the ABCs of DOAS

Dedicated Outdoor Air Systems

By Wayne Morris, Associate Member ASHRAE

To curb the initial cost of a building's mechanical systems, designers have expended much time and effort to use a single HVAC unit to handle ventilation as well as mechanical cooling. High-occupancy spaces, such as classrooms, pose a particular challenge. This article explores the benefits of "splitting" the cooling load, that is, using a dedicated outdoor-air unit to treat high latent-cooling load of outdoor air and zone-mounted terminal units to treat high sensible-cooling load of indoor air. Treating outdoor air *separately* makes it easy to verify sufficient ventilation airflow and enforces the maximum limit for relative humidity in occupied zones.

This article also explores techniques to minimize inefficient reheating and recooling of the outdoor air while simplifying the system and maximizing thermal comfort.

Defining the Challenge

Why is it so difficult to provide adequate dehumidification with a constant-volume, mixed-air system? The answer lies in the fact that the *sensible- and latent-cooling loads on the cooling equipment do not peak at the same time*. When it's hot outside, sensible-cooling load can exceed latent-cooling load. By contrast, when it's cool and rainy outside, latent-cooling load can approach or even exceed sensible cooling load.

Constant-volume, mixed-air HVAC units traditionally are *selected* with sufficient cooling capacity to handle the dry-bulb design load and *controlled* by a thermostat that matches the sensible-cooling capacity of the coil with the sensible-cooling load in the space. Equipment cooling *capacity,* both sensible and latent, decreases as the space sensible-cooling load drops. In most climates, the combination of less latent-cooling capacity and a lower space SHR elevates the relative humidity in the space at part-load conditions.

An "off-the-shelf," packaged unitary air conditioner may further aggravate this situation. Such equipment is designed to operate with a supply-airflow-to-cooling-capacity ratio of 350 to 400 cfm/ton (47 to 54 L/s per kW). In hot, humid climates, offsetting the ventilation load for high-occupancy spaces may require that the unit deliver no more than 200 to 250 cfm/ton (27 to 34 L/s per kW) to achieve the dew point of 45°F to 50°F (7°C to 10°C) needed for adequate dehumidification.

About the Author

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'The right combination of cooling capacity and supply airflow may not exist in packaged air conditioners....'

Selecting a packaged unitary air conditioner with enough cooling capacity (tonnage) to meet the high ventilation load results in "excess" supply airflow, that is, more supply airflow than would otherwise be needed to meet the sensible-cooling load in the space. To avoid overcooling the space, the air conditioner must deliver the supply air at a warmer dry-bulb temperature. Unfortunately, this reduces the dehumidification capacity of the coil and raises the relative humidity in the space, especially at part load. The right combination of cooling capacity and supply airflow (large compressors, small fan) simply may not exist in packaged air conditioners with prematched refrigeration and air-handling components.

Dedicated Treatment of Outdoor Air

One way to overcome the challenges imposed by a constant-volume system is to design it as a dedicated outdoor-air system (DOAS). This approach allows each component of the HVAC system to do what it does best. *Zone-level* heating and cooling equipment provides occupants with air circulation and thermal comfort by modulating cooling-coil capacity to match the sensible-cooling load in the space. Any latent cooling that occurs locally is coincidental; latent load does not affect the selection of the zone-level equipment. Meanwhile, a *central, dedicated outdoor-air unit* sufficiently dehumidifies and tempers the outdoor air to meet both the latent-cooling load and the ventilation requirements for all the spaces in the system.

DOAS's can be designed to deliver conditioned outdoor air either directly to each occupied space or to the individual terminal units or air handlers serving those spaces. The advantages and disadvantages of each configuration must be considered.

Delivering Outdoor Air Directly to the Space

The DOAS in *Figure 1* consists of unit ventilators and a dedicated outdoor-air unit, which delivers conditioned outdoor air to each occupied space via separate ductwork and diffusers. This configuration accommodates horizontal and vertical discharge arrangements as well as a wide variety of equipment, including water-source heat pumps, vertical or horizontal fan–coils, constant-volume DX rooftop units, split systems, blower–coils and through-the-wall air conditioners (PTACs).

Advantages:

• Providing a separate path for ventilation airflow makes it easier to measure the amount of outdoor air brought into the building and ensure it reaches children in occupied zones.

• Dedicated ventilation diffusers allow easy airflow measurement and balancing during system installation.

• Separately conditioning the outdoor air avoids imposing ventilation loads on the local HVAC units, circumventing the airflow/ton challenge of providing adequate dehumidification capacity without overcooling the space.

Disadvantages:

• Providing the extra ductwork and diffusers needed for parallel air paths (outdoor air and recirculated return air) can increase the initial cost of the system.

• Delivering conditioned outdoor air and recirculated air through separate diffusers may not adequately mix the two air streams within the space.

• Separate, parallel paths for ventilation air and recirculated return air increase the total system airflow to the space, which may slightly increase overall fan-energy consumption.

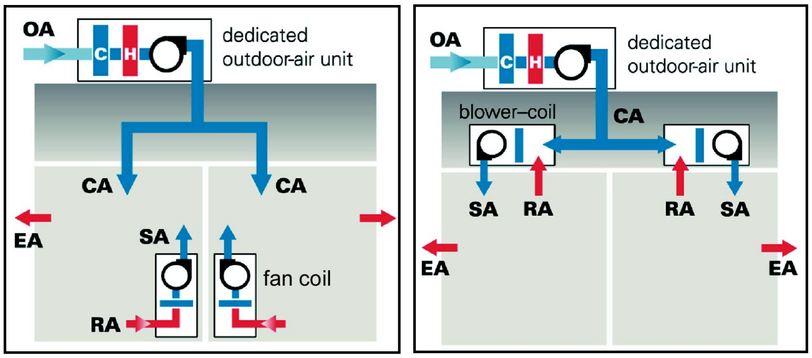


Figure 1 (left): Conditioned OA delivered directly to the space. Figure 2 (right): Conditioned OA delivered to local terminal units or air handlers.

Delivering OA to Local Terminal Units

The DOAS in *Figure 2* also uses a dedicated outdoor-air unit to handle the ventilation load. Ductwork carries the conditioned outdoor air to each terminal unit (typically blower-coils, horizontal fan-coils, or water-source heat pumps), discharging it near or directly into the inlet. The conditioned outdoor air then mixes with recirculated return air and passes through the cooling coil of the terminal unit, which delivers the mixed air to the space.

Advantages:

• Using the same ductwork and diffusers to deliver ventilation air and recirculated return air to the space avoids the additional first cost for separate air paths.

• Delivering ventilation air and recirculated return air together (rather than via separate paths) result in a slightly lower total air volume to the space, which may result in slightly less fan-energy consumption than a direct-to-space DOAS design.

• Ventilation air and recirculated return air mix thoroughly before entering the space through the supply-air diffusers, making it easier to achieve uniform thermal comfort throughout the occupied space.

Disadvantages:

• The local terminal unit must operate whenever the space requires ventilation. Control of the terminal units and dedicated outdoor unit must be coordinated to provide both adequate dehumidification during unoccupied periods and a pre-occupancy purge cycle.

• If the conditioned outdoor air is delivered near, rather than directly into, the terminal units, it may be necessary to bring more outdoor air into the building so that each space receives sufficient ventilation.

• Balancing the ventilation airflow may be more difficult because the mixing components are located in the ceiling rather than in the space.

Practical Example: Selecting a Dedicated OA Unit

Let's see how the procedure outlined in this article can help to size a dedicated outdoor-air (OA) unit that provides ventilation directly to four classrooms in a Jacksonville, Fla. school:

1. Peak enthalpy condition (0.4% design wet-bulb condition from *ASHRAE Handbook—Fundamentals*):

T_{oa} = 91°F DB, 79°F WB

2. Target space condition (designer's choice):

T_{sp} = 74°F DB (setpoint)
RH_{sp} = 60% (maximum limit)
$\therefore W_{sp}$ = 75.2 gr/lb

3. Space latent and sensible loads (from load calculations) in Btu/h:

${}^{Rm\,1}Q_l$ = 5,250; ${}^{Rm\,1}Q_s$ = 29,750
${}^{Rm\,2}Q_l$ = 5,465; ${}^{Rm\,2}Q_s$ = 26,775
${}^{Rm\,3}Q_l$ = 5,697; ${}^{Rm\,3}Q_s$ = 26,927
${}^{Rm\,4}Q_l$ = 5,250; ${}^{Rm\,4}Q_s$ = 28,262

4. Outdoor airflow (based on ANSI/ASHRAE Standard 62-2001) in cfm:

${}^{Rm\,1}V_{oa}$ = 15 cfm/p × 29 people = 435
${}^{Rm\,2}V_{oa}$ = 15 cfm/p × 30 people = 450
${}^{Rm\,3}V_{oa}$ = 15 cfm/p × 32 people = 480
${}^{Rm\,4}V_{oa}$ = 15 cfm/p × 29 people = 435
${}^{System}V_{ot} = \sum (V_{oa})$ = **1,800 cfm**

5. Key humidity-ratio rise (calculated) in gr/lb, that is, $\Delta W = Q_l / (0.69 \times V_{oa})$:

${}^{Rm\,1}\Delta W$ = 5,250 / (0.69 × 435) = 17.5
${}^{Rm\,2}\Delta W$ = 5,465 / (0.69 × 450) = **17.6**
${}^{Rm\,3}\Delta W$ = 5,697 / (0.69 × 480) = 17.2
${}^{Rm\,4}\Delta W$ = 5,250 / (0.69 × 435) = 17.5

6. Conditioned-air dew point (calculated):

$W_{ca} = W_{sp} - \Delta W$
= 75.2 – 17.6 = 57.6 gr/lb
$\therefore T_{ca}$ = **52°F DP**

7. Conditioned-air dry bulb (designer's choice):

T_{ca} = **71°F DB** (neutral), with reheat
or
T_{ca} = **52°F DB** ("cold"), no reheat

Select the dedicated OA unit to deliver **1,800 cfm** of ventilation air at a dew point of 52°F when the outdoor temperatures are 91°F DB and 79°F WB. To deliver neutral-temperature air, include reheat to temper the ventilation air to 71°F DB.

Designing a Dedicated Outdoor Air System

Local terminal units may perform latent cooling when the sensible-cooling load in the space is high. However, given the uncertainty of the latent-cooling capacity available from the terminal unit at part-load conditions, it is prudent to size the outdoor air unit (OAU) to handle the *entire* latent (indoor *and* outdoor) load on the system. This approach may result in spaces that are slightly drier than the design target, but it will ensure relative humidity does not exceed the maximum limit.

Selecting the Outdoor Air Unit

Required airflow, dew point, and dry-bulb temperature of the conditioned outdoor air (CA) are key to selecting the dedicated OA unit. The inset,"Practical Example: Selecting a Dedicated OA Unit," demonstrates the selection logic described later.

1. Determine the enthalpy-based design condition. The weather (entering-coil) condition with the highest enthalpy—that is, peak wet bulb and mean coincident dry bulb—generally sets the peak cooling capacity required from the dedicated OA unit. The *ASHRAE Handbook—Fundamentals* provides climatic design data for various geographical locations.

2. Choose the target condition for the occupied space. Most designers choose a dry-bulb temperature of 72°F to 76°F (22°C to 24°C) as the setpoint. To discourage microbial growth, ANSI/ASHRAE Standard 62–2001, *Ventilation for Acceptable Indoor Air Quality,* recommends a relative humidity of 60% or less during cooling. Therefore, select the dedicated OA unit to limit the relative humidity in all spaces to no more than 60%.

Designing a dedicated ventilation system that limits the space relative humidity to a lower value (50%, for example) requires larger equipment and increases energy consumption.

3. Determine the latent load for each space. Latent loads in spaces, which are usually unaffected by weather conditions, can be calculated with the help of load-design software.

4. Calculate the system-level outdoor airflow by finding and summing the minimum ventilation-airflow values of the occupied spaces. Minimum ventilation requirements are set by local codes or by ANSI/ASHRAE Standard 62-2001.

5. Determine the largest rise among the space humidity ratios. Constant-volume terminal units without dehumidification enhancements may not remove enough moisture at a partial sensible load. By providing conditioned air that is drier than the air in each space, the dedicated OA unit can offset the local latent loads and maintain the relative humidity at or below the 60% maximum limit.

Note: The space with the highest latent load may not be the space with the largest humidity-ratio increase.

6. Calculate the required humidity ratio and corresponding dew point for the conditioned air delivered by the dedicated OA unit. To do so, subtract the largest humidity-ratio rise from the target humidity ratio for the spaces.

The dedicated ventilation system in *Figure 1* removes the entire outdoor load *and* the latent load generated within each space. When the "latent-critical" space is at the relative humidity limit, all other spaces are drier.

Note: Designs that deliver conditioned air at a humidity ratio or dew-point temperature that equals the space target cannot handle local latent loads. These designs may result in a smaller dedicated OA unit, but they require larger terminal units and usually result in a higher relative humidity.

7. Choose the required dry-bulb temperature of the conditioned air. If the system design requires neutral-temperature conditioned air (that is, conditioned air that approximates the target for the space), pick a dry-bulb value of 70°F to 75°F (21°C to 24°C).

If the design requires "cold" ventilation air, use a dry-bulb temperature that is about equal to the required dew point.

Practical Example: Effect On Terminal-Unit Sizing

If the dedicated outdoor-air (OA) unit handles the peak latent load for the four-classroom system in our Jacksonville, Florida example, then the terminal units only need sufficient capacity to satisfy the peak sensible load.

Select the terminal units to deliver the required sensible capacity (depending on whether the dedicated OA unit delivers neutral or "cold" conditioned air, T_{ca}) at the assumed supply-air temperature and calculated supply airflow.

Procedure	Neutral CA (T_{ca} = 71°F DB)	"Cold" CA (T_{ca} = 52°F DB)
1. Target space condition, T_{sp}, (designer choice)	74°F DB	74°F DB
2. Sensible cooling provided by ventilation system (calculated), that is, $Q_{ca} = 1.085 \times V_{oa} \times (T_{sp} - T_{ca})$:		
${}^{Rm\,1}Q_{ca} = 1.085 \times 435 \times (74 - T_{ca}) =$	1,416 Btu/h	10,383 Btu/h
${}^{Rm\,2}Q_{ca} = 1.085 \times 450 \times (74 - T_{ca}) =$	1,465 Btu/h	10,742 Btu/h
${}^{Rm\,3}Q_{ca} = 1.085 \times 480 \times (74 - T_{ca}) =$	1,562 Btu/h	11,458 Btu/h
${}^{Rm\,4}Q_{ca} = 1.085 \times 435 \times (74 - T_{ca}) =$	1,416 Btu/h	10,383 Btu/h
3. Sensible cooling load provided by terminal unit (calculated), that is, $Q_{tu} = Q_s - Q_{ca}$:		
${}^{Rm\,1}Q_{tu} = 29{,}750 - {}^{Rm\,1}Q_{ca} =$	28,334 Btu/h	19,367 Btu/h
${}^{Rm\,2}Q_{tu} = 26{,}775 - {}^{Rm\,2}Q_{ca} =$	25,310 Btu/h	16,033 Btu/h
${}^{Rm\,3}Q_{tu} = 26{,}927 - {}^{Rm\,3}Q_{ca} =$	25,365 Btu/h	15,469 Btu/h
${}^{Rm\,4}Q_{tu} = 28{,}262 - {}^{Rm\,4}Q_{ca} =$	26,846 Btu/h	17,879 Btu/h
4. Supply airflow from terminal unit (given supply air temperature), that is, $V_{tu} = Q_{tu}/[1.085 \times (T_{sp} - T_{sa})]$		
${}^{Rm\,1}V_{sa} = Q_{tu1}/[1.085 \times (74 - 55)] =$	1,374 cfm	939 cfm
${}^{Rm\,2}V_{sa} = Q_{tu2}/[1.085 \times (74 - 55)] =$	1,228 cfm	778 cfm
${}^{Rm\,3}V_{sa} = Q_{tu3}/[1.085 \times (74 - 55)] =$	1,230 cfm	750 cfm
${}^{Rm\,4}V_{sa} = Q_{tu4}/[1.085 \times (74 - 55)] =$	1,302 cfm	867 cfm

Selecting the Terminal Units

How you select the dedicated OA unit can significantly affect the required cooling capacity (supply airflow and dry-bulb temperature) for each terminal unit. The inset titled "Practical Example: Effect on Terminal-Unit Sizing" demonstrates the following selection logic. *Note: This example reflects OA delivered directly to the space.*

1. Determine the entering air temperature. In this example, the terminal units "see" only recirculated return air, which has a temperature equaling the space target.

2. Calculate the sensible cooling provided by the ventilation system. *"Cold"* conditioned air provides more sensible cooling than *neutral* conditioned air, which is slightly cooler than the space target.

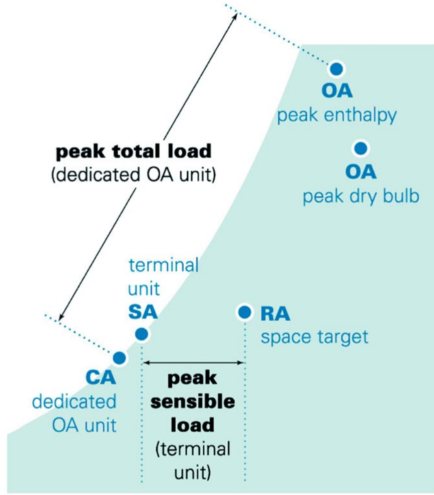
3. Determine the sensible cooling load for each terminal unit. Subtract the sensible cooling provided by the ventilation system from the peak sensible load (at the peak dry-bulb temperature), which was calculated for each occupied space with the help of load-design software.

4. Find the required supply airflow for each terminal unit, given an arbitrary supply-air temperature and the sensible cooling load that the terminal unit must satisfy. Unit placement and geometry influence the selection of a target supply-air temperature, as does the design of the dedicated ventilation system. Delivering "cold" conditioned air reduces the required terminal-unit airflow *(and* permits smaller terminal-unit cabinets) than a system that delivers neutral-temperature conditioned air.

Figure 3 offers a psychrometric summary of the equipment-selection conditions.

Ways to Improve Efficiency

Various options such as **energy recovery** can improve operating effi-



When designing a dedicated outdoor-air system (DOAS) that delivers conditioned air directly to occupied spaces:

- Size the dedicated OA unit so that it also handles the latent loads in the spaces at the peak enthalpy condition.
- Size each terminal unit to handle the sensible cooling load in the space at the peak dry-bulb condition.

Figure 3: DOAS selection summary.

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ciency. If the dedicated outdoor air unit delivers neutral-temperature air, recovered or transferred energy can provide *tempering* (reheat). Such designs can recover heat from the refrigeration circuit in a direct-expansion (DX) unit or from an airstream, usually by arranging an air-to-air heat exchanger in series with the cooling coil.

Using a total-energy wheel (or any other air-to-air heat exchanger) to *precondition* the entering outdoor air reduces both cooling and heating loads. Smaller loads not only enable you to downsize the dedicated OA unit, but can also reduce operating cost. To take advantage of this benefit, however, most of the building exhaust must be routed back to the air handler.

Setpoint reset can improve system efficiency, too. Raising the humidity ratio (dew point) of the conditioned air while limiting the humidity in the critical space avoids "over-drying" and lowers the operating cost of the dedicated OA unit. Similarly, raising the conditioned-air dry-bulb temperature while avoiding heating in all spaces can reduce the cost of operating the terminal units if the dedicated OA unit uses recovered energy for reheat. It also delays cooling-to-heating changeover in two-pipe systems.

To effectively control relative humidity, **unoccupied dehumidification** may be necessary. A dedicated OA unit with DX cooling can provide this function without operating chillers and cooling towers. Implementation requires a return-air path to the dedicated OA unit and a humidity sensor in the critical space.

Key Concepts

- **Always provide conditioned air that is drier than the space.** This practice minimizes the required terminal-unit capacity and adequately controls the indoor relative humidity *without* additional dehumidification enhancements in the terminal units.
- **Deliver "cold" conditioned air whenever possible.** "Neutral" conditioned air increases the capacity needed from the terminal units and requires reheat at the dedicated OA unit.
- **Select equipment to limit the relative humidity to 60%.** Targeting a drier-than-necessary condition in the space requires larger equipment and consumes more energy.

The design approach outlined in this article greatly simplifies the task of creating a comfortable, well-ventilated indoor environment using constant-volume, mixed-air systems. It assigns a dedicated outdoor unit to treat the ventilation air and to manage the latent-cooling load for the building. The sensible-cooling load in the space is addressed independently by local terminal units. Dividing the building's total cooling load in this fashion makes it easier to effectively ventilate and dehumidify high-occupancy spaces such as classrooms.