085 \times 450 \; cfm \times (74 - 71\,°F)$$

$$= 1,465 \; Btu/hr$$

$$(Q_s = 1.21 \times 0.21 \; m^3/s \times [23.3 - 21.7\,°C])$$

$$(= 0.41 \; kW)$$

Fan–coil at peak dew-point condition:

$$Q_s = 17,850 - 1,465 \; Btu/hr$$

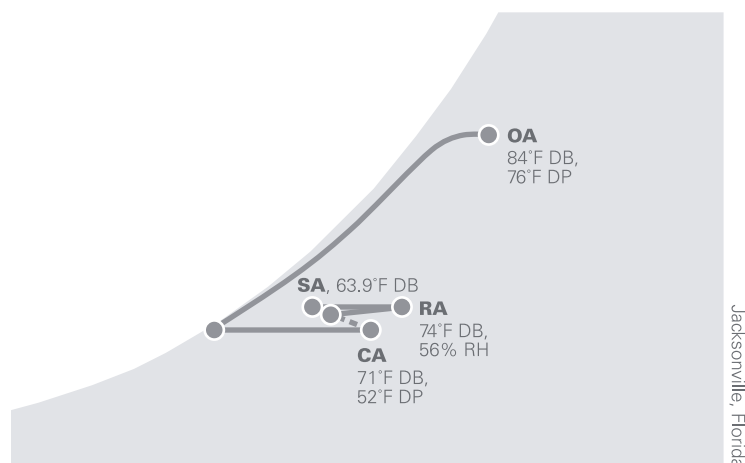$$= 1.085 \times 1,500 \; cfm \times (74\,°F - T_{supply})$$

$$\therefore T_{supply} = 63.9\,°F$$

$$(Q_s = 5.2 - 0.41 \; kW)$$

$$(= 1.21 \times 0.7 \; m^3/s \times [23.3\,°C - T_{supply}])$$

$$(\therefore T_{supply} = 17.7\,°C)$$

Figure 77. Dehumidification performance of dry, neutral-temperature, conditioned outdoor air at the peak dew-point condition

Dehumidifying with Dedicated Outdoor Air

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

$$Q_s = 1.085 \times 450\ cfm \times (74 - 71\,°F)$$
$$= 1,465\ Btu/hr$$

$$(Q_s = 1.21 \times 0.21\ m^3/s \times [23.3 - 21.7\,°C])$$
$$(= 0.41\ kW)$$

Fan–coil on a mild, rainy day:

$$Q_s = 12,250 - 1,465\ Btu/hr$$
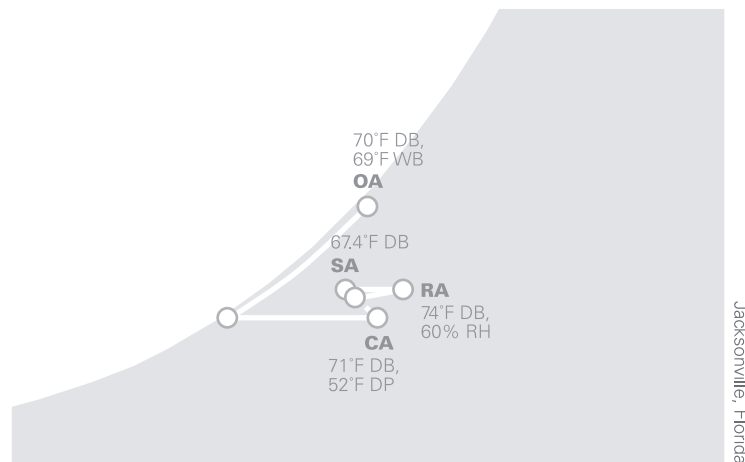$$= 1.085 \times 1,500\ cfm \times (74\,°F - T_{supply})$$
$$\therefore T_{supply} = 67.4\,°F$$

$$(Q_s = 3.6 - 0.41\ kW)$$
$$(= 1.21 \times 0.7\ m^3/s \times [23.3\,°C - T_{supply}])$$
$$(\therefore T_{supply} = 19.7\,°C)$$

Figure 78. Dehumidification performance of dry, neutral-temperature, conditioned outdoor air on a mild, rainy day

70°F DB, 69°F WB
OA

67.4°F DB
SA

RA
74°F DB, 60% RH

CA
71°F DB, 52°F DP

Jacksonville, Florida

On a mild and rainy day (Figure 78, p. 82), the dedicated outdoor-air handler still produces 71°F DB, 52°F DP (21.7°C DB, 11.1°C DP) air; the cooling-coil load decreases to 2.0 tons (7.0 kW) and the reheat load is still 9.3 MBh (2.7 kW). In the classroom, the fan–coil modulates to supply 67.4°F DB (19.7°C DB) air, which corresponds to a cooling load of 0.9 tons (3.2 kW). The resulting relative humidity is 60 percent.

Dry, cold air. In contrast, some dedicated outdoor-air systems deliver air at a *cold* temperature, usually 45°F to 55°F DB (7°C to 13°C DB). These systems cool and dehumidify the outdoor air, but do not reheat it. Typically, the conditioned outdoor air is delivered directly to the mixing box of local air-handling equipment, such as blower–coils, unit ventilators, air handlers, packaged rooftop air conditioners, or dual-duct VAV terminals. Some systems, however, are designed to deliver cold ventilation air directly to occupied spaces.

Using the example classroom, Figure 79 shows the psychrometric analysis of the full-load, peak dry-bulb condition. The dedicated outdoor-air handler cools and dehumidifies the entering air to 52°F DP (11.1°C DP), then delivers the conditioned air to the classroom. The dew point of this conditioned outdoor air is low enough to offset the latent cooling load in the classroom. At the peak dry-bulb condition for Jacksonville, Florida, the cooling-coil load for the dedicated outdoor-air handler is 3.0 tons (10.6 kW). Because the ventilation air is delivered cold, there is no reheat load.

The dry-bulb temperature of the conditioned outdoor air is much cooler than the space, so it provides a greater sensible-cooling effect than dry, neutral-temperature air. It also permits the use of a fan–coil with less cooling capacity. *At the full-load, peak dry-bulb condition,* the fan–coil in the classroom cools

Dehumidifying with Dedicated Outdoor Air

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

$$Q_s = 1.085 \times 450 \ cfm \times (74 - 52 \, °F)$$

$$= 10{,}742 \ Btu / hr$$

$$(Q_s = 1.21 \times 0.21 \ m^3/s \times [23.3 - 11.1 \, °C])$$

$$(= 3.1 \ kW)$$

Fan–coil at peak dry-bulb condition:

$$Q_s = 29{,}750 - 10{,}742 \ Btu / hr$$

$$= 1.085 \times 1{,}050 \ cfm \ \times (74 \, °F - T_{supply})$$

$$\therefore T_{supply} = 57.3 \, °F$$

$$(Q_s = 8.7 - 3.1 \ kW)$$

$$(= 1.21 \times 0.5 \ m^3/s \times [23.3 \, °C - T_{supply}])$$

$$(\therefore T_{supply} = 14.1 \, °C)$$

Figure 79. Dehumidification performance of dry, cold, conditioned outdoor air at the peak dry-bulb condition
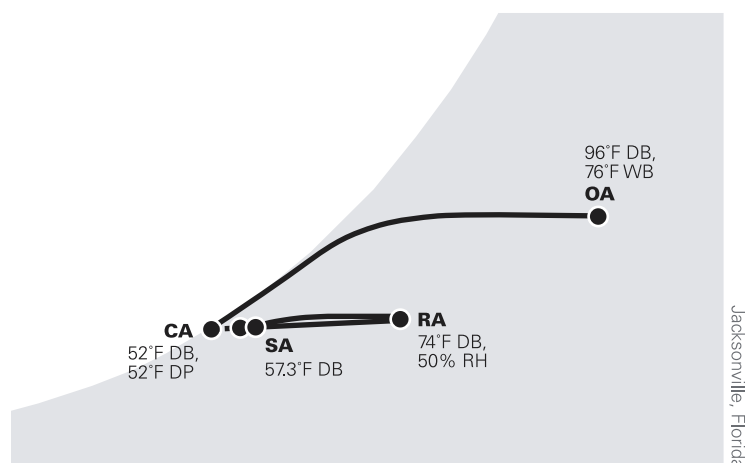
only 1,050 cfm (0.5 m³/s) of recirculated return air to 57.3°F DB (14.1°C DB), which offsets the remaining sensible-cooling load. The cooling load on the fan–coil is 1.8 tons (6.3 kW), and the resulting relative humidity is 50 percent.

At the part-load, peak dew-point condition for the classroom (Figure 80), the dedicated outdoor-air handler delivers the conditioned air, CA, at the same temperature as for the peak dry-bulb condition. Because the outdoor enthalpy is higher, however, the cooling load increases to 3.4 tons (12.0 kW). Meanwhile, the fan–coil modulates to match the thermostat setpoint by raising the supply-

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

$$Q_s = 1.085 \times 450 \ cfm \times (74 - 52 \, °F)$$

$$= 10{,}742 \ Btu / hr$$

$$(Q_s = 1.21 \times 0.21 \ m^3/s \times [23.3 - 11.1 \, °C])$$

$$(= 3.1 \ kW)$$

Fan–coil at peak dew-point condition:

$$Q_s = 17{,}850 - 10{,}742 \ Btu / hr$$

$$= 1.085 \times 1{,}050 \ cfm \ \times (74 \, °F - T_{supply})$$

$$\therefore T_{supply} = 67.8 \, °F$$

$$(Q_s = 5.2 - 3.1 \ kW)$$

$$(= 1.21 \times 0.5 \ m^3/s \times [23.3 \, °C - T_{supply}])$$

$$(\therefore T_{supply} = 19.8 \, °C)$$

Figure 80. Dehumidification performance of dry, cold conditioned outdoor at the peak dew-point condition
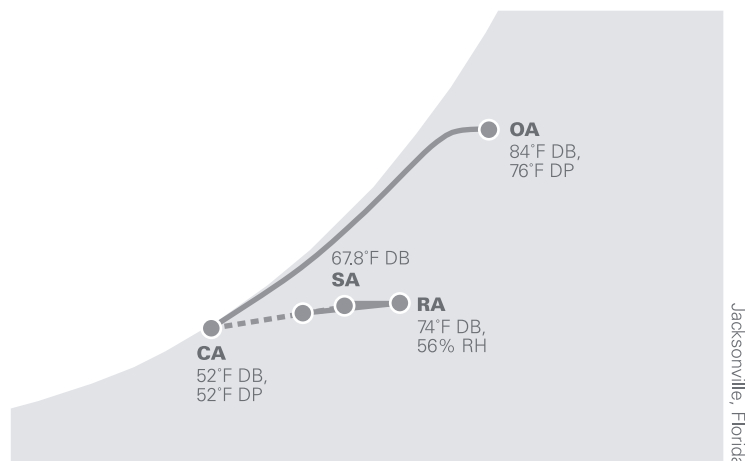
Dehumidifying with Dedicated Outdoor Air

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

$$Q_s = 1.085 \times 450 \; cfm \times (74 - 52\,°F)$$

$$= 10,742 \; Btu/hr$$

$$(Q_s = 1.21 \times 0.21 \; m^3/s \times [23.3 - 11.1\,°C])$$

$$(= 3.1 \; kW)$$

Fan–coil on a mild, rainy day:

$$Q_s = 12,250 - 10,742 \; Btu/hr$$

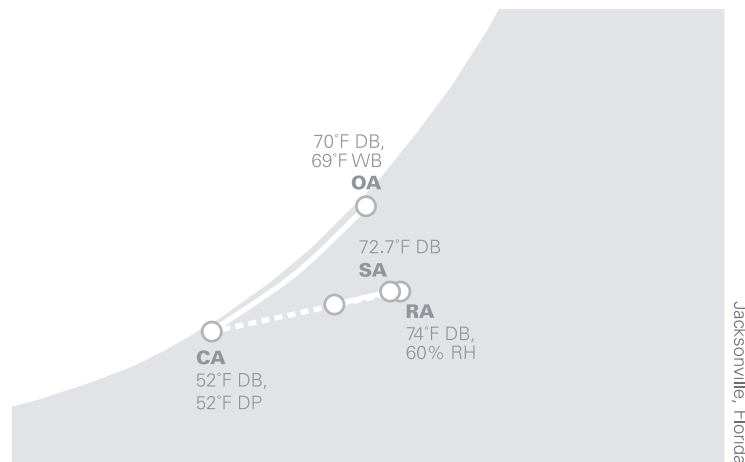$$= 1.085 \times 1,050 \; cfm \times (74\,°F - T_{supply})$$

$$\therefore T_{supply} = 72.7\,°F$$

$$(Q_s = 3.6 - 3.1 \; kW)$$

$$(= 1.21 \times 0.5 \; m^3/s \times [23.3\,°C - T_{supply}])$$

$$(\therefore T_{supply} = 22.6\,°C)$$

Figure 81. Dehumidification performance of dry, cold, conditioned outdoor air on a mild, rainy day

air temperature to 67.8°F DB (19.8°C DB). The resulting relative humidity is 56 percent, and the cooling load on the fan–coil is 0.67 tons (2.4 kW).

On a mild and rainy day (Figure 81), the dedicated outdoor-air handler still produces 52°F DP (11.1°C DP) air, but the cooling load decreases to 2.0 tons (7.0 kW). Fan–coil capacity modulates to supply air at 72.7°F DB (22.6°C DB), which results in a 60 percent-relative humidity and a cooling load of 0.12 tons (0.42 kW).

"Neutral" or "cold"?. Either design of a dedicated outdoor-air system can effectively control indoor humidity as long as the conditioned outdoor air is sufficiently dry. Table 9 helps to illustrate the benefits and tradeoffs of these designs by comparing the loads and airflows for the full-load, peak dry-bulb condition.

If the conditioned outdoor air is delivered directly to the space, providing it at a neutral dry-bulb temperature simplifies local comfort control. If delivered to a ceiling plenum, the neutral-temperature air prevents condensate from forming on the plenum surfaces.

Systems that provide dry, *cold* conditioned air avoid the energy costs of reheating the outdoor air to a neutral dry-bulb temperature. They also require less overall cooling capacity than dedicated outdoor-air systems that deliver dry, neutral-temperature air. The low dry-bulb temperature offsets part of the sensible-cooling load in the space. Similarly, when cold conditioned air is delivered to other HVAC equipment, it mixes with recirculated return air from the space; the resulting mixed-air enthalpy is lower, which in turn reduces the cooling capacity required by the coils in the HVAC terminals.

Dehumidifying with Dedicated Outdoor Air

Table 9. Comparison of coil loads and airflows for dedicated outdoor-air systems[1]

	Dry, neutral CA[2] to space		Dry, cold CA[2] to space	
Cooling-coil load:				
■ dedicated outdoor-air handler	3.0 tons	(10.6 kW)	3.0 tons	(10.6 kW)
■ fan–coil	2.8 tons	(9.8 kW)	1.8 tons	(6.3 kW)
■ total	5.8 tons	(20.4 kW)	4.8 tons	(16.9 kW)
Reheat-coil load:				
■ dedicated outdoor-air handler	9.3 MBh	(2.7 kW)	0 MBh	(0.0 kW)
■ fan–coil	—		—	
Fan airflow:				
■ dedicated outdoor-air handler	450 cfm	(0.21 m³/s)	450 cfm	(0.21 m³/s)
■ fan–coil	1,500 cfm	(0.7 m³/s)	1,050 cfm	(0.5 m³/s)
■ total	1,950 cfm	(0.9 m³/s)	1,500 cfm	(0.7 m³/s)

[1] Coil loads and airflows are based on the full-load, peak dry-bulb condition for a classroom in Jacksonville, Florida. The classroom holds 30 occupants, has an area of 10,000 ft³ (283 m³), and a target condition of 74°F DB (23.3°C DB), 50% RH.

[2] CA = Conditioned outdoor air provided by a dedicated outdoor-air handler/system

Compared with "neutral-temperature" designs, dedicated outdoor-air systems that deliver cold air provide additional benefits:

■ Less fan airflow and, therefore, lower fan-energy consumption. (Neutral-temperature conditioned air provides the space with less sensible cooling, so the local units must provide more cooling to achieve the same effect.)

■ Smaller HVAC terminals, which can either lower the initial cost and increase usable floor space *or* provide an acoustical benefit by keeping the same-sized cabinet and operating the fan at a lower speed. (In the examples summarized in Table 9, the "cold air" design represents a 30 percent reduction in both cooling-coil capacity and fan airflow.)

Dehumidifying with Dedicated Outdoor Air

Application Considerations

Humidity Control during Unoccupied Periods

As discussed in a previous chapter, "Dehumidification Primer" (p. 17), around-the-clock control of humidity can greatly reduce the risk of microbial growth on building surfaces and furnishings.

Application considerations

■ Delivering conditioned outdoor air directly to the space permits after-hours humidity control *without operating the local HVAC terminals*. If the dedicated outdoor-air unit includes a return-air path, close the outdoor-air damper and open the return-air damper to avoid conditioning unneeded outdoor air. (The ventilation requirement is significantly less after-hours, when there are few or no occupants in the building.)

If the outdoor air is delivered to local HVAC terminals, then the fans in the local units must operate in conjunction with the dedicated outdoor-air unit.

■ Using a packaged, direct-expansion air conditioner to condition the outdoor air eliminates after-hours operation of the central, chilled water plant.

■ To control after-hours operation of the dedicated outdoor-air unit, install a humidity sensor in the space. Use the sensor to enforce the maximum humidity limit for periods when the space is unoccupied.

> Including an air-to-air energy recovery device to precondition the outdoor air can help justify the cost of routing the building exhaust back to the dedicated outdoor-air handler (which provides the return-air path during after-hours operation). Outdoor-air preconditioning is discussed on pp. 98–99. ■

Building Pressurization

Along with ventilation, the dedicated outdoor-air handler can provide makeup air for local exhaust systems (such as restroom exhaust fans and kitchen exhaust hoods) and combustion processes. In some buildings — either due to oversight or by design — the dedicated outdoor-air unit is turned off during unoccupied periods, while local exhaust fans and processes continue to operate. This creates negative pressure within the building, and causes unconditioned outdoor air to infiltrate the building envelope. Infiltration of humid air can raise the humidity indoors; it can also lead to condensation within building walls and on cold surfaces in the occupied space.

While the dedicated outdoor-air unit operates, a central relief fan directly controls building pressure. Based on input from a building-pressure sensor, the capacity of the relief fan modulates to maintain the desired difference between the indoor and outdoor static pressures.

Application considerations

■ If the system control scheme disables ventilation when the building is unoccupied, make sure that it also turns off all local exhaust fans and

Dehumidifying with Dedicated Outdoor Air

combustion processes. Provide a manual override to permit operation during after-hours cleaning.

■ Monitor indoor humidity at all times and operate the dedicated outdoor-air unit whenever humidity rises to an unacceptable level.

■ Wind, variable operation of local exhaust fans, and "stack effect" can cause pressure fluctuations inside the building, even when the HVAC system is properly balanced. To prevent infiltration and avoid overpressurization during economizer operation, consider implementing a system design that directly controls building pressure.

Refer to *Building Pressurization Control,* Trane applications engineering manual AM-CON-17, to learn more about the design and control of HVAC systems that regulate building pressure. ■

■ If the dedicated outdoor-air unit serves multiple spaces *with different occupancy schedules,* consider providing a means of varying the intake airflow to meet changing ventilation requirements and to prevent overpressurization when some of the local exhaust fans are turned off.

Economizer Cooling

Designers usually size dedicated outdoor-air handlers to condition only the minimum amount of outdoor air required for ventilation. When this is the case, the supply fan in the dedicated outdoor-air handler *and* the ductwork are too small to take advantage of economizer-cooling opportunities by bringing in more outdoor air. There are other ways to provide an economizer cycle, however, depending on where the conditioned ventilation air is delivered and on the other equipment in the system.

Separate economizer-air path

When ventilation air is delivered directly to the occupied spaces, economizer cooling can be provided via outdoor-air intakes at the local HVAC terminals. Classroom unit ventilators and fan–coils, which are installed along perimeter walls, simplify the addition of a separate outdoor-air path for economizer cooling. When outdoor conditions are suitable, the local outdoor-air damper modulates between fully closed and fully open to control space temperature. The local outdoor-air damper remains closed whenever the outdoor air is too warm or too cold for economizer operation.

The dedicated outdoor-air unit typically is operated to provide conditioned air during both "economizer" and "non-economizer" modes. This practice permits the minimum outdoor airflow to be monitored and recorded during all occupied hours.

When ventilation air is delivered to the mixing boxes of other air handlers rather than to each space, each local unit can be configured to include two outdoor-air paths (Figure 82, p. 88). One path, from the dedicated outdoor-air unit, provides minimum outdoor airflow for ventilation. The second path

Dehumidifying with Dedicated Outdoor Air

Figure 82. Dedicated outdoor-air system with two outdoor-air paths for local economizing
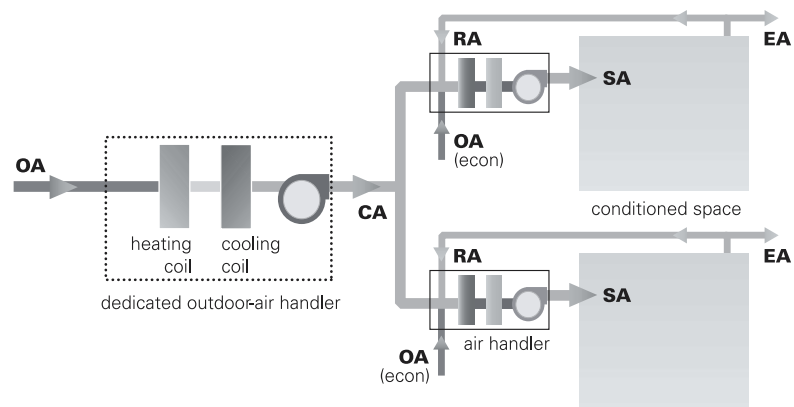
permits economizer cooling. For example, a dedicated outdoor-air unit can deliver conditioned outdoor air to the mixing boxes of several packaged rooftop air conditioners. Each mixing box includes two outdoor-air dampers: one for conditioned ventilation air and the other for untreated "economizer" air.

During normal (mechanical) cooling or heating operation, the economizer damper is closed. During economizer cooling, the economizer damper modulates as needed between fully closed and fully open to offset as much of the cooling load as possible. If the system includes a device that measures outdoor airflow, the damper in the ventilation path typically remains open, regardless of the economizer damper position. This makes it possible to monitor and record the minimum outdoor airflow for ventilation during all occupied hours.

Application considerations

■ Local economizer operation requires some form of building-pressure control. Bringing more outdoor air into the building for economizer cooling requires more relief airflow.

■ Providing two outdoor-air paths to each local air handler may require additional ductwork and controls.

Dehumidifying with Dedicated Outdoor Air

Weather permitting, outdoor air is a source of free cooling for the building. But as the air warms, it will eventually increase energy usage by creating a cooling load. To avoid the unwanted load, the high-limit shutoff disables the economizer and the HVAC system provides the minimum outdoor airflow for ventilation.

Standard 90.1 identifies five common types of high-limit shutoff for the control of airside economizers:

■ fixed dry-bulb temperature

■ differential dry-bulb temperature

■ fixed enthalpy

■ differential enthalpy

■ electronic hybrid enthalpy/ temperature

Not all control methods are appropriate in all climates. The right choice will depend on climate, hours of occupancy, method of air distribution (constant- or variable-volume), and operating-cost savings ■

For more information about using a plate-and-frame heat exchanger to provide free cooling in a chilled water system, refer to the Trane *Engineers Newsletter* titled "A New Era of Free Cooling" (ENEWS-20/3). It is archived in the "newsletters" section of www.trane.com/commercial. ■

Implications of ASHRAE Standard 90.1. Section 6.3.1 in the 2001 version of the standard defines requirements for HVAC systems with economizers. In the case of airside economizers, the section specifies damper characteristics, how to control the economizer dampers, and how to relieve air from the building to prevent overpressurization. It also prescribes how and when to shut off the economizer cycle when the weather is no longer suitable for free cooling; high-limit shutoff requirements vary by climate and control method.

Although compliance with these requirements minimizes energy use, it may not provide acceptable humidity control at all operating conditions or in all climates. For instance, Standard 90.1 permits the use of the fixed-dry-bulb-temperature high limit in any climate. When applied in a humid locale, the high limit can only disable economizer cooling when the outdoor temperature is 65°F DB (18.4°C DB) or higher. A previous example (p. 31) demonstrated what can happen in a constant-volume system when the shutoff setting is too high; in that case, the indoor humidity reached 75 percent!

Understand that Standard 90.1 does *not* mandate an economizer if the cooling capacity of each local HVAC unit is less than either 65,000 Btu/hr (19 kW) or 135,000 Btu/hr (38 kW), depending on climate—which is typically true of systems with dedicated outdoor-air handlers and local HVAC terminals. If you include an economizer anyway, the control requirements in Section 6.3.1 will not apply. Therefore, you can devise a control strategy that more effectively controls humidity.

When using fixed dry-bulb temperature for economizer control in a constant-volume system, choose a shutoff setting that is low enough to avoid bringing mild-but-humid outdoor air into the building. For better dehumidification, however, consider a more sophisticated control method that uses an indoor humidity sensor to disable the economizer whenever the relative humidity in the space exceeds 60 percent.

In constant-volume applications for which Standard 90.1 *does* require an economizer, investigate control methods such as fixed enthalpy or electronic hybrid enthalpy.
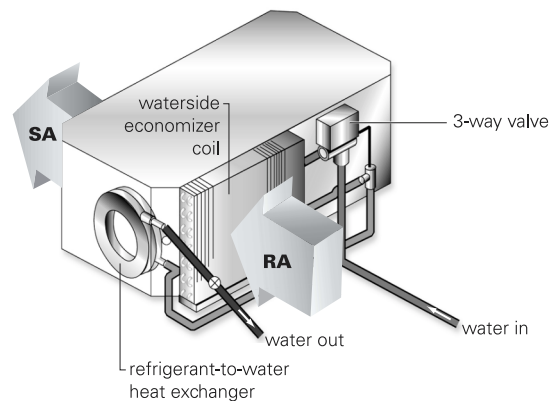
Waterside economizing

Waterside economizers can satisfy many of the economizer requirements defined in Section 6.3.1 of ASHRAE Standard 90.1–2001 and in local codes.

One way to provide "free" cooling in a chilled water system is to install a plate-and-frame heat exchanger between the chilled water and condenser-water loops, upstream of the chillers. When the leaving-tower water is cold enough, circulating it through the heat exchanger will satisfy at least part of the cooling load and reduce or eliminate the need to operate the chillers.

Dehumidifying with Dedicated Outdoor Air

Figure 83. Waterside-economizer coil in a water-source heat pump

Water-source heat pump (WSHP) systems are candidates for waterside economizing, too. For much of the year, some of the heat pumps provide cooling while others simultaneously provide heating. During the winter, the perimeter heat pumps extract heat from the water loop. The cool water then can be used by the interior heat pumps to offset the heating loads generated by lights, people, and office equipment. Diverting the cool loop water through a separate economizer coil (Figure 83) allows the heat pump to cool the entering air without operating the compressor. Because the economizer coil is situated upstream of the main coil, it can also supplement mechanical cooling when possible.

Reset Control Strategies

Many dedicated outdoor-air systems are designed to deliver conditioned outdoor air at a constant dry-bulb temperature and at a dew point that does not exceed the setpoint. This control approach is simple because it allows the dedicated outdoor-air unit to operate independently of the local HVAC terminals. It also maintains the indoor humidity at or below the upper limit.

Implementing a reset control strategy can help minimize the additional energy cost of separately treating outdoor air. The reset strategy can be based on dew point, dry-bulb temperature, or airflow.

Reset the supply-air dew point

When sensible-cooling loads in the space are high, the local HVAC terminals may offset part of the dehumidification load by performing latent cooling. When latent-cooling loads in the space are low, the dehumidification load is less, too. In either situation, the resulting indoor humidity will be lower than the upper limit, which means that the dew-point temperature for the conditioned outdoor air can be reset upward.

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Dedicated outdoor-air handler serves a single space
Install the humidity sensor either in the space or in the return-air path. When the humidity in the space drops below the upper limit, the sensor resets the dew point of the conditioned air upward only enough to enforce the maximum humidity and, in doing so, reduces cooling-energy consumption. To enforce the maximum humidity limit when the space humidity rises, the sensor resets the dew point of the conditioned air downward.

Dedicated outdoor-air handler serves multiple spaces
One option is to install a single humidity sensor in a "representative" space or in a common return-air path. When the humidity of the monitored air is low enough, the sensor resets the conditioned-air dew point upward to reduce cooling-energy use. Although economical to install, using a single sensor to represent several spaces necessarily reflects the "average" humidity. Depending on the load characteristics of the spaces, the actual humidity may exceed the limit in one space and be less than the limit in another.

(Figure 84) To resolve this multiple-space problem, install a humidity sensor in each space that is served by the dedicated outdoor-air handler, or at least in those spaces where high humidity is expected. A building automation system (BAS) monitors all of the spaces to determine the critical space with the highest humidity. Then, the BAS sends a signal to the dedicated outdoor-air unit to reset the leaving-air dew point only enough to enforce the maximum humidity limit in the critical space. This approach optimizes system performance: By responding to actual conditions, the system maintains the humidity at or below the limit in *all* spaces while consuming no more cooling energy than necessary.

Figure 84. Control arrangement for fan–coil system with dedicated outdoor-air
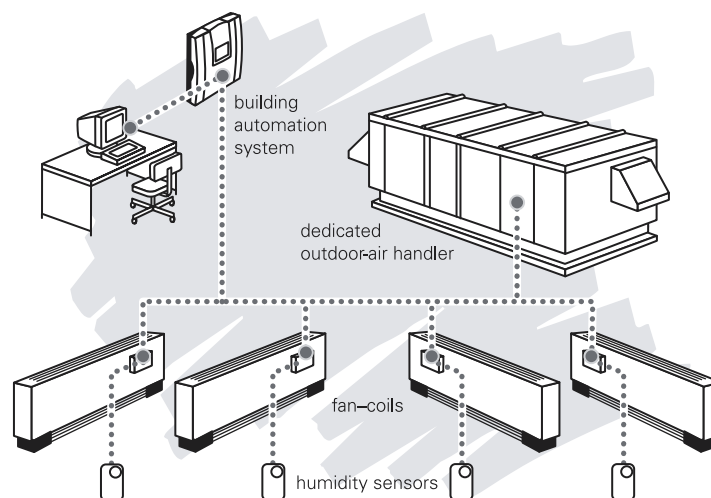
Dehumidifying with Dedicated Outdoor Air

Reset the supply-air dry-bulb temperature

When a dedicated outdoor-air handler delivers air that is cooler than the space setpoint, it reduces the sensible-cooling load on the local HVAC terminals. If the space load is already low, the terminal may need to add heat to avoid overcooling the space. Resetting the dry-bulb temperature of the conditioned outdoor air by modulating the reheat capacity of the dedicated outdoor-air unit minimizes the reheat energy used by the local terminals.

Dedicated outdoor-air handler serves a single space
To reset the dry-bulb temperature of the conditioned outdoor air, install a thermostat in the space to modulate the reheat capacity of the dedicated outdoor-air handler. Commonly used in hotel–hallway applications (discussed on p. 93), the dedicated outdoor-air handler enforces the humidity limit in the space without overcooling it.

Dedicated outdoor-air handler serves multiple spaces
To most effectively implement a reset strategy, use a building automation system to monitor all of the HVAC terminals and to identify the "critical space" — that is, the space with the lowest sensible-cooling load. If the terminals contain chilled water coils, the BAS monitors the position of the chilled-water control valves. The space with the lowest load is served by the terminal whose control valve is the most closed. (If the terminals contain refrigerant coils, the BAS monitors compressor operation; in that case, the critical space is the one with the shortest compressor run-time.)

Based on a signal from the BAS, the dedicated outdoor-air handler then modulates its *reheat* capacity, resetting the dry-bulb temperature of the conditioned air upward only enough so that the terminal with the lowest sensible-cooling load operates at near-zero cooling capacity. Notice that reset is accomplished by modulating reheat capacity, not cooling capacity. The premise is that the dew point of the conditioned air is controlled independently to meet the humidity-control requirements for the spaces. This optimization strategy provides conditioned outdoor air that offsets as much of the sensible-cooling load in each space as possible *without* requiring any of the terminals to activate their heating coils. This can be beneficial if the dedicated outdoor-air handler recovers reheat energy from some other part of the system, thereby avoiding the use of "new" energy for reheat, both at the dedicated outdoor-air unit and at the terminals.

Dynamically resetting the dry-bulb temperature of the conditioned outdoor air is particularly beneficial in a two-pipe system…especially if a packaged, DX air conditioner serves as the dedicated outdoor-air unit. The packaged DX unit can delay the changeover from cooling to heating because it is not connected to the chiller or boiler; in doing so, it saves energy and improves occupant comfort. When the two-pipe system is in *cooling mode* (boiler off), the dry-bulb temperature supplied by the dedicated outdoor-air unit is reset upward to maintain a *heating load* of zero on the terminal in the critical space. When the

Dehumidifying with Dedicated Outdoor Air

Figure 85. Conditioned outdoor air delivered to a common hallway
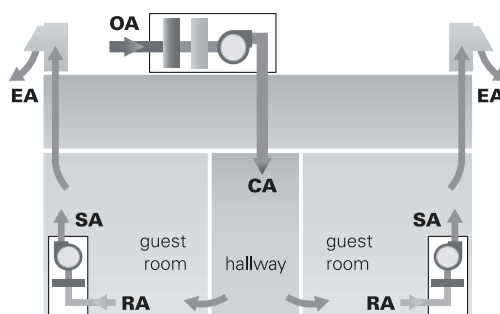
two-pipe system is in *heating mode* (chiller off), the dry-bulb temperature is reset downward to maintain a *cooling load* of zero on the terminal in the critical space.

If one dedicated outdoor-air handler serves multiple spaces with widely differing loads — for example, interior spaces with nearly constant full-load cooling and perimeter spaces that require heating — then for the interior spaces, it may be necessary to size the terminals as if the dedicated outdoor-air unit delivers neutral-temperature air. This practice may prevent downsizing of some of the terminals, so the benefits of this strategy must be weighed against the first-cost impact of terminal-unit sizing.

Dedicated outdoor-air handler serves a common hallway
In some applications, such as hotels, the dedicated outdoor-air handler ventilates an entire floor of guest rooms by delivering conditioned outdoor air to a central hallway (Figure 85). Local exhaust then draws the ventilation air into each guest room from the hallway. In addition to providing outdoor air for ventilation, the dedicated outdoor-air handler also controls the temperature and humidity in the hallway.

In this case, size the dedicated outdoor-air unit based on either the ventilation airflow requirement of the spaces it serves, *or* the sensible load in the hallway, whichever is the highest. A humidity sensor installed in the hallway resets the dew point of the conditioned outdoor air, maintaining the humidity in the hallway at or below the desired limit. A thermostat, also installed in the hallway, resets the dry-bulb temperature of the conditioned outdoor air as needed to maintain the setpoint.

Dehumidifying with Dedicated Outdoor Air

Reset outdoor-air quantity

Most dedicated outdoor-air handlers are designed to deliver a constant flow of conditioned outdoor air; however, applications in which the population varies widely during normal building operation may benefit from variable intake airflow. Automatically adjusting the quantity of conditioned outdoor air to match the system's current ventilation requirements can reduce operating costs without sacrificing indoor air quality.

With separate dampers and a method for fan-capacity control, a dedicated outdoor-air system can be designed to vary the outdoor airflow to each space based on the estimated population. Population estimates can originate from occupancy sensors (occupied or unoccupied), carbon dioxide (CO$_2$) sensors, or time-of-day schedules in a building automation system (BAS).

Reheating Conditioned Air with Recovered Heat

Dedicated outdoor-air handlers that reheat the outdoor air, after cooling it to a low dew point for dehumidification, can use either "new" energy (gas, electricity, hot water, or steam) or sensible heat recovered from another part of the system.

Section 6.3.2.3 of the U.S. energy standard, ASHRAE 90.1–2001 (see excerpt on p. 8) defines the circumstances, or "exceptions," in which new-energy reheat is permitted. For example, Exception A permits the use of new-energy reheat if the system first reduces the supply-air volume to the minimum ventilation rate specified by ASHRAE Standard 62. Because the quantity of air typically supplied by the dedicated outdoor-air unit already equals the minimum ventilation airflow required by Standard 62, its airflow cannot be reduced further. Therefore, in most cases, Standard 90.1 will not prohibit new-energy reheat in dedicated outdoor-air units that deliver neutral-temperature air for ventilation.

Recovering heat for reheat may be desirable even if it is not required by Standard 90.1 because it reduces system operating costs. Sources of recoverable heat include:

■ Hot refrigerant in a refrigeration system

■ Condenser water in a water-cooled, chilled water system

■ Another air stream, or another location within the same air stream

Refrigerant heat recovery

Systems that include a packaged, DX-type air conditioner as the dedicated outdoor-air unit are excellent candidates for refrigerant-heat recovery (Figure 86). Because reheat is required while the cooling coil dehumidifies the outdoor air, recovering heat from the refrigeration system saves reheat energy. A condenser-reheat coil transfers sensible heat from the hot refrigerant to the

Dehumidifying with Dedicated Outdoor Air

Figure 86. Packaged DX, dedicated outdoor-air unit with condenser-reheat coil
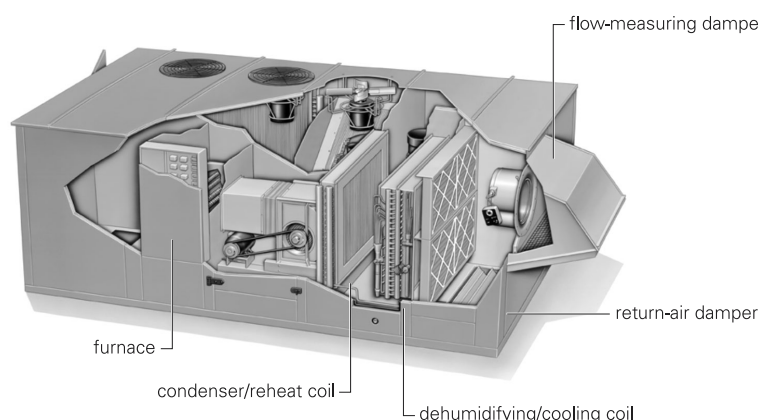
air downstream of the cooling coil. Refrigerant heat recovery is discussed further on pp. 57–58.

Condenser-water heat recovery

Sensible heat also can be recovered from the warm water leaving the condenser of a water-cooled chiller. Instead of pumping the condenser water to the cooling tower, it can be circulated to one or more dedicated outdoor-air handlers and used for reheat. Any water-cooled chiller can provide sensible heat for reheat. Also, water is more easily distributed to multiple units than refrigerant. For examples of common system configurations, see pp. 53–56.

Air-to-air heat recovery/transfer

Sensible heat for reheat can be recovered from another air stream—or another location in the same air stream—using an air-to-air heat exchanger, such as a coil loop, a fixed-plate heat exchanger, a heat pipe, or a rotary sensible-heat exchanger (heat wheel). When the air-to-air heat exchanger provides reheat in a dedicated outdoor-air system, it can be applied in one of two configurations: series or parallel.

Series configuration
Figure 87 (p. 96) shows a dedicated outdoor-air handler that includes a fixed-plate heat exchanger. The heat exchanger, which could be any type of sensible-energy recovery device, is applied in a series, or "wrap-around," configuration. The heat exchanger removes sensible heat from the air upstream of the cooling/dehumidifying coil and transfers it downstream to reheat the air leaving the coil.

Applying the air-to-air heat exchanger in a series arrangement conserves "new" energy that would otherwise be used to reheat the outdoor air. It also reduces the cooling load by precooling the outdoor air upstream of the

Consult *Air-to-Air Energy Recovery in HVAC Systems,* Trane applications engineering manual SYS-APM003-EN, for more information about using an air-to-air heat exchanger for reheat in a dedicated outdoor-air system. Topics include series and parallel configurations and a coil loop with three coils. ■

Dehumidifying with Dedicated Outdoor Air

Figure 87. Series heat-recovery arrangement for reheating conditioned outdoor air
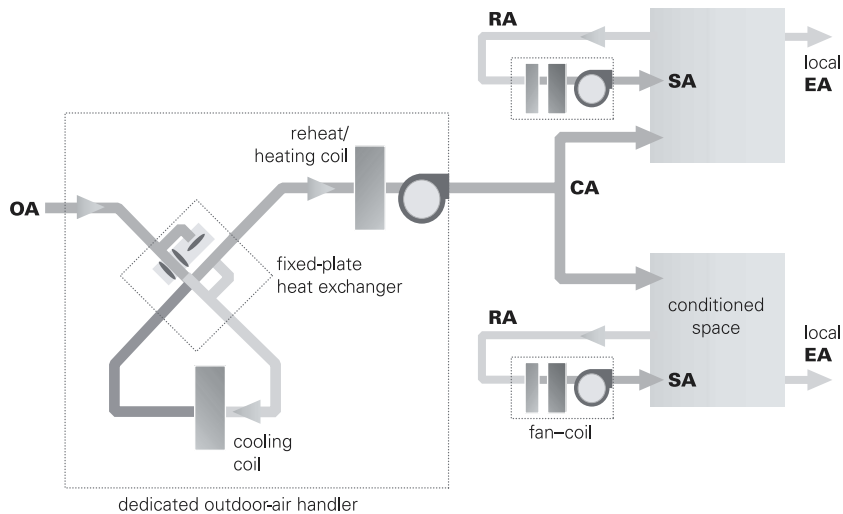
In the strict sense of the term, "heat recovery" does not include the series arrangement of an air-to-air heat exchanger. Rather than capturing heat from another source (exhaust air, for example), the series arrangement moves heat from one location to another within the same air stream. ■

dehumidifying coil. In this application, reheat is required whenever the cooling coil dehumidifies, even at design conditions, because the conditioned outdoor air is delivered at a "neutral" dry-bulb temperature. The precooling effect provided by the heat exchanger reduces the design load on the cooling coil; it also may permit downsizing of the cooling plant.

During cold weather, when the dedicated outdoor-air unit must heat the entering air, the cooling coil is turned off. Because the air-to-air heat exchanger is arranged in series with the cooling coil, it provides no benefit. An arrangement that bypasses the cooling coil and air-to-air heat exchanger during cold weather would save fan energy by reducing the airside pressure drop.

Parallel configuration
Figure 88 shows the same dedicated outdoor-air handler, this time with the air-to-air heat exchanger applied in a parallel arrangement. The heat exchanger recovers sensible heat from the exhaust air and releases it downstream of the cooling/dehumidifying coil to reheat the leaving air. (This example shows a fixed-plate heat exchanger, but any type of air-to-air, sensible-heat exchanger will work.)

Like the series arrangement, applying the air-to-air heat exchanger in a parallel arrangement saves the "new" energy that would otherwise be needed to reheat the air. Unlike the series arrangement, however, the parallel arrangement does not reduce the load on the cooling coil, nor does it enable downsizing of the cooling plant. This is because the heat recovered from the exhaust air would be rejected from the building anyway.

Dehumidifying with Dedicated Outdoor Air

Figure 88. Parallel heat-recovery arrangement for reheating conditioned outdoor air
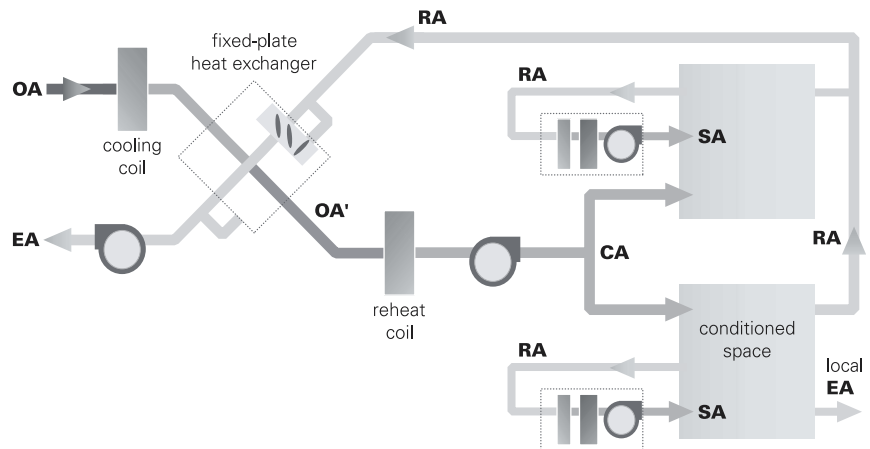
Another important advantage characterizes the parallel arrangement: When it is cold outside and the dedicated outdoor-air handler must heat the entering outdoor air, the parallel-arranged, air-to-air heat exchanger can recover heat from the exhaust air to warm the entering outdoor air. This lowers the operating costs for heating and may allow downsizing of the heating plant.

Cold climates may require a means of frost prevention for the air-to-air heat exchanger during full-recovery, heating operation. If the dedicated outdoor-air handler includes a chilled-water cooling coil, it may also require freeze protection because the heat exchanger adds heat *downstream* of the coil.

Series or parallel?
In dedicated outdoor-air handlers that supply neutral-temperature air, both arrangements reduce the "new" energy required for reheat. The series arrangement can transfer more heat whenever the outdoor air is warmer than the exhaust air. It also saves cooling energy, which may allow downsizing of the cooling plant. By comparison, the parallel configuration depends less on outdoor conditions and transfers more heat whenever the outdoor air is cooler than the exhaust air. (This is because the temperature of the exhaust air is relatively constant.) The parallel configuration also can operate during the heating mode, which saves heating energy and may allow downsizing of the heating plant.

Implementing the parallel configuration requires additional ductwork to bring most of the building's exhaust air back to a central location; the series configuration does not. Given the difference in initial cost, air-to-air heat recovery is commonly implemented using the series arrangement unless the system includes an additional energy-recovery device (for example, a total-

Dehumidifying with Dedicated Outdoor Air

Figure 89. "Dual-function" air-to-air energy recovery
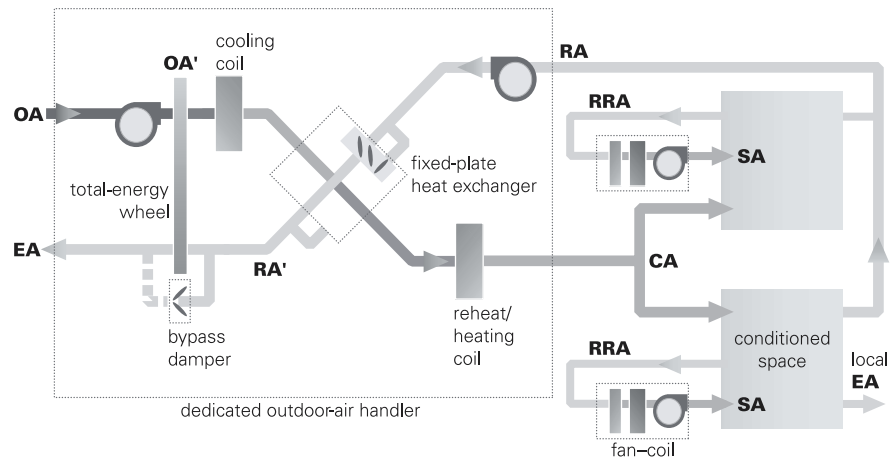
energy wheel) for outdoor-air preconditioning. Figure 89 illustrates a dedicated outdoor-air handler that includes "dual-function" energy recovery.

Ultimately, the right choice for a given project will depend on the balance of initial cost, cooling- and heating-energy savings, and the increased fan energy imposed by the additional static-pressure drop through the heat exchanger.

Preconditioning Outdoor Air with Recovered Energy

The previous section discussed the use of air-to-air energy recovery for reheating conditioned outdoor air. This type of energy recovery also is suited for *preconditioning* outdoor air as it enters the dedicated outdoor-air handler.

Figure 90 illustrates a dedicated outdoor-air handler that includes a total-energy wheel to transfer energy between the outdoor air and exhaust air. During the summer, when the outdoor air is hot and humid, the *total*-energy recovery device can precool and predry the outdoor air by transferring both sensible heat and moisture to the exhaust air. During the winter, when the outdoor air is cold and dry, the same device can preheat and prehumidify the outdoor air by recovering both sensible heat and moisture from the exhaust air and transferring it to the entering outdoor air.

Substituting a *sensible*-energy recovery device (such as a coil loop, fixed-plate heat exchanger, heat pipe, or sensible-energy wheel) may reduce the initial cost of the device, but it precludes the opportunity to predry or prehumidify the entering outdoor air.

Using air-to-air energy recovery to precondition the outdoor air can reduce the operating cost of the dedicated outdoor-air handler; it may even allow downsizing of the mechanical cooling, heating, and humidification equipment.

Consult *Air-to-Air Energy Recovery in HVAC Systems,* Trane applications engineering manual SYS-APM003-EN, to learn more about using an air-to-air heat exchanger to precondition outdoor air. The manual discusses compliance with ASHRAE Standard 90.1 and various energy-recovery technologies; it also outlines control modes for dedicated outdoor-air systems. ■

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Figure 90. Parallel heat-recovery arrangement for preconditioning outdoor air
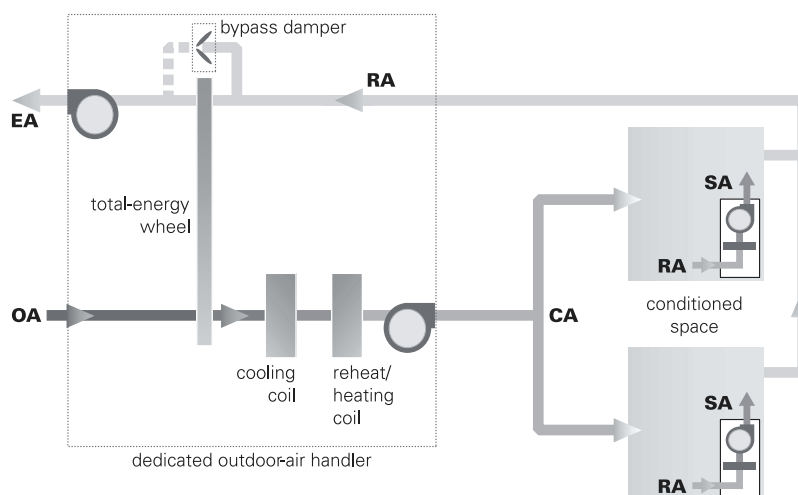
As demonstrated on pp. 27–29, however, a total-energy recovery device does little to control space humidity. A cold coil (or an active desiccant system) is needed to dehumidify the air to a dew point that is drier than the air in the space.

Any sensible- or total-energy recovery device can precondition outdoor air. The choice between them usually is based on the initial cost and energy-savings potential in the specific application. In either case, the parallel arrangement required for preconditioning also requires additional ductwork to bring most of the building-exhaust air back to a central location.

Control of the energy-recovery device
Integrating the operation of the energy-recovery device into the control sequences for the dedicated outdoor-air handler (and the rest of the HVAC system) is critical to realize the anticipated cost savings and to reduce the payback period. Although the simplest strategy cycles the device with the fan, this approach does not maximize energy savings.

At many outdoor conditions, the capacity of the energy-recovery device must modulate or the device must be turned off to avoid transferring unwanted moisture or sensible heat to the entering outdoor air. This is especially true for dedicated outdoor-air units that deliver dry, *cold* air rather than reheating the conditioned air to a neutral temperature.

Operating the energy-recovery device at full capacity under these conditions, which *could* represent 15 percent to 25 percent of the annual operating hours, will waste energy by operating the wheel and the dedicated outdoor-air handler in opposition to each other. ■

Afterword

The dehumidification performance of a cold-coil HVAC system hinges on the ability of the cooling coil to reduce the *dew point* of the passing air below that of the air in the space. Single-zone, constant-volume, mixed-air systems are most often used to condition entering outdoor air; they also can be the most problematic when it comes to dehumidification, particularly at part-load conditions. VAV systems generally provide effective, coincidental dehumidification over a range of indoor load conditions.

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

When properly designed and controlled, however, the HVAC system can significantly reduce the moisture content of indoor air. Increased humidity in the space can also result from higher-than-necessary supply airflow…either due to an overly conservative estimate of the space sensible-cooling load or the use of oversized, packaged DX equipment. When selecting cooling equipment, especially for constant-volume applications, exercise particular care to avoid oversizing the design airflow. Analyze system dehumidification performance at both full- and part-load conditions, and consider the advantages and disadvantages of each system enhancement. The right choice for a given project depends on the climate, building use, available budget, and operating-cost goals. ∎

Appendix A: Psychrometric Analysis

An imaginary, elementary-school classroom in Jacksonville, Florida, provides the context for the examples of dehumidification performance that appear throughout this manual. In the chapter titled "Dehumidifying with Constant-Volume Mixed Air" (pp. 19–60), the classroom is air conditioned by the basic single-zone, constant-volume HVAC system shown in Figure 91.

A thermostat compares the dry-bulb temperature in the classroom to the setpoint. The system then modulates the capacity of the cooling coil accordingly, which adjusts the supply-air temperature until the temperature in the classroom matches the setpoint.

As the coil cools the passing air stream, it reduces the air's moisture content via condensation on the coil surface. The thermostat determines the coil-surface temperature and, therefore, the supply-air temperature. Modulating the coil to increase its sensible-cooling capacity makes the coil surface colder, which causes more condensation and (therefore) more dehumidification. Reducing coil capacity results in a warmer coil that provides less dehumidification.

Predicting the dehumidification performance of an HVAC system entails a psychrometric analysis of its operation, both at full-load *and at the range of part-load* conditions. This section demonstrates the analysis process by examining the performance of this basic system at two operating points…

- Full load, peak dry bulb: 96°F DB, 76°F WB (35.7°C DB, 24.5°C)

- Part-load, peak dew point: 76°F DP, 84°F DB (24.6°C DP, 28.8°C DB)

The outcome of this two-part analysis is summarized on pp. 20–21.

> 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. ■

Figure 91. Basic, constant-volume HVAC system

- volume: 10,000 ft³
- occupancy: 30 people
- setpoint: 74°F DB
- desired humidity: 50% RH
- outdoor airflow: 15 cfm/person
- supply airflow: 9 air changes/hr or 1,500 cfm

Appendix A: Psychrometric Analysis

Full-Load, Peak Dry-Bulb Condition

Step 1: Plot the outdoor condition (OA) and the desired indoor condition (RA) on a psychrometric chart. The *ASHRAE Handbook– Fundamentals* lists the 0.4 percent peak-dry-bulb condition for Jacksonville, Florida, as 96°F DB, 76°F WB (35.7°C DB, 24.5°C WB). For this example, the desired condition in the classroom is 74°F DB (23.3°C DB) with a relative humidity of 50 percent. See Figure 92.

Step 2: Calculate the sensible- and latent-cooling loads in the space. At the peak dry-bulb condition, the sensible and latent loads calculated for the classroom are 29,750 Btu/hr (8.7 kW) and 5,250 Btu/hr (1.5 kW), respectively. These loads yield a space sensible-heat ratio (SHR) of 0.85, which means that 85 percent of the classroom's cooling load is sensible and 15 percent is latent.

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

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

Step 3: Calculate the temperature and quantity of air required to offset the sensible-cooling load in the space. Having already determined the space sensible-cooling load and the desired indoor temperature, either: define the supply airflow and calculate the supply-air

Figure 92. Outdoor (peak dry bulb) and indoor conditions for Jacksonville classroom
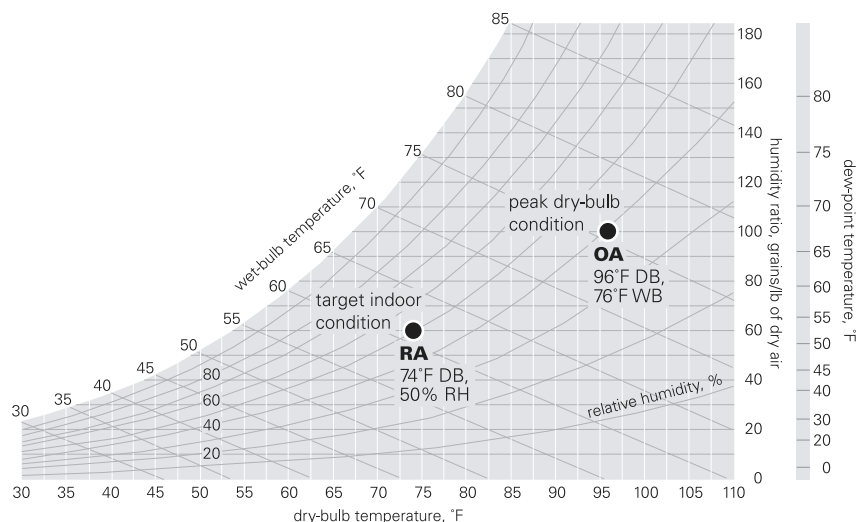
Appendix A:
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temperature, *or* choose the supply-air temperature and calculate the required airflow. In either case, solve following equation:

$$Q_s = 1.085 \times V_{sa} \times (T_{sp} - T_{sa})$$

$$(Q_s = 1.21 \times V_{sa} \times [T_{sp} - T_{sa}])$$

> where,
> Q_s = sensible-cooling load in the space, Btu/hr (kW)
> T_{sa} = dry-bulb temperature of the supply air, °F (°C)
> T_{sp} = desired dry-bulb temperature for the space, °F (°C)
> V_{sa} = supply airflow delivered to the space, cfm (m³/s)

For this example, assume that the supply airflow for the classroom is nine air changes per hour. The volume of the classroom is 10,000 ft³ (283 m³), so the required supply airflow is 1,500 cfm (0.7 m³/s):

$$V_{sa} = \frac{10,000\ ft^3/air\ change \times 9\ air\ changes/hr}{60\ min/hr} = 1,500\ cfm$$

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

To offset the sensible-cooling load and satisfy the thermostat setpoint by maintaining 74°F DB (23.3°C DB) in the classroom, the supply airflow must be delivered at 55.7°F DB (13.1°C DB):

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

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

Step 4: Calculate the entering-air condition for the cooling coil. The air entering the cooling coil is a mixture of outdoor air and recirculated air returning from the space. For this example, assume that the condition of the return air is the same as the air in the classroom — 74°F DB (23.3°C DB) and a relative humidity of 50 percent.

According to ASHRAE Standard 62–2001, properly ventilating the classroom requires 15 cfm (8 L/s) of outdoor air per person. Given the design occupancy of 30 people, and because the system only serves this classroom, 450 cfm

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Figure 93. Plotting the entering mixed-air condition for the cooling coil
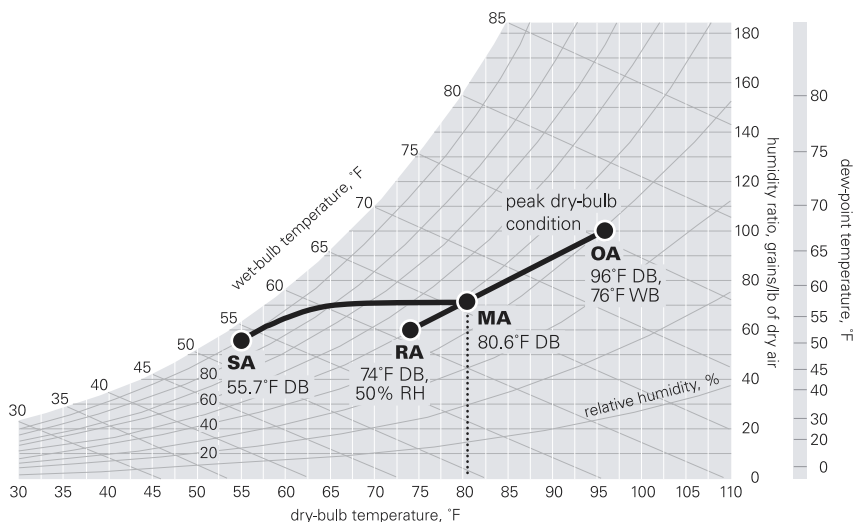
(0.21 m³/s) of the 1,500 cfm (0.7 m³/s) supply airflow must be outdoor air. The resulting dry-bulb temperature of this mixture is 80.6°F (26.8°C):

$$T_{ma} = \frac{(450\ cfm\ \times 96°F\ ) + (1{,}050\ cfm \times 74°F\ )}{450\ cfm\ + 1{,}050\ cfm} = 80.6°F$$

$$\left(T_{ma} = \frac{[0.21\ m^3/s \times 35.7°C\ ] + [\,0.5\ m^3/s \times 23.3°C\ ]}{0.21\ m^3/s\ + 0.5\ m^3/s} = 26.8°C \right)$$

When plotted on a psychrometric chart (Figure 93), the mixed-air (MA) condition lies on the straight line that connects the outdoor-air (OA) and return-air (RA) conditions. In this case, the condition of the mixed air, which will enter the cooling coil, lies at the intersection of OA–RA and the line that corresponds to the calculated dry-bulb temperature, 80.6°F (26.8°C).

Step 5: Determine the leaving-coil condition of the supply air. Using the curvature of the adjacent "coil curves" as a guide, draw a curve from the mixed-air condition (MA) until it intersects the supply-air temperature (SA) calculated in Step 3—in this case, 55.7°F (13.1°C).

Note: The Trane psychrometric chart includes a series of curves depicting the changes in dry-bulb and dew-point temperatures that occur as air passes through a "typical" cooling coil. Derived from laboratory tests, these "coil curves" represent various coil geometries at different air and coolant temperatures. For a more precise prediction of performance, use software that accurately models the actual coil geometry.

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Figure 94. Using supply-air temperature and the sensible-heat ratio to find the resulting air condition in the space
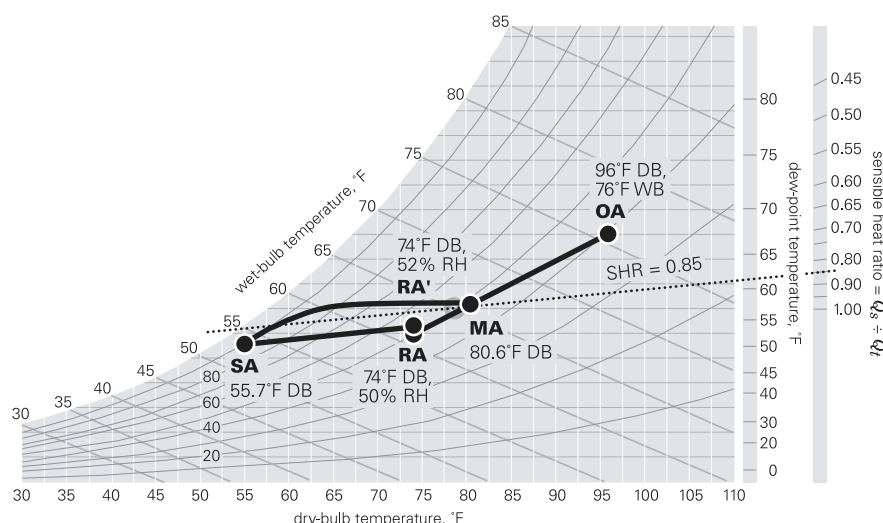
Step 6: Determine the air condition of the space. The space sensible-heat ratio (SHR) determines the resulting air condition in the space for a given supply-air temperature:

a Establish the slope of the SHR line by drawing a line that connects the appropriate value on the SHR scale — 0.85, in this case — with the index point in the center of the chart.

b Using the SHR line as a reference, draw a parallel line from the supply-air condition (SA) to the line representing the desired space dry-bulb temperature — that is, from 55.7°F DB (13.1°C) to 74°F (23.3°C).

(Figure 94) Although the target humidity for the classroom is 50 percent, the 0.85 SHR and 55.7°F DB (13.1°C) supply air will yield a relative humidity of roughly 52 percent. To "close the loop," we must repeat the psychrometric analysis (Steps 1–6) until successive iterations produce the same value for RA.

Repeat Step 1. The outdoor-air condition (OA) remains the same; but in this iteration, RA' represents the indoor condition.

Repeat Steps 2 and 3. Because the sensible- and latent-cooling loads in the classroom are unchanged, the supply airflow and supply-air dry-bulb temperature also remain the same.

Repeat Step 4. Plot the new mixed-air condition (MA') by drawing a line that connects the outdoor-air condition (OA) to the *new* return-air condition (RA'). MA' lies at the intersection of OA–RA' and the dry-bulb temperature that was calculated for the mixed air, 80.6°F DB (26.8°C DB).

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Figure 95. "Closing the psychrometric loop" for the peak dry-bulb condition
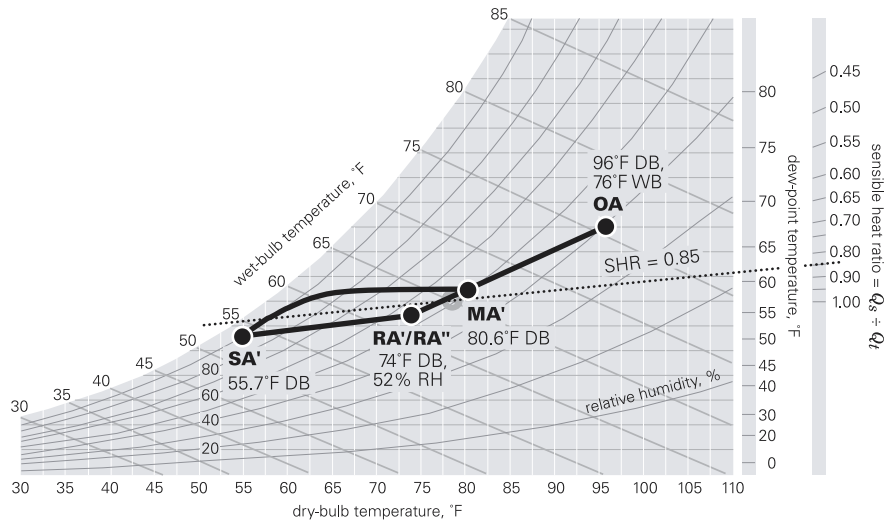
Repeat Step 5. To find the new supply-air condition (SA'), draw a curve from the new mixed-air condition (MA') to the supply-air temperature of 55.7°F (13.1°C). Use the curvature of the adjacent "coil curves" as a guide.

Repeat Step 6. Using the 0.85 SHR line as a reference, draw a parallel line from the supply-air condition (SA') to the line representing the desired space temperature, 74°F (23.3°C) in this case. Figure 95 shows that the resulting air condition (RA'') in the space now coincides with the starting point.

At the full-load, peak dry-bulb condition for this example, controlling the basic constant-volume system to 74°F DB (23.3°C DB) results in a relative humidity of about 52 percent.

Step 7: Calculate the required cooling-coil capacity. Solving the following equation shows that the cooling coil must provide 4.78 tons (16.8 kW) of cooling at the full-load, peak dry-bulb condition:

$$Q_T = 4.5 \times V_{sa} \times (h_{ma} - h_{sa})$$

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

$$(Q_T = 1.2 \times V_{sa} \times [h_{ma} - h_{sa}])$$

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

where,
h_{ma} = enthalpy of mixed air entering the cooling coil, Btu/lb (kJ/kg)

In the Q_T equations at right, 4.5 and 1.2 are derived from properties of the air; they are *not* constants. At the "standard" air condition, which is 69°F (21°C) dry air at sea level, the product of density and the conversion for time (I-P units only) equals 4.5 (1.2). A different air condition or elevation will result in a different value. ■

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Psychrometric Analysis

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h_{sa} = enthalpy of supply air leaving the cooling coil, Btu/lb (kJ/kg)
Q_T = total cooling load on the coil, Btu/hr (kW)
V_{sa} = supply airflow passing through the coil, cfm (m³/s)

Part-Load, Peak Dew-Point Condition

Note: The steps involved in a psychrometric analysis remain the same, regardless of the operating conditions. This section details these steps again by way of illustration.

Step 1: Plot the outdoor (OA) and target indoor (RA) conditions on a psychrometric chart. The *ASHRAE Handbook–Fundamentals* lists the peak dew-point condition for Jacksonville, Florida, as 76°F DP, 84°F DB (24.6°C DP, 28.8°C DB). For this example, the desired condition in the classroom remains at 74°F DB (23.3°C DB) with a relative humidity of 50 percent.

Step 2: Calculate the sensible- and latent-cooling loads in the space.
The outdoor condition at the peak dew point is comparatively cooler than at the peak dry bulb. Because less heat conduction occurs, the sensible-cooling load in the space drops to 17,850 Btu/hr (5.2 kW). However, the latent-cooling load remains the same because the occupants are still in the classroom. These loads represent a space sensible-heat ratio (SHR) of 0.77:

$$SHR = \frac{17,850\,Btu/hr}{17,850\,Btu/hr\,+\,5,250\,Btu/hr} = 0.77$$

$$\left(SHR = \frac{5.2\;kW}{5.2\;kW\,+\,1.5\;kW} = 0.77 \right)$$

Space sensible and latent loads, and therefore the space SHR, vary with outdoor conditions, the time of year, and the time of day. Unlike the peak dry-bulb condition, where you can assume (for most North American locations) that the highest dry-bulb temperature occurs during the midafternoon hours of July or August, such an assumption is not valid for the peak dew-point condition.

Hourly weather data for Jacksonville indicates that the 0.4 percent, peak dew-point condition occurs on the occasions listed below. Notice that the time of each occurrence differs from one date to the next. Depending on the

		Outdoor condition	
Date	Time of day	Dry bulb	Dew point
August 23	8:00 a.m.		
September 1	1:00 p.m.	84.0°F DB	76.1°F DP
September 2	4:00 p.m.		

The occurrences of the peak dew-point in Jacksonville appear to be concentrated in late August/early September. By contrast, a similar analysis for St. Louis, Missouri, reveals that this condition occurs in late June/early July. Clearly, it is unwise to assume that the peak dew-point condition occurs during the same month, regardless of location. ∎

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orientation of the exterior surfaces of a given space, as well as the occupancy schedule, the space SHR can vary dramatically between these occasions.

The classroom in this example has one, west-facing exterior wall with windows. Of the three, peak dew-point occurrences, the lowest space SHR occurs at 8:00 a.m. because the solar and conduction loads are smaller. A psychrometric analysis will yield substantially different results, depending on which peak dew-point occurrence is used to calculate the cooling loads in the space.

| Date | Time of day | Cooling loads in classroom, Btu/hr | | Space SHR |
		Sensible	Latent	
August 23	8:00 a.m.	17,850	5,250	0.77
September 1	1:00 p.m.	29,975	5,250	0.85
September 2	4:00 p.m.	29,250	875	0.97

Step 3: Calculate the temperature and quantity of air required to offset the space sensible-cooling load. Because a constant-volume system serves the classroom in this example, the supply airflow must remain at 1,500 cfm (0.7 m³/s). To compensate for the smaller space sensible-cooling load and still maintain the 74°F DB (23.3°C) setpoint, the system must deliver the supply airflow at 63°F (17.2°C):

$$Q_s = 1.085 \times 1,500\,cfm \times (74°F - T_{sa}) = 17,850\,Btu/hr \therefore T_{sa} = 63.0°F$$

$$(Q_s = 1.21 \times 0.7\,m³/s \times [23.3°C - T_{sa}] = 5.2\,kW \therefore T_{sa} = 17.2°C)$$

Step 4: Calculate the entering-air condition for the cooling coil. The coil cools a mixture of outdoor air and recirculated return air. Assuming that the return-air condition is identical to the condition in the classroom — 74°F DB (23.3°C) DB and 50 percent relative humidity — the resulting mixed-air dry-bulb temperature is 77°F (24.9°C):

$$T_{ma} = \frac{(450\,cfm \times 84°F) + (1,050\,cfm \times 74°F)}{450\,cfm + 1,050\,cfm} = 77.0°F$$

$$\left(T_{ma} = \frac{[0.21\,m³/s \times 28.8°C] + [0.5\,m³/s \times 23.3°C]}{0.21\,m³/s + 0.5\,m³/s} = 24.9°C \right)$$

On the psychrometric chart (Figure 96), mixed-air condition MA lies on the straight line that connects the outdoor-air and return-air conditions, OA and RA.

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Figure 96. Part-load, peak dew-point conditions for the Jacksonville, Florida, classroom
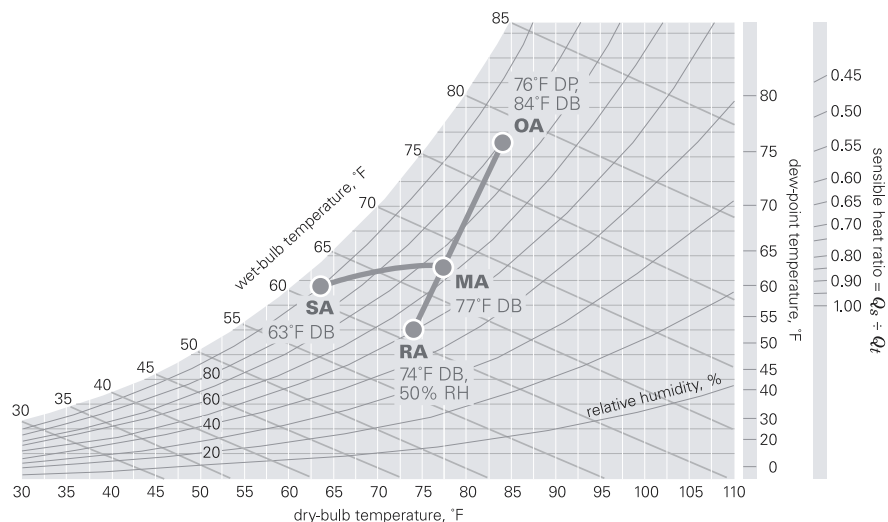
Step 5: Determine the leaving-coil, supply-air condition. Using the curvature of the adjacent "coil curves," draw a curve from the mixed-air condition (MA) until it intersects the supply-air temperature (SA) calculated in Step 3—in this case, 63°F (17.2°C).

Step 6: Determine the air condition in the space. (Figure 97) Draw a 0.77 SHR line that intersects both the supply-air condition and the target dry-bulb

Figure 97. Resulting air condition (RA') in the space does not match the target (RA)
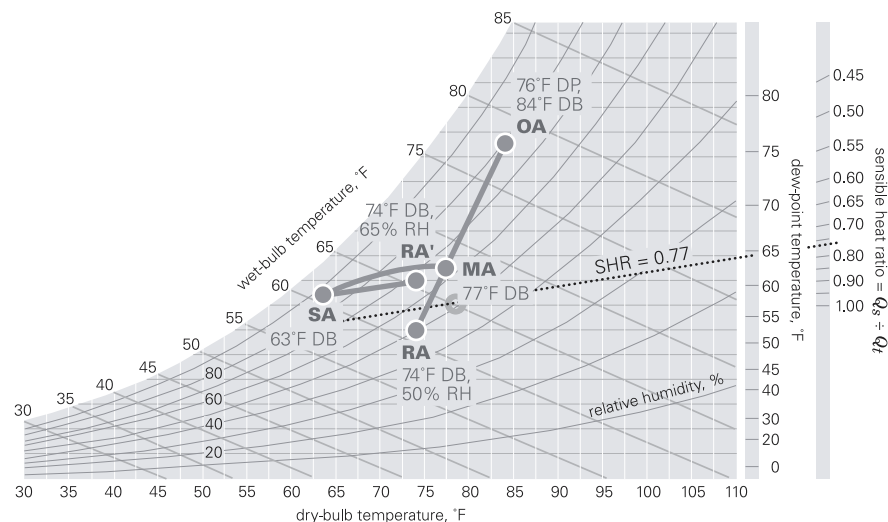
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Figure 98. "Closing the psychrometric loop" for the peak dew-point condition
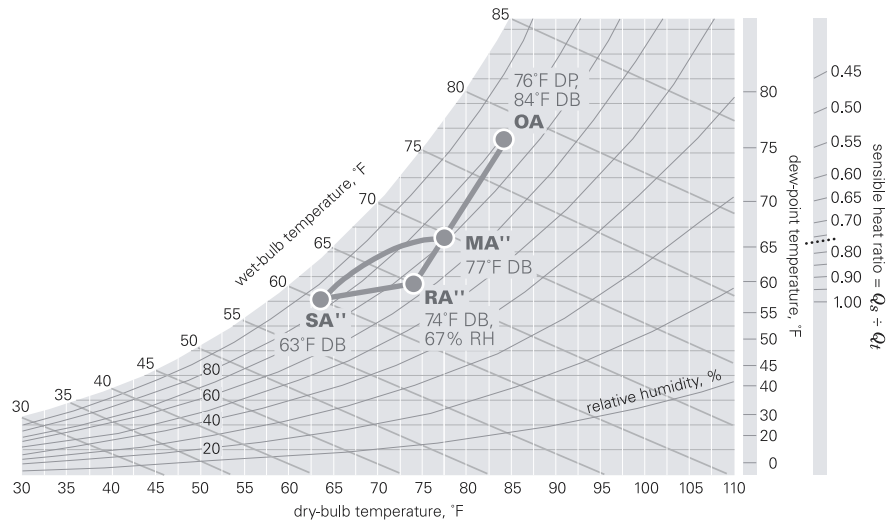
temperature for the classroom, 74°F (23.3°C). This time, the space sensible-heat ratio and supply-air temperature yield a relative humidity of 65 percent.

Repeat Steps 1–6: Because the starting and ending conditions are not the same, repeat the psychrometric analysis as often as necessary to "close the loop." The outdoor-air condition (OA) remains the same, but RA' now represents the indoor condition.

(Figure 98) For this example, completing the third iteration finally yields a space condition that matches the starting point. At the part-load, peak dew-point condition, controlling the classroom's basic constant-volume system to 74°F DB (23.3°C DB) results in a relative humidity of approximately 67 percent (RA'').

Step 7: Calculate the required cooling-coil capacity. Solving the following equation shows that the total cooling-coil load to maintain the target dry-bulb temperature is 3.66 tons (12.7 kW)…but without adequately dehumidifying the space.

$$Q_T = 4.5 \times 1,500\,cfm \times (34.1 - 27.6\,Btu/lb) = 43,875\,Btu/hr \text{ or } 3.66 \text{ tons}$$

$$(Q_T = 1.2 \times 0.7\,m^3/s \times [79.3 - 64.2\,kJ/kg] = 12.7\,kW) \blacksquare$$

Appendix B: Designing a Dedicated Outdoor-Air System

Two fundamental decisions underlie the design of a dedicated outdoor-air system and affect the selection of the dedicated outdoor-air handler and local HVAC terminals:

■ Whether to deliver the conditioned outdoor air directly to the space or to local HVAC terminals

■ Whether to condition the outdoor air to a neutral or cold dry-bulb temperature

(Figure 99) To provide the desired dehumidification, the dedicated outdoor-air handler commonly is sized to offset the local latent loads in the space, plus the sensible and latent loads of the outdoor air brought into the building for ventilation. The local HVAC terminals may also help to dehumidify the space when the sensible load is high, yielding an indoor humidity that is drier than the target. As a rule of thumb:

■ Size the dedicated outdoor-air handler so that it also offsets the space latent loads *and* the total ventilation load at the peak outdoor-enthalpy condition (pp. 112–115).

■ Size each local HVAC terminal so that it offsets the sensible-cooling load in the space at the peak dry-bulb condition. Design capacity will vary with the system configuration, that is…

　▮ neutral to space (pp. 117–118)　　▮ neutral to unit (pp. 120–122)

　▮ cold to space (pp. 119–120)　　　▮ cold to unit (pp. 122–124)

The rest of this chapter describes the selection process for each case.

Figure 99. Typical division of loads in dedicated outdoor-air systems
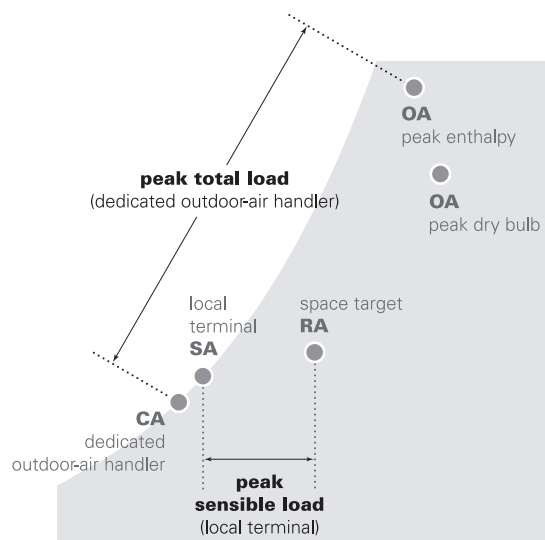
Appendix B: Designing a Dedicated Outdoor-Air System

Selecting the Dedicated Outdoor-Air Handler

The following steps establish the required airflow, dew point, and dry-bulb temperature for the conditioned air.

Step 1: Determine the entering-air condition. Three factors dictate the cooling capacity required from the dedicated outdoor-air handler: airflow, the enthalpy of the entering outdoor air, and the enthalpy of the conditioned air leaving the cooling coil. If the outdoor airflow is constant, then the basis of design is the condition resulting in the greatest difference in enthalpy across the cooling coil.

Indoor latent loads fluctuate with occupancy and processes, as well as with ambient conditions and wind through infiltration. These variables can make it difficult to discover when the greatest enthalpy difference occurs. However, if the latent loads within the space are relatively constant and infiltration is minimal, assume that the greatest enthalpy difference occurs at the highest outdoor-air enthalpy.

In most climates, the peak latent ventilation load occurs at a lower dry-bulb temperature and higher dew point than the outdoor-air condition that produces the peak sensible ventilation load. The *ASHRAE Handbook—Fundamentals* is a popular source for climatic data representing the outdoor design conditions for many locations. To aid the design of cooling and dehumidifying systems, the handbook includes:

- Peak dry-bulb and mean-coincident wet-bulb temperatures (sensible-design condition)

- Peak dew-point and mean-coincident dry-bulb temperatures (latent-design condition)

- Peak wet-bulb and mean-coincident dry-bulb temperatures (enthalpy-design condition)

Table 10 lists the 0.4 percent, cooling-design data for Jacksonville, Florida. Plotting these values on the psychrometric chart (Figure 100) illustrates that the highest outdoor enthalpy exists at the peak wet-bulb condition. In this case, the enthalpy of the outdoor air is 8 percent higher than it is at the peak dry-bulb (sensible-cooling design) condition.

Note: Using the peak dry-bulb condition as the basis of design will undersize the dedicated outdoor-air handler, making it unable to properly dehumidify the outdoor air at certain part-load conditions. Remember that the primary purpose of the dedicated outdoor-air system is to properly control space humidity at all load conditions.

The examples throughout this manual are based on "0.4 percent" data from the ASHRAE climatic data tables. This percentage indicates that the temperature is likely to equal or exceed the design value for 35 hours each year. Some design engineers choose to use more extreme conditions; others base their designs on the "1 percent" or "2 percent" values, which represent more hours. ■

Appendix B: Designing a Dedicated Outdoor-Air System

Table 10. Design weather conditions for cooling/dehumidifying in Jacksonville, Florida[1]

	Design condition	Enthalpy
Peak dry bulb, mean-coincident wet bulb	96°F (35.7°C) DB, 76°F (24.5°C) WB	39.3 Btu/lb (91.4 kJ/kg)
Peak dew point, mean-coincident dry bulb	76°F (24.6°C) DP, 84°F (28.8°C) DB	41.5 Btu/lb (96.5 kJ/kg)
Peak wet bulb, mean-coincident dry bulb	79°F (26.1°C) WB, 91°F (32.8°C) DB	42.4 Btu/lb (98.6 kJ/kg)

[1] Source: *2001 ASHRAE Handbook–Fundamentals,* Chapter 27, Table 1B (0.4% condition)

Figure 100. Comparison of outdoor-air enthalpies at 0.4% cooling-design conditions for Jacksonville, Florida (see Table 10)
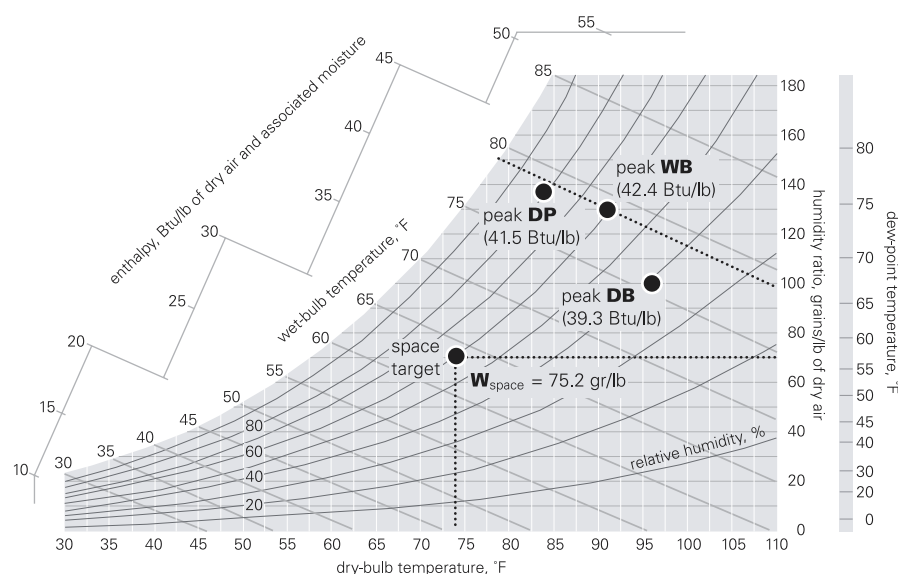
Step 2: Choose the maximum limit for space humidity. ASHRAE Standard 62–2001 recommends maintaining the indoor humidity at less than 60 percent-relative humidity (RH) to prevent high relative humidity at surfaces and thereby minimize microbial growth on building materials and furnishings. Therefore, select the dedicated outdoor-air handler with enough capacity to prevent the humidity in any space from exceeding 60 percent at the worst-case condition. The indoor humidity may be drier than 60 percent at less extreme conditions, depending on how the system is controlled.

In this example, combining the 74°F (23.3°C) setpoint for the space with a maximum relative humidity of 60 percent corresponds to a humidity ratio of 75.2 grains/lb (10.8 g/kg) or a dew point of approximately 59°F (15°C).

Appendix B: Designing a Dedicated Outdoor-Air System

Note: Selecting the unit to enforce a lower humidity limit (50 percent, for example) is possible, but requires larger equipment and consumes more energy.

Step 3: Determine the latent loads in the space. The dedicated outdoor-air handler will offset the local latent loads in the space it serves, as well as the total ventilation load. Common sources of latent load include respiration from people, processes (such as cooking), and the infiltration of humid outdoor air through cracks and other openings in the building structure; see "Sources and Effects of Indoor Moisture," p. 2.

For this example, the dedicated outdoor-air handler serves four classrooms of a school in Jacksonville, Florida. Table 11 lists the latent load for each space; in this case, the latent loads presumedly remain constant whenever the building is occupied.

Step 4: Determine the total airflow that the dedicated outdoor-air handler must deliver. According to ASHRAE Standard 62–2001, a classroom requires 15 cfm (0.007 m³/s) of outdoor air per person for ventilation. Given the design occupancies and associated outdoor-air requirements listed in Table 11, the dedicated outdoor-air handler in this example must deliver a total outdoor airflow of 1,815 cfm (0.86 m³/s).

Table 11. Design criteria for a dedicated outdoor-air system that serves four classrooms in Jacksonville, Florida (example)

Space characteristics	Classroom 101	Classroom 102	Classroom 103	Classroom 104
Sensible load	29,750 Btu/hr (8.7 kW)	26,775 Btu/hr (7.8 kW)	26,927 Btu/hr (7.9 kW)	28,262 Btu/hr (8.3 kW)
Latent load	5,250 Btu/hr (1.5 kW)	5,465 Btu/hr (1.6 kW)	5,697 Btu/hr (1.7 kW)	5,250 Btu/hr (1.5 kW)
Sensible-heat ratio (SHR)	0.85	0.83	0.83	0.84
Occupants	30	30	32	29
Required outdoor airflow	450 cfm (0.21 m³/s)	450 cfm (0.21 m³/s)	480 cfm (0.23 m³/s)	435 cfm (0.20 m³/s)
Humidity ratio of conditioned air, W_{ca}	58.3 grains/lb (8.34 g/kg)	57.6 grains/lb (8.24 g/kg)	58.0 grains/lb (8.29 g/kg)	57.7 grains/lb (8.25 g/kg)

Step 5: Determine which zone requires the driest conditioned outdoor air. Because the dedicated outdoor-air handler will offset the latent loads in each space (as well as the total ventilation load), the conditioned outdoor air must be dry enough to enforce the maximum humidity limit in the

Appendix B: Designing a Dedicated Outdoor-Air System

In the Q_L equations at right, 0.69 and 3.0 are derived from the properties of air; they are *not* constants. At the "standard" air condition, which is 69°F (21°C) dry air at sea level, the product of density, the latent heat of water vapor, and a conversion factor for units— 7,000 grains/lb (1,000 grams/kg) and 60 min/hr—equals 0.69 (3.0). A different air condition or elevation will result in a different value. ∎

worst-case space. Use the following equation to calculate the required conditioned-air humidity ratio, W_{ca}, for each space:

$$Q_L = 0.69 \times V_{oa} \times (W_{sp} - W_{ca})$$

$$(Q_L = 3.0 \times V_{oa} \times [W_{sp} - W_{ca}])$$

where,

Q_L = latent load in the space, Btu/hr (kW)
V_{oa} = conditioned outdoor airflow, cfm (m³/s), which is supplied to the space by the dedicated outdoor-air handler
W_{ca} = humidity ratio of the conditioned outdoor air, grains/lb (grams/kg)
W_{sp} = maximum limit for the humidity ratio in the space, grains/lb (grams/kg)

For example, to assure that the humidity (W_{sp}) in Classroom 101 does not exceed the maximum limit of 75.2 grains/lb (10.8 g/kg), the humidity ratio of the conditioned outdoor air, W_{ca}, must be 58.3 grains/lb (8.34 g/kg).

Table 11 shows the results of this calculation for all four classrooms. Although the highest latent load exists in Classroom 103, the "critical space" is Classroom 102 because it requires the driest air (lowest humidity ratio, W_{ca}). Supplying the conditioned outdoor air at a humidity ratio of 57.6 gr/lb (8.24 g/kg) will offset the latent load in each classroom and assure that the humidity in Classroom 102 does not exceed the maximum limit; lower humidities will result in the other classrooms.

Step 5 (Classroom 101):

$$Q_L = 5,250\ Btu/hr$$
$$= 0.69 \times 450\ cfm \times (75.2\ gr/lb - W_{ca})$$
$$\therefore W_{ca} = 58.3\ gr/lb$$

$$(Q_L = 1.5\ kW)$$
$$(= 3.0 \times 0.21\ m^3/s \times [10.8\ g/kg - W_{ca}])$$
$$(\therefore W_{ca} = 8.34\ g/kg)$$

Step 6: Determine the required dew point for the conditioned outdoor air. With the help of a psychrometric chart (Figure 101, p. 116), we find that a humidity ratio of 57.6 grains/lb (8.24 g/kg) is equivalent to a dew-point temperature of 52°F (11.1°C).

Step 6:

$$Q_T = 4.5 \times 1,815\ cfm \times (42.4 - 21.4\ Btu/lb)$$
$$= 171,518\ Btu/hr = 14.3\ tons$$

$$(Q_T = 1.2 \times 0.86\ m^3/s \times [98.6 - 49.8\ kJ/kg])$$
$$(= 50.4\ kW)$$

Dehumidifying 1,815 cfm (0.86 m³/s) of outdoor air from the peak wet-bulb condition to a 52°F (11.1°C) dew point requires 14.3 tons (50.4 kW) of cooling capacity.

Incidentally, lowering the maximum humidity limit to 50 percent would require 16.9 tons (59.4 kW)…an 18 percent increase in capacity.

Step 7: Determine the supply-air dry-bulb temperature for the dedicated outdoor-air handler. If the system design requires *neutral*-temperature conditioned air, then the air leaving the dedicated outdoor-air handler must be reheated to the desired dry-bulb temperature. This is typically between 70°F and 75°F (21°C and 24°C). For our example, assume that the air is reheated to 71°F (21.7°C).

If the system design is based on *cold* conditioned air rather than neutral-temperature air, then the dry-bulb temperature from the dedicated outdoor-air

Appendix B: Designing a Dedicated Outdoor-Air System

Figure 101. Plotting the humidity ratio to determine the equivalent dew-point temperature
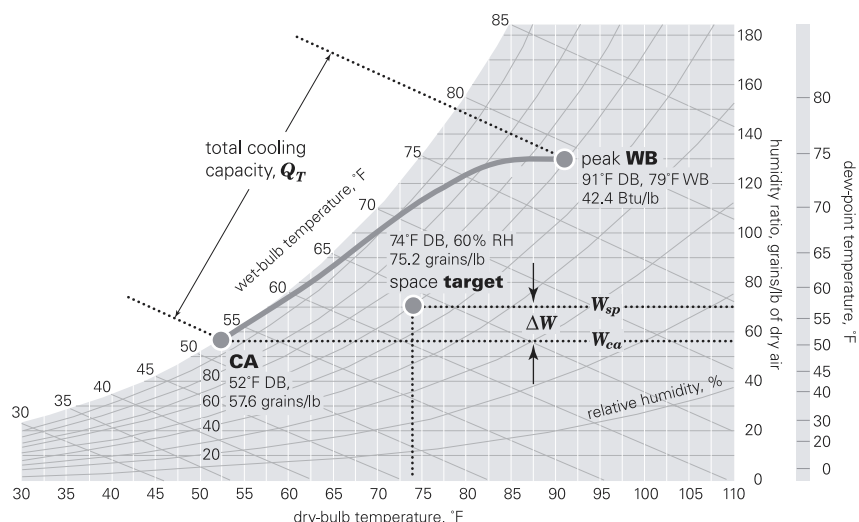
handler depends on the supply-air dew point. In our example, assuming that saturated air leaves the cooling coil, then the leaving-air dry-bulb temperature is 52°F (11.1°C).

Note: For simplicity, our example does not include the effect of fan heat. A draw-through fan arrangement will increase the dry-bulb temperature of the conditioned outdoor air. The slightly warmer air offsets less of the sensible load in the space, which will affect the selection criteria for the local HVAC terminals.

Selecting the Local HVAC Terminals

Using the dedicated outdoor-air handler to offset the latent loads in the space reduces the total cooling capacity that the local HVAC terminals must provide and, therefore, affects the selection of these units. For this discussion, a "local HVAC terminal" represents any type of air-handling equipment that controls the sensible-cooling load in the space, including fan–coils, unit ventilators, water-source heat pumps, blower–coils, air handlers (chilled water or DX cooling), packaged rooftop air conditioners, and vertical self-contained air conditioners.

The following instructions describe how to determine the required cooling capacity, supply airflow, and supply-air dry-bulb temperature for the local HVAC terminals when they are used in conjunction with a dedicated outdoor-air handler. Design criteria for the local HVAC terminals depend on *where* the conditioned outdoor air is delivered (to the space or to the unit), and at *what* dry-bulb temperature (neutral or cold). To illustrate how these factors affect the selection of HVAC terminals, the process is repeated for neutral-to-space,

Table 12. Sensible-cooling-design criteria for Classroom 101

Sensible load	29,750 Btu/hr (8.7 kW)
Latent load	5,250 Btu/hr (1.5 kW)
Sensible-heat ratio (SHR)	0.85
Required outdoor airflow	450 cfm (0.21 m³/s)
Target condition	74°F (23.3°C) DB, 60% RH max
"Neutral" conditioned outdoor air	71°F (21.7°C) DB, 52°F (11.1°C) DP
"Cold" conditioned outdoor air	52°F (11.1°C) DB, 52°F (11.1°C) DP

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cold-to-space, neutral-to-unit, and cold-to-unit systems using Classroom 101 as an example. (Classroom 101 was introduced on p. 114; Table 12 summarizes its design criteria.)

Neutral CA to space

(Figure 102) When the conditioned outdoor air (CA) is delivered directly to the space at a dry-bulb temperature that approximately equals room air, it has little effect on the sensible-cooling load in the space.

Step 1: Determine the entering-air condition for the HVAC terminal.
Because the outdoor air is delivered directly to the space, all of the air that enters the HVAC terminal's cooling coil is recirculated return air; therefore, the room-air condition becomes the entering-air condition for the local HVAC terminal. For this example, the dry-bulb temperature of the recirculated return air in Classroom 101 is 74°F (23.3°C).

Step 2: Calculate the sensible-cooling load on the HVAC terminal.
If it is cooler than room air, the conditioned outdoor air reduces the sensible-cooling load on the HVAC terminal. To determine the required cooling-coil capacity, subtract the amount of sensible cooling provided by the conditioned outdoor air $(Q_{s,\,ca})$ from the sensible-cooling load in the space $(Q_{s,\,space})$ at the peak dry-bulb condition:

$$Q_{s,\,coil} = Q_{s,\,space} - Q_{s,\,ca}$$

where,

$Q_{s,\,ca}$ = sensible cooling provided by conditioned outdoor air, Btu/hr (kW)

$Q_{s,\,coil}$ = sensible-cooling load on the terminal coil, Btu/hr (kW)

$Q_{s,\,space}$ = sensible-cooling load in the space, Btu/hr (kW)

Figure 102. Typical HVAC system with "neutral-to-space" conditioned outdoor air
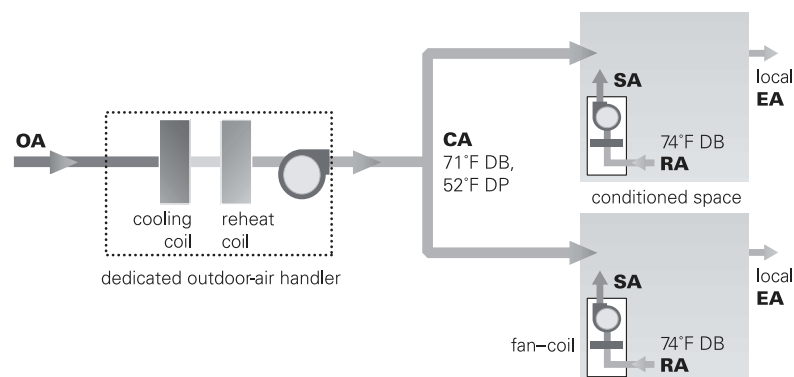
Appendix B: Designing a Dedicated Outdoor-Air System

Step 2 (Classroom 101):

$$Q_{s, ca} = 1.085 \times 450 \; cfm \times (74 - 71 °F)$$
$$= 1,465 \; Btu/hr$$

$$(Q_{s, ca} = 1.21 \times 0.21 \; m^3/s \times [23.3 - 21.7 °C])$$
$$(= 0.41 \; kW)$$

$$Q_{s, coil} = 29,750 - 1,465 = 28,285 \; Btu/hr$$

$$(Q_{s, coil} = 8.7 - 0.41 = 8.29 \; kW)$$

$Q_{s, ca}$ can be determined using the following equation:

$$Q_{s, ca} = 1.085 \times V_{oa} \times (T_{sp} - T_{ca})$$

$$(Q_{s, ca} = 1.21 \times V_{oa} \times [T_{sp} - T_{ca}])$$

where,

T_{ca} = dry-bulb temperature of the conditioned outdoor air, °F (°C)

T_{sp} = desired dry-bulb temperature for the space, °F (°C)

V_{oa} = conditioned outdoor airflow delivered to the space, cfm (m³/s)

The conditioned outdoor air in this example is only 3°F (1.6°C) cooler than the air in Classroom 101, and only slightly reduces the sensible-cooling load in the space. The HVAC terminal must offset the remaining 28,285 Btu/hr (8.29 kW).

Step 3: Determine the airflow and dry-bulb temperature for supply air from the HVAC terminal. After determining both the sensible-cooling load and the target dry-bulb temperature for the space, either:

■ Define the supply airflow and use that value to calculate the supply-air temperature; or,

■ Choose the supply-air temperature and use that value to calculate the required airflow.

If we assume that the HVAC terminal for Classroom 101 must deliver nine air changes per hour, then the required supply airflow is 1,500 cfm (0.7 m³/s), and the resulting supply-air temperature is 56.6°F DB (13.5°C DB).

Step 3 (Classroom 101):

$$Q_{s, coil} = 28,285 \; Btu/hr$$
$$= 1.085 \times 1,500 \; cfm \times (74 °F - T_{sa})$$
$$\therefore T_{sa} = 56.6 °F$$

$$(Q_{s, coil} = 8.29 \; kW)$$
$$(= 1.21 \times 0.7 \; m^3/s \times [23.3 °C - T_{sa}])$$
$$(\therefore T_{sa} = 13.5 °C)$$

$$Q_{s, coil} = 28,285 \; Btu/hr$$
$$= 1.085 \times 1,050 \; cfm \times (74 °F - T_{sa})$$
$$\therefore T_{sa} = 49.2 °F$$

$$(Q_{s, coil} = 8.29 \; kW)$$
$$(= 1.21 \times 0.5 \; m^3/s \times [23.3 °C - T_{sa}])$$
$$(\therefore T_{sa} = 9.6 °C)$$

Alternatively, a designer may choose to select the HVAC terminal so that the total airflow circulated in the space — that is, conditioned outdoor air delivered directly to the space plus return air recirculated by the HVAC terminal — equals nine air changes per hour. When we apply this strategy to Classroom 101, the HVAC terminal must provide only 1,050 cfm (0.5 m³/s) of supply air, but at the much lower supply-air temperature of 49.2°F (9.6°C) DB.

Note: Delivering cold supply air through an HVAC terminal that is located in the occupied space may create uncomfortable drafts. To mitigate this risk, either deliver the supply air through overhead, high-aspiration diffusers, which will promote better mixing and increase air throw, or select the HVAC terminals based on a supply-air temperature that is closer to 55°F (12.8°C). The design of the HVAC terminal can make it difficult to achieve low supply-air temperatures, too; the available coil configurations may dictate the combination of supply airflow and supply-air temperature.

For this example, the HVAC terminal that was selected for Classroom 101 will cool 1,500 cfm (0.7 m³/s) of recirculated return air to 56.6°F (13.5°C).

Appendix B: Designing a Dedicated Outdoor-Air System

Cold CA to space

(Figure 103) When the conditioned outdoor air is delivered directly to the space at a dry-bulb temperature that is significantly cooler than room air, it offsets a much larger portion of the sensible-cooling load in the space.

Step 1: Determine the entering-air condition for the HVAC terminal.
Because the outdoor air is delivered directly to the space, all of the air that enters the HVAC terminal's cooling coil is recirculated return air; therefore, the room-air condition becomes the entering-air condition for the local HVAC terminal. For this example, the dry-bulb temperature of the recirculated return air in Classroom 101 is 74°F (23.3°C).

Step 2: Determine the sensible-cooling load on the HVAC terminal.
If it is cooler than room air, the conditioned outdoor air reduces the sensible-cooling load on the HVAC terminal. To determine the required cooling-coil capacity, subtract the amount of sensible cooling provided by the conditioned outdoor air $(Q_{s,\,ca})$ from the sensible-cooling load in the space $(Q_{s,\,space})$ at the peak dry-bulb condition:

$$Q_{s,\,coil} = Q_{s,\,space} - Q_{s,\,ca}$$

where,

$Q_{s,\,ca}$ = sensible cooling provided by conditioned outdoor air, Btu/hr (kW)

$Q_{s,\,coil}$ = sensible-cooling load on the terminal coil, Btu/hr (kW)

$Q_{s,\,space}$ = sensible-cooling load in the space, Btu/hr (kW)

Figure 103. Typical HVAC system with "cold-to-space" conditioned outdoor air
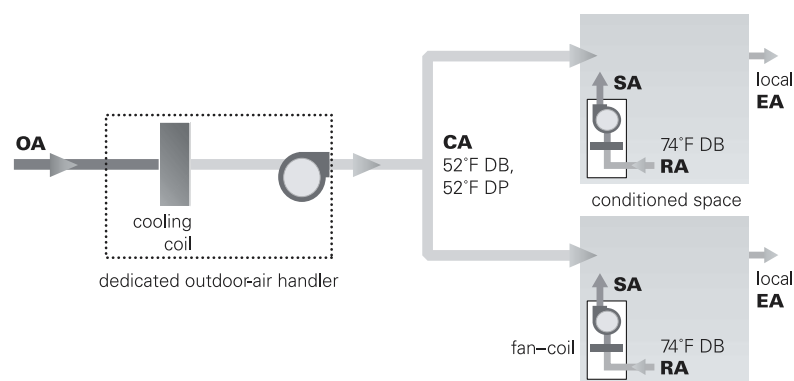
Appendix B: Designing a Dedicated Outdoor-Air System

Step 2 (Classroom 101):

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

$$(Q_{s,\,ca} = 1.21 \times 0.21\ m^3/s \times [23.3 - 11.1\,°C])$$
$$(= 3.13\ kW)$$

$$Q_{s,coil} = 29,750 - 10,742 = 19,008\ Btu/hr$$

$$(Q_{s,coil} = 8.7 - 3.13 = 5.57\ kW)$$

$Q_{s,\,ca}$ can be determined using the following equation:

$$Q_{s,\,ca} = 1.085 \times V_{oa} \times (T_{sp} - T_{ca})$$
$$(Q_{s,\,ca} = 1.21 \times V_{oa} \times [T_{sp} - T_{ca}])$$

where,
T_{ca} = dry-bulb temperature of the conditioned outdoor air, °F (°C)
T_{sp} = desired dry-bulb temperature for the space, °F (°C)
V_{oa} = conditioned outdoor airflow delivered to the space, cfm (m³/s)

The conditioned outdoor air in this example is 22°F (12.2°C) colder than the air in Classroom 101, which offsets 36 percent of the sensible-cooling load in that space. The HVAC terminal must offset the remaining 19,008 Btu/hr (5.57 kW).

Note: This example omits the effect of fan heat. A draw-through fan arrangement will increase the dry-bulb temperature of the conditioned outdoor air, which will affect the selection of the HVAC terminal.

Step 3: Determine the airflow and dry-bulb temperature for supply air from the HVAC terminal. After you know both the sensible-cooling load and the target dry-bulb temperature for the space, you can either:

■ Define the supply airflow and use that value to calculate the supply-air temperature; or,

■ Choose the supply-air temperature and use that value to calculate the required airflow.

Step 3 (Classroom 101):

$$Q_{s,coil} = 19,008\ Btu/hr$$
$$= 1.085 \times 1,050\ cfm \times (74\,°F - T_{sa})$$
$$\therefore\ T_{sa} = 57.3\,°F$$

$$(Q_{s,coil} = 5.57\ kW)$$
$$(= 1.21 \times 0.5\ m^3/s \times [23.3\,°C - T_{sa}])$$
$$(\therefore\ T_{sa} = 14.1\,°C)$$

For our example, we assumed that Classroom 101 requires nine air changes per hour, or 1,500 cfm (0.7 m³/s). Because the dedicated outdoor-air handler delivers 450 cfm (0.21 m³/s) directly to the space, the local HVAC terminal must condition only 1,050 cfm (0.5 m³/s) of recirculated return air. The required supply-air temperature is 57.3°F DB (14.1°C DB).

Neutral CA to unit

(Figure 104) When the conditioned outdoor air is delivered to HVAC terminals instead of directly to the space, it affects the condition of the mixed air that enters the HVAC terminal's cooling coil.

Step 1: Determine the sensible-cooling load on the HVAC terminal. Because the conditioned outdoor air is not delivered directly to the space, it has no direct effect on the sensible-cooling load in the space. Therefore, the sensible-cooling load for Classroom 101 is 29,750 Btu/hr (8.7 kW) at the peak dry-bulb condition.

Appendix B: Designing a Dedicated Outdoor-Air System

Figure 104. Typical HVAC system with "neutral-to-unit" conditioned outdoor air

Step 2 (Classroom 101):

$Q_s = 29,750\ Btu/hr$

$\quad = 1.085 \times 1,500\ cfm \times (74\,°F - T_{sa})$

$\quad \therefore\ T_{sa} = 55.7\,°F$

$(Q_s = 8.7\ kW)$

$\quad (= 1.21 \times 0.7\ m^3/s \times [23.3\,°C - T_{sa}])$

$\quad (\therefore\ T_{sa} = 13.1\,°C)$

To predict the relative humidity for recirculated return air from the space, perform a psychrometric analysis of the peak dry-bulb condition. ■

Step 2: Determine the airflow and dry-bulb temperature for supply air from the HVAC terminal. After you know both the sensible-cooling load and the target dry-bulb temperature for the space, you can either:

■ Define the supply airflow and use that value to calculate the supply-air temperature; or,

■ Choose the supply-air temperature and use that value to calculate the required airflow.

If we assume that the HVAC terminal for Classroom 101 must deliver nine air changes per hour, then the required supply airflow for the classroom is 1,500 cfm (0.7 m³/s) and the resulting supply-air temperature is 55.7°F DB (13.1°C DB).

Step 3: Determine the entering mixed-air condition (T_{ma}) for the HVAC terminal. Before the conditioned outdoor air passes through the cooling coil of the HVAC terminal, it mixes with recirculated return air from the space. For this example, we assumed that the recirculated return air did not pick up heat as it traveled from the space to the HVAC terminal. Therefore, 73.1°F DB (22.8°C DB) mixed air for Classroom 101 results from combining:

■ 450 cfm (0.21 m³/s) of conditioned outdoor air at 71°F DB and 52°F DP (21.7°C DB, 11.1°C DP)

■ 1,050 cfm (0.5 m³/s) of recirculated return air at 74°F DB (23.3°C DB) and 49 percent-relative humidity

A psychrometric analysis (Figure 105, p. 122) of the peak dry-bulb condition shows that the mixed-air condition (MA) lies on the straight line that would connect the conditions of the conditioned outdoor air (CA) and the return air

Appendix B: Designing a Dedicated Outdoor-Air System

Figure 105. Peak dry-bulb condition for HVAC terminals that receive neutral-temperature conditioned outdoor air (example)
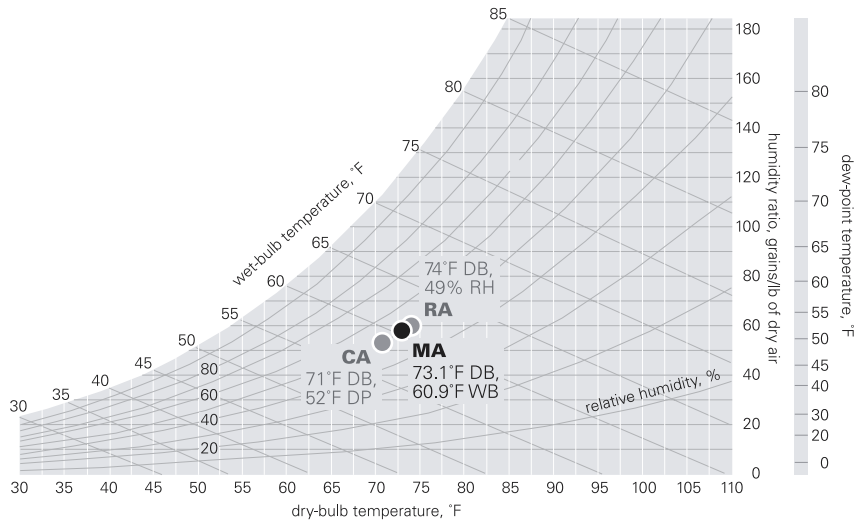
(RA) from the space. For Classroom 101, the entering mixed-air condition at the HVAC terminal is 73.1°F DB and 60.9°F WB (22.8°C DB, 16.1°C WB).

$$T_{ma} = \frac{(450\ cfm \times 71\ °F) + (1{,}050\ cfm \times 74\ °F)}{450\ cfm + 1{,}050\ cfm} = 73.1\ °F$$

$$\left(T_{ma} = \frac{[0.21\ m^3/s \times 21.7\ °C] + [0.5\ m^3/s \times 23.3\ °C]}{0.21\ m^3/s + 0.5\ m^3/s} = 22.8\ °C \right)$$

Therefore, when the HVAC terminal for Classroom 101 is part of a "neutral-to-unit," dedicated outdoor-air system, it must be selected to cool 1,500 cfm (0.7 m³/s) of mixed air from 73.1°F DB, 60.9°F WB (22.8°C DB, 16.1°C WB) to a supply-air temperature of 55.7°F (13.1°C).

Cold CA to unit

(Figure 106) When the conditioned outdoor air is delivered to HVAC terminals instead of directly to the space, it affects the condition of the mixed air entering the terminal's cooling coil.

Step 1: Determine the sensible-cooling load on the HVAC terminal.
Because the conditioned outdoor air is not delivered directly to the space, it has no direct effect on the sensible-cooling load in the space. Therefore, the sensible-cooling load for Classroom 101 is 29,750 Btu/hr (8.7 kW) at the peak dry-bulb condition.

Appendix B: Designing a Dedicated Outdoor-Air System

Figure 106. Typical HVAC system with "cold-to-unit" conditioned outdoor air

Step 2 (Classroom 101):

$Q_s = 29,750 \, Btu/hr$

$\quad = 1.085 \times 1,500 \, cfm \times (74\,°F - T_{sa})$

$\quad \therefore \, T_{sa} = 55.7\,°F$

$(Q_s = 8.7 \, kW)$

$\quad (= 1.21 \times 0.7 \, m^3/s \times [23.3\,°C - T_{sa}])$

$\quad (\therefore \, T_{sa} = 13.1\,°C)$

To predict the relative humidity for recirculated return air from the space, perform a psychrometric analysis of the peak dry-bulb condition. ∎

Step 2: Determine the airflow and dry-bulb temperature for supply air from the HVAC terminal. After you know both the sensible-cooling load and the target dry-bulb temperature for the space, you can either:

■ Define the supply airflow and use that value to calculate the supply-air temperature; or,

■ Choose the supply-air temperature and use that value to calculate the required airflow.

If we assume that the HVAC terminal must provide Classroom 101 with nine air changes per hour, then the supply airflow for the classroom is 1,500 cfm (0.7 m³/s) and the resulting supply-air temperature is 55.7°F DB (13.1°C DB).

Step 3: Determine the entering mixed-air condition (T_{ma}) for the HVAC terminal. Before the conditioned outdoor air passes through the cooling coil of the HVAC terminal, it mixes with recirculated return air from the space. If we assume that the recirculated return air from Classroom 101 does not pick up heat en route to the HVAC terminal, then:

■ 450 cfm (0.21 m³/s) of conditioned outdoor air at 52°F DB and 52°F DP (11.1°C DB, 11.1°C DP) mixes with…

■ 1,050 cfm (0.5 m³/s) of recirculated return air at 74°F DB (23.3°C DB) and 49 percent-relative humidity, resulting in 67.4°F DB (19.8°C DB) mixed air.

$$T_{ma} = \frac{(450 \, cfm \times 52\,°F) + (1,050 \, cfm \times 74\,°F)}{450 \, cfm + 1,050 \, cfm} \quad = 67.4\,°F$$

$$\left(T_{ma} = \frac{[0.21 \, m^3/s \times 11.1\,°C] + [0.5 \, m^3/s \times 23.3\,°C]}{0.21 \, m^3/s + 0.5 \, m^3/s} \quad = 19.8\,°C \right)$$

Appendix B: Designing a Dedicated Outdoor-Air System

Figure 107. Peak dry-bulb condition for HVAC terminals receiving cold, conditioned outdoor air
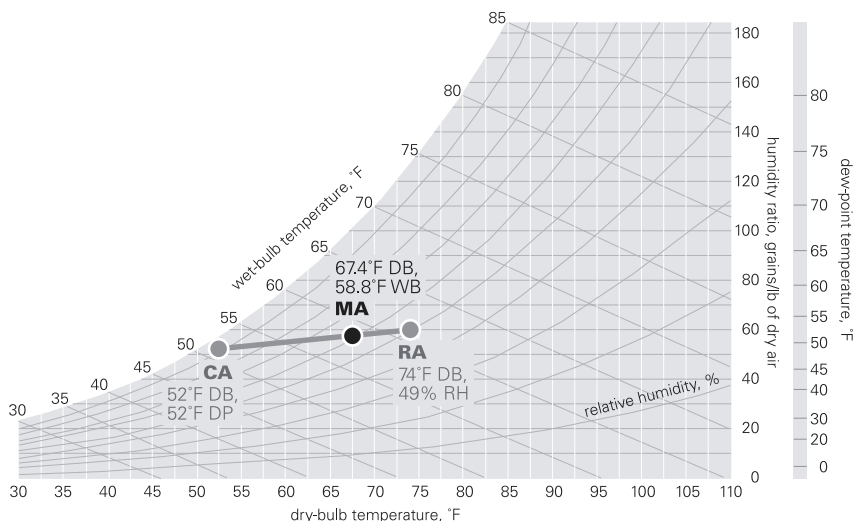
A psychrometric analysis (Figure 107) of the peak dry-bulb condition shows that the mixed-air condition (MA) lies on the straight line that connects the points representing the conditioned outdoor air (CA) and the return air (RA) from the space. For Classroom 101, the entering mixed-air condition at the HVAC terminal is 67.4°F DB and 58.8°F WB (19.8°C DB, 14.9°C WB).

Therefore, when the HVAC terminal for Classroom 101 is part of a "cold-to-unit," dedicated outdoor-air system, it must be selected to cool 1,500 cfm (0.7 m³/s) of mixed air from 67.4°F DB, 58.8°F WB (19.8°C DB, 14.9°C WB) to a supply-air temperature of 55.7°F (13.1°C).

Note: This example omits the effect of fan heat. A draw-through fan arrangement will increase the dry-bulb temperature of the conditioned outdoor air, as well as the condition of the mixed air entering the HVAC terminal. ■

Glossary

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active desiccant. See *desiccant*.

adsorption. The process by which a material accumulates moisture from the air without undergoing a chemical change.

air handler (AHU). A piece of equipment that moves and/or conditions air to maintain the thermal comfort of an indoor environment.

air-to-air energy recovery. The transfer of sensible heat, or sensible heat *plus* moisture, between two or more air streams, or between two locations within the same air stream.

airside economizer. A device and/or control strategy that conserves energy by introducing cool outdoor air to reduce the mechanical cooling requirement. Operation is based on one of the following: a fixed outdoor-air temperature; a fixed outdoor-air enthalpy; a comparison of return- and outdoor-air temperatures; *or* a comparison return- and outdoor-air enthalpies.

ANSI. American National Standards Institute (www.ansi.org)

ANSI/ASHRAE Standard 55. Titled *Thermal Environmental Conditions for Human Occupancy*, this standard specifies the combinations of indoor environments and personal factors that will produce thermal conditions acceptable to 80 percent or more of the occupants.

ANSI/ASHRAE Standard 62. Titled *Ventilation for Acceptable Indoor Air Quality*, this standard specifies minimum ventilation rates necessary to minimize the potential for adverse, building-related health effects and to maintain odor-related comfort.

ANSI/ASHRAE/IESNA Standard 90.1. Titled *Energy Standard for Buildings Except Low-Rise Residential Buildings*, this standard sets minimum design requirements that encourage energy efficiency throughout the building—that is, for the envelope, lighting, motors, HVAC, and service-water heating systems.

ASHRAE. American Society of Heating, Refrigerating, and Air-Conditioning Engineers (www.ashrae.org)

coil loop. An airside, energy-recovery device consisting of two or more finned-tube coils that are piped together in a closed loop to transfer sensible heat. A small pump circulates the working fluid through the two coils. Also called *coil runaround loop* or *runaround loop*.

conditioned outdoor air (CA). Outdoor air that has been heated, cooled, humidified, and/or dehumidified by a dedicated outdoor-air unit before being delivered directly to the conditioned space or to the mixing boxes of other local units. Also called *conditioned air*.

Glossary

constant-volume system. Type of air-conditioning system that meets the changing load conditions in the space by varying the temperature of an unchanging volume of supply air.

dedicated outdoor-air unit. An air handler or packaged air conditioner that exclusively cools, heats, dehumidifies, and/or humidifies all of the outdoor air brought into the building for ventilation. The conditioned outdoor air may be delivered directly to the conditioned space or to other air handlers or terminal equipment. Also called *makeup-air unit, 100 percent-outdoor-air unit,* or *fresh-air unit*.

desiccant. An absorbent or adsorbent substance, liquid or solid, that removes water or water vapor from a material. A *passive* desiccant system naturally releases collected moisture to a drier air stream. An *active* desiccant system releases collected moisture with the help of an external source of heat, thereby improving the moisture-collecting ability of the desiccant.

direct expansion (DX). A type of refrigerating system in which the heat-transfer medium is in direct contact with the air or fluid to be cooled or heated. For example, refrigerant inside an evaporator coil absorbs heat directly from the air passing through the coil.

dual-path air handler. An air handler that uses separate cooling coils to condition the outdoor air and the recirculated return air before the mixture is delivered to the space.

enthalpy (h). The total amount of "heat" energy, both sensible and latent, in the air at a given condition.

exhaust air (EA). Air removed from the conditioned space and expelled from the building by mechanical ventilation. *See also* recirculated return air (RRA) *and* return air (RA).

face-split coil circuiting. A type of direct-expansion, evaporator arrangement that divides the coil into parallel, independently controlled sections. Sections are activated or deactivated to match the coil capacity with the cooling load. Also called *horizontal-split coil circuiting* and *parallel-flow coil circuiting*.

fixed-plate heat exchanger. An airside, energy-recovery device that consists of thermally conductive plates. It transfers sensible heat between two air streams that flow perpendicular to each other.

heat pipe. An airside, energy-recovery device consisting of refrigerant-filled, finned tubes. Heating one end of the pipe causes the refrigerant to vaporize and migrate to the other end, where it condenses and releases sensible heat. Gravity or a capillary wick returns the liquid to the hot end of the pipe.

Glossary

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intertwined coil circuiting. A type of direct-expansion, evaporator arrangement that uses one of two distributors to feed every other tube in each coil row, dividing the coil into overlapping, independently controlled sections. Sections are activated or deactivated to match the coil capacity with the cooling load.

mixed air (MA). Combination of outdoor air (OA) and recirculated return air (RRA).

mixed-air bypass. A method of part-load coil control that changes the temperature of the supply air downstream of the coil by varying the amount of air that passes through and around the coil. Airflow is controlled by the linked face dampers and bypass dampers positioned on the entering-air side of the coil. *See also* return-air bypass.

neutral air. Supply-air condition characterized by a dry-bulb temperature and humidity ratio that do not impose a heating or cooling load in the space.

outdoor air (OA). Air brought into the building — either by a ventilation system or through openings provided for natural ventilation — from outside the building. Also called *fresh air,* but distinguished from *outside air,* which ASHRAE defines as air surrounding a refrigerated or conditioned space.

outdoor-air preconditioning. The process of cooling, dehumidifying, heating, and/or humidifying outdoor air by transferring energy (either sensible heat *or* sensible heat plus moisture) to and from the exhaust air stream.

passive desiccant. See *desiccant*.

peak dew-point condition. Design dew-point and mean-coincident dry-bulb temperatures for a specific geographic location, as tabulated in the *ASHRAE Handbook—Fundamentals*.

peak dry-bulb condition. Design dry-bulb and mean-coincident wet-bulb temperatures for a specific geographic location, as tabulated in the *ASHRAE Handbook—Fundamentals*.

peak wet-bulb condition. Design wet-bulb and mean-coincident dry-bulb temperatures, as tabulated in the *ASHRAE Handbook—Fundamentals*.

primary air (PA). Treated air that is delivered to a terminal device, where it mixes with recirculated return air via induction or fan power.

recirculated return air (RRA). Air that is taken from, and then delivered to, the conditioned space(s) — usually after passing through an air-conditioning system. *See also* return air (RA).

Glossary

return air (RA). Air removed from the conditioned space(s) and either recirculated or exhausted. *See also* recirculated return air (RRA) *and* exhaust air (EA).

return-air bypass. A method of part-load coil control that changes the temperature of the supply air downstream of the coil by mixing the air that passes through the coil with return air that bypasses the coil. Both airflows are controlled by linked face dampers and bypass dampers positioned on the entering-air side of the coil. *See also* mixed-air bypass.

sensible-design condition. Design dry-bulb and mean-coincident wet-bulb temperatures for a specific geographic location, as tabulated in the *ASHRAE Handbook—Fundamentals. See also* peak dry-bulb condition.

sensible-heat ratio (SHR). Ratio of sensible-heat gain to total (sensible plus latent) heat gain.

sensible-energy wheel. An energy-recovery device that rotates through two air streams, transferring sensible heat (temperature) from one air stream to the other. The heat-transfer media consists of a matrix of channels, which direct the two air streams in opposite directions, approximately parallel to each other. Also called a *heat wheel*.

supply air (SA). Air mechanically delivered to the conditioned space for ventilation, heating, cooling, humidification, and/or dehumidification.

supply-air tempering. The process of adding sensible heat to the air downstream of the dehumidifying coil, which allows independent control of both latent and sensible loads in the space. Only systems that directly control supply-air dew point or humidity in the space are candidates for supply-air tempering.

total-energy recovery. The transfer of sensible heat (temperature) and latent heat (moisture) between two or more air streams or between two locations within the same air stream.

total-energy wheel. A desiccant-coated, energy-recovery device that rotates through two air streams, transferring sensible heat (temperature) and latent heat (moisture) from one air stream to the other. Structured as a matrix of channels, the heat-transfer media directs the air streams through the wheel in a counterflow arrangement. The desiccant typically regenerates at room temperature. Also known as *enthalpy wheel* or *passive desiccant wheel*.

variable-air-volume (VAV) system. Type of air-conditioning system that varies the volume of constant-temperature air supplied to meet the changing load conditions in the space. ∎

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