design tips for effective, efficient
Dedicated Ventilation Systems

from the editor…
Often the Engineers Newsletter *limits its coverage of design concepts to theoretical discussions. This issue tackles the subject of ventilation system design, but does so by adopting a practical, workaday approach. If you design HVAC systems for multiple-space applications, we think you'll appreciate this* EN*'s "how-to" instructions for determining equipment capacities. You'll also find ideas for improving the efficiency of dedicated ventilation systems.*

For years, most HVAC installations have been "mixed-air" systems in which an air handler combined outdoor air (OA) and return air, then conditioned the resulting mixture and delivered it to occupied spaces in the building. Such systems made it easy to provide proper ventilation: the designer could set the outdoor-air damper or outdoor airflow to accommodate the "worst-case" ventilation requirements. Although this simplistic solution satisfied the immediate concern, indoor air quality, it also required high outdoor airflow to compensate for the low ventilation efficiency of the system.

"Ventilation efficiency" describes the percentage of first-pass outdoor air that dilutes indoor contaminants before being exhausted from the building. The higher the percentage of "used" ventilation air, the more efficient the system.

Ventilation becomes less efficient when the mixed-air system serves multiple spaces with differing ventilation needs. One space (the "critical space") receives precisely enough outdoor air for adequate ventilation; all other spaces are overventilated. There are various ways to improve ventilation efficiency, but the **dedicated ventilation system** is an increasingly popular solution.

Cold or Neutral Conditioned Air?

When some spaces ventilated by a dedicated OA unit need heating while others need cooling, designers usually choose to deliver *neutral-temperature* conditioned air. Dry conditioned air must be reheated to the desired dry-bulb temperature (a subjective value that typically ranges from 70°F to 74°F). Neutral-temperature air minimizes the additional heating load that the dedicated OA unit imposes on the terminal units.

If most of the spaces ventilated by the dedicated OA unit are thermally similar (that is, *all* of them need either heating or cooling), designers may choose to deliver *"cold"* conditioned air. The conditioned-air dry-bulb temperature depends on the required supply-air dew point, typically between 50°F and 55°F. "Cold" conditioned air means smaller, therefore less costly, terminal units. ■

What Is a Dedicated Ventilation System?

A dedicated ventilation system uses a separate air handler to condition the outdoor air before delivering it to the outdoor/mixed-air intakes of other HVAC units or directly to the occupied spaces. In either case, terminal units handle the local space loads.

Whether it is called a *dedicated OA unit* (as in this *EN*), a *100-percent OA unit,* or a *fresh air unit,* the air handler must dehumidify, cool, heat, and sometimes humidify the incoming outdoor air. For now, let's focus our attention on dehumidification and cooling…

Most unit designs use chilled water or DX coils to cool the air and either cold-coil or active-desiccant technology to dry it. For this discussion, we'll look only at cold-coil dehumidification because it is less expensive, less complex, and more widely understood and applied than active-desiccant technology. Also, cold-coil designs sufficiently dehumidify conditioned air for most commercial and institutional applications.

Implications for System Design

To demonstrate how a dedicated ventilation system affects equipment selection, let's consider the system in Figure 1 (page 2). A dedicated OA unit meets the minimum ventilation requirement by treating the outdoor air and then delivering the conditioned air (CA) directly to the occupied spaces. Fan–coils, controlled by thermostats,

satisfy the local sensible load in each space.
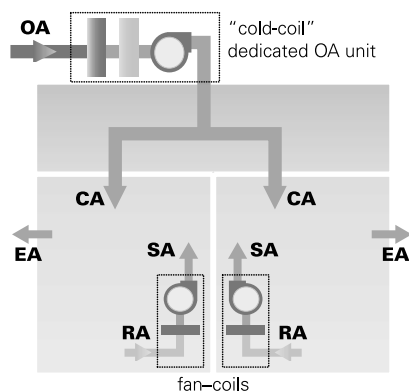
Selecting the dedicated OA unit.

Required airflow, dew point, and dry-bulb temperature of the conditioned air (CA) are key to the selection of the dedicated OA unit. The inset at right demonstrates the selection logic described below.

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 as the setpoint. To discourage microbial growth, ASHRAE Standard 62–1999 recommends a relative humidity of 60 percent or less during cooling. Therefore, select the dedicated OA unit to limit the relative humidity in all spaces to no more than 60 percent.

Designing a dedicated ventilation system that limits the space relative humidity to a lower value (50 percent, for example) adds unnecessary cost

Practical Example: Selecting a Dedicated OA Unit

Let's see how the procedure outlined on pages 2–3 can help us size a dedicated outdoor-air (OA) unit that provides ventilation directly to four classrooms in a Jacksonville, Florida school:

1 Peak enthalpy condition (from *ASHRAE Handbook–Fundamentals*):
T_{oa} = 91°F DB, 79°F WB

2 Target space condition (designer's choice):
T_{ra} = 74°F DB (setpoint)
RH_{ra} = 60% (maximum limit)
∴ W_{ra} = 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 ASHRAE Standard 62–1999) 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} = $W_{ra} - \Delta W$
= 75.2 – 17.6 = 57.6 gr/lb
∴ 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.

Figure 1. Dedicated Ventilation System

because it 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 ASHRAE Standard 62–1999.

5 Determine the largest rise among the space humidity ratios. As noted in a previous *EN* (volume 29, number 4), constant-volume terminal units without dehumidification enhancements usually don't 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-percent maximum limit.

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

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 higher indoor relative humidities.

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.

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

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 on this page demonstrates the following selection logic.

1 Determine the entering air temperature. 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. (See "Cold or Neutral Conditioned Air?" on page 1.)

3 Determine the sensible cooling load for each terminal unit. Subtract the sensible cooling of 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 required terminal-unit airflow *(and* permits smaller terminal-unit cabinets) than a system that delivers neutral-temperature conditioned air.

Figure 2 offers a psychrometric summary of the equipment-selection conditions.

Figure 2. Selection Summary

When designing a dedicated ventilation system 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 load in the space at the peak sensible (dry-bulb) condition.

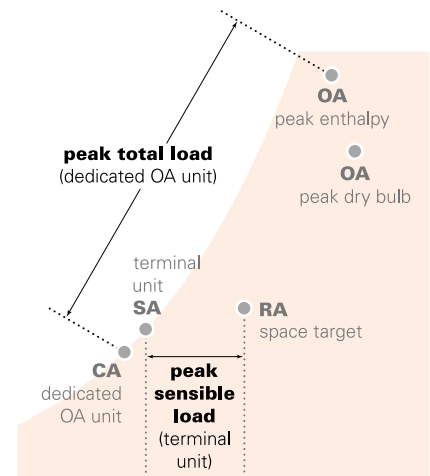## Practical Example: Effect on Terminal-Unit Sizing

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_{ra}, (designer choice)	74°F DB	74°F DB
2 Sensible cooling provided by ventilation system (calculated), that is, $Q_{ca} = 1.085 \times V_{oa} \times (T_{ra} - T_{ca})$:		
$^{Rm\,1}Q_{ca} = 1.085 \times 435 \times (74 - T_{ca}) =$	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, $V_{tu} = Q_{tu}/[1.085 \times (T_{ra} - T_{sa})]$:		
$^{Rm\,1}V_{sa} = Q_{tu1}/[1.085 \times (74 - 55)] =$	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

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Ways to Improve Efficiency

Various options such as **energy recovery** can improve operating efficiency. If the ventilation system delivers neutral-temperature air, recovered or transferred energy can provide *tempering*. Such designs can recover heat from the refrigeration circuit in a direct-expansion (DX) unit or from an air stream, 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. (For details, see *EN* volume 29, number 5.)

Although most dedicated OA units are designed to deliver a constant flow of conditioned air, applications with fluctuating occupancies may benefit from **variable airflow**. Automatically adjusting the flow of conditioned air to match the system's current ventilation requirements can reduce operating costs without sacrificing indoor air quality.

Adding an **airside economizer** also reduces operating costs, but requires a duct system that is entirely separate from the dedicated OA unit. That's because the dedicated OA unit is usually selected to accommodate only minimum outdoor airflow and does not handle recirculated return air. (Alternatively, airside economizing can occur at the terminal units, but effective operation also requires a means for modulated exhaust.)

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 "overdrying" and lowers the operating cost of the dedicated OA unit. Similarly, raising the conditioned-air dry-bulb temperature while avoiding heating in all spaces reduces the cost of operating the terminal units. 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 percent.** Targeting a drier-than-necessary condition in the space requires oversized equipment and consumes more energy. ■

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