Common Pitfalls in Design And Operation of a DOAS

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A dedicated outdoor air system (DOAS) uses separate equipment to condition all of the outdoor air brought into a building for ventilation and delivers it to each zone, either directly or in conjunction with local HVAC equipment serving those same zones. The local HVAC equipment is used to control zone temperature (*Figure 1*).

The *ASHRAE Design Guide for Dedicated Outdoor Air Systems* was published in 2017.[1] As part of that project, the authors conducted interviews with various industry experts, along with site visits and reviews of several projects that incorporated a DOAS. This feedback helped steer the content toward its goal of being a practical guide for the working-level HVAC designer.

Based in part on that feedback, this article discusses five pitfalls commonly seen in the design and control of a DOAS, and ways to avoid them.

Insufficient Dehumidification

In most climates, the dedicated OA unit should be sized to dehumidify the outdoor air to a dew point that is drier than the zone (*Figure 2*). This will remove the sensible and latent loads associated with ventilation, and offset some (or all) of the latent loads in the zones; thus requiring the zone-level HVAC equipment to only offset the sensible cooling loads in the zones.[1,2,3]

This practice can adequately limit indoor humidity levels at all load conditions, without the need for added dehumidification enhancements in the zone-level HVAC equipment.

The following equation is used to calculate how dry the air must be in order for the zone outdoor airflow to offset the entire latent load in each zone, such that the indoor humidity level is maintained at or below the desired upper limit:

$$W_{CA} = W_{zone} - Q_{latent,zone}/(0.69 \times V_{oz})$$

$$[W_{CA} = W_{zone} - Q_{latent,zone}/(3.0 \times V_{oz})]$$

where

W_{CA} = required humidity ratio of the conditioned air leaving the unit, grains/lb (g/kg)

W_{zone} = desired upper limit for humidity ratio of the occupied space, grains/lb (g/kg)

$Q_{latent,zone}$ = design latent load in the zone, Btu/h (W)

V_{oz} = design zone outdoor airflow, cfm (L/s)

[Note: In the equations above, 0.69 (3.0) is not a constant, but is derived from the properties of air at

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FIGURE 1 Example DOAS with various types of zone-level HVAC equipment.

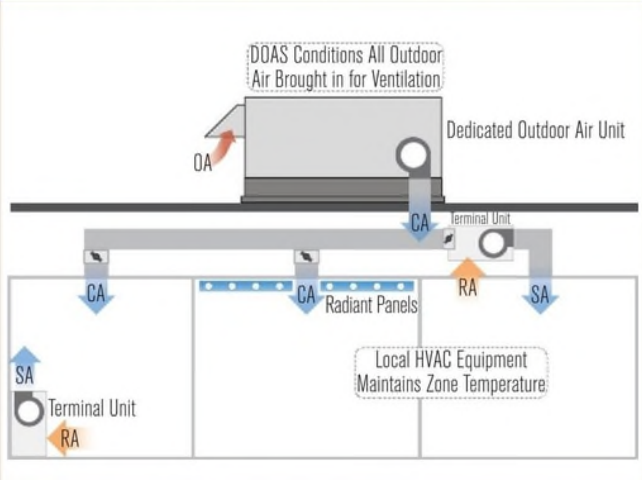
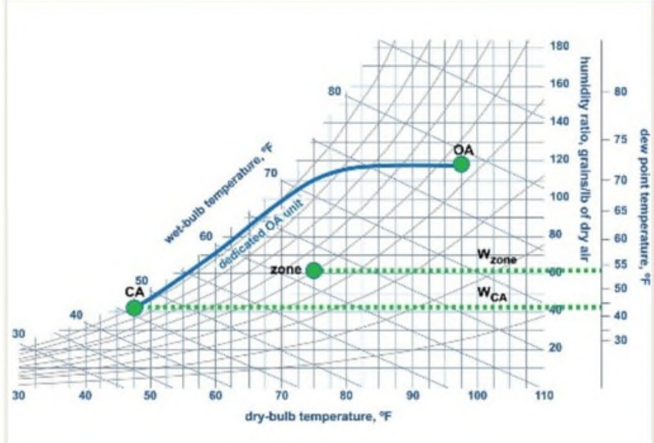


FIGURE 2 Dehumidify the OA to a dew point drier than the zone.



"standard" conditions. Air at other conditions and elevations will cause this factor to change.]

Fear of Over-Cooling Zones

Some dedicated OA systems are designed to dehumidify the outdoor air to a low dew point, and then reheat it to approximately zone dry-bulb temperature ("neutral" air). However, when a chilled-water or direct-expansion (DX) coil is used for dehumidification, a by-product of that process is that the dry-bulb temperature of the air leaving the coil (CC) is colder than the zone. If this dehumidified outdoor air is reheated to neutral (CA), the sensible cooling performed by the dedicated OA unit is wasted (*Figure 3*).

In contrast, if the dedicated OA unit dehumidifies the outdoor air, but then delivers the conditioned air "cold" (not reheated to neutral), the cold dry-bulb temperature offsets part of the sensible cooling load in the zone. This often results in downsized local HVAC equipment and lower overall system energy use.[1,4]

The decision to design for "neutral" air often seems to stem from the designer's fear of overcooling the zone under low-load conditions. But in many applications, this fear is likely unwarranted (or at least overstated).

Consider an example k–12 school classroom, in which the sensible heat gain from each occupant is 250 Btu/h (73 W) and the zone temperature setpoint (DBT_{zone}) is 73°F (23°C). Using the default occupant density, ASHRAE Standard 62.1-2016 requires 13 cfm (6.7 L/s) of outdoor air per person. If the dedicated OA unit delivers the conditioned outdoor air (CA) at 55°F (13°C) dry bulb, the sensible cooling provided by this conditioned outdoor air ($Q_{sensible,CA}$) is 254 Btu/h per person (81 W/person):

$$\begin{aligned} Q_{sensible,CA} &= 1.085 \times V_{oz} \times (DBT_{zone} - DBT_{CA}) \\ &= 1.085 \times 13 \text{ cfm/person} \times (73°\text{F} - 55°\text{F}) \\ &= 254 \text{ Btu/h per person} \end{aligned}$$

$$[Q_{sensible,CA} = 1.21 \times 6.7 \text{ L/s per person} \times (23°\text{C} - 13°\text{C}) = 81 \text{ W/person}]$$

[Note: In the equations above, 1.085 (1.21) is not a constant, but is derived from the properties of air at "standard" conditions. Air at other conditions and elevations will cause this factor to change.]

In this case, the outdoor airflow delivered to the classroom almost exactly offsets the sensible heat gain from the occupants. For this application, implementing demand-controlled ventilation—to reduce outdoor airflow when people leave the zone—would be a more efficient approach to avoid over-cooling, rather than reheating the dehumidified OA to a "neutral" temperature at all times.

Therefore, deliver the conditioned OA at a cold (not "neutral") temperature, whenever possible. To avoid over-cooling zones at low-load conditions:

- Implement demand-controlled ventilation;
- Activate heat in the zone-level HVAC equipment (this is likely more efficient when only a few zones are being over-cooled); and
- Reheat the dehumidified air at the dedicated OA unit using recovered energy, but only when necessary to prevent over-cooling.

FIGURE 3 Reheating wastes the sensible cooling performed by the DOAS.

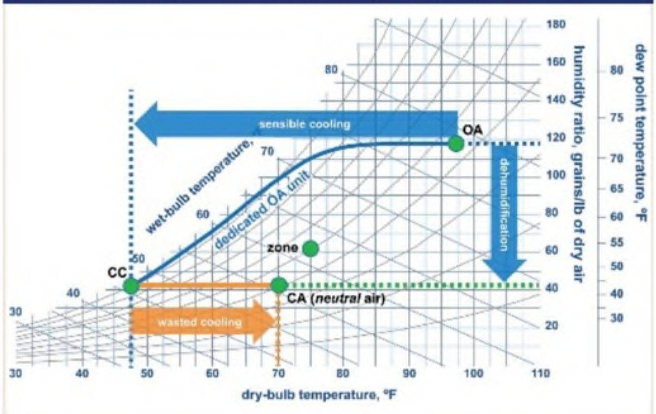
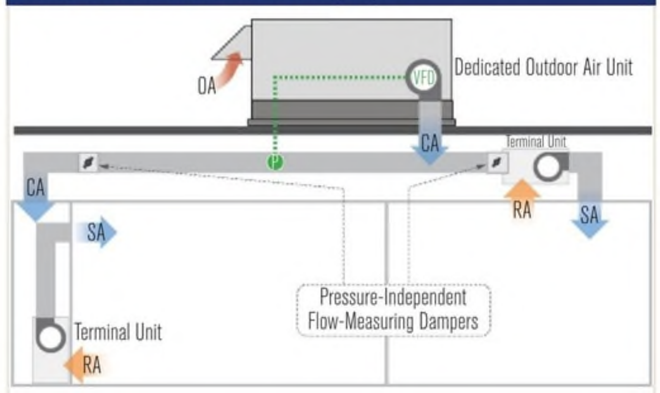


FIGURE 4 Pressure-independent, flow-measuring dampers ensure required outdoor airflow to each zone as terminal fan changes speed.



Further, ASHRAE/IES Standard 90.1-2016 now prohibits reheating (or heating) this ventilation air above 60°F (16°C) whenever most zones require cooling:

> "**6.5.2.6** Ventilation Air Heating Control. Units that provide ventilation air to multiple zones and operate in conjunction with zone heating and cooling systems shall not use heating or heat recovery to warm supply air above 60°F (16°C) when representative building loads or outdoor air temperature indicate that the majority of zones require cooling."

Interrupted or Insufficient Ventilation

Many types of zone-level HVAC equipment are equipped with variable-speed fan control. Ensuring that code-required outdoor airflow is delivered to the zone as the fan speed changes can be a challenge.

In some cases, the DOAS is designed to deliver conditioned outdoor air (CA) directly to each zone (*Figure 1*).[5,6] Since the outdoor air is not distributed through the zone-level HVAC equipment, the terminal unit can operate with variable-speed fan control (or even cycle the fan off), without impacting how much outdoor air is delivered to the zone.

In other cases, the DOAS is designed to deliver the conditioned OA to the inlet of each terminal unit. So if the terminal fan changes speed (or cycles off), less outdoor airflow is delivered to the zone.[6] One approach to address this challenge is to install pressure-independent, flow-measuring dampers in the DOAS ductwork (or in the terminal units) to ensure that the required outdoor airflow is delivered to each zone (*Figure 4*). As the various terminal fans (or the fan inside the dedicated OA unit) change speed, this damper modulates to maintain the desired quantity of outdoor air delivered. Adding these dampers also makes it possible to implement demand-controlled ventilation.

No (or Limited) Use of Exhaust-Air Energy Recovery

Because all the outdoor air for ventilation is brought in at a central location, a DOAS is typically a good candidate for exhaust-air energy recovery. Unfortunately, it is often a primary target for cost-cutting, and removed from many system designs.

Even when it does survive the cut, sometimes the design of the system limits its value. Consider the example depicted in *Figure 5*. In the left-hand image, the dedicated OA unit brings in 10,000 cfm (4710 L/s) of outdoor air at design conditions, with all of this air passing through the supply-side of the energy-recovery device. However, only 7,000 cfm (3300 L/s) of air is passing through the exhaust-side of the device. This is because 1,000 cfm (470 L/s) of air is exhausted locally (through restroom exhaust fans, for example) and another 2,000 cfm (940 L/s) leaks out through the envelope due to desired positive building pressurization. These unbalanced airflows (exhaust airflow/outdoor airflow = 0.70) result in less energy recovered.

If demand-controlled ventilation (DCV) is used, this unbalance worsens. In the right-hand image (*Figure 5*), the dedicated OA unit is only bringing in 6,000 cfm (2830 L/s), due to the reduced number of people in the building. But both the local exhaust airflow and leakage due to building pressurization remain unchanged, as they are not a function of population. This leaves only 3,000 cfm (1415 L/s) of air passing through the exhaust-side of the energy-recovery device, which results in even more unbalanced airflows (exhaust airflow/outdoor airflow = 0.50).

Centralizing all local exhaust airflows will result in more closely balanced airflows, maximizing energy savings. And, ASHRAE Standard 62.1-2016 allows the use of restroom exhaust (Class 2 air) for the purpose of exhaust-air energy recovery, as long as the Exhaust Air Transfer Ratio (EATR) does not exceed 10%. (EATR is one of the performance parameters included in AHRI Standard 1060.[7])

> **"5.16.3.2.5** Class 2 air shall not be recirculated or transferred to Class 1 spaces.
> **Exception:** When using any energy recovery device, recirculation from leakage, carryover, or transfer from the exhaust side of the energy recovery device is permitted. Recirculated Class 2 air shall not exceed 10% of the outdoor air intake flow."

Another solution is to use a dual-path dedicated OA unit, where the outdoor airflow passing through the supply-side of the energy-recovery device is equal to the airflow passing through the exhaust side of the device, and any excess outdoor air needed for pressurization is conditioned through a separate path.[8]

Standard 90.1 Compliance

Some in the industry are confused as to how certain requirements in Standard 90.1 apply to a DOAS.

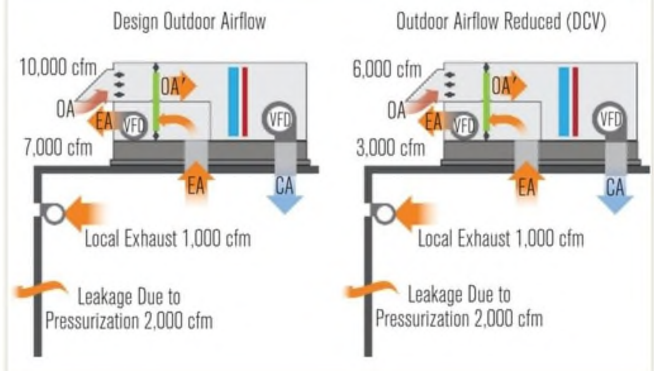
Minimum Equipment Efficiencies

Until AHRI Standard 920 was published in June 2013, there was no rating standard for DX dedicated OA equipment, so past editions of Standard 90.1 did not include minimum efficiency requirements for this class of equipment.[9]

Minimum efficiency requirements for DX dedicated OA equipment were added to Section 6.4.1.1 in the 2016 version of Standard 90.1.[10]

> "Dedicated outdoor air systems (DOAS) ... are used in many buildings covered by ASHRAE 90.1. However, the current ASHRAE 90.1 standard has no minimum energy efficiency requirements for this equipment. Through AHRI, manufacturers of DOAS developed Standard 920 to establish common rating conditions for these products. This proposal establishes for the first time a product class for DOAS."
>
> ~Excerpt from Addendum *CD* to Standard 90.1-2013

FIGURE 5 Unbalanced airflows for exhaust-air energy recovery.



These efficiencies are based on AHRI Standard 920, using the Integrated Seasonal Moisture Removal Efficiency (ISMRE) and Integrated Seasonal Coefficient of Performance (ISCOP).[9,10]

Fan Power Limit

As part of its prescriptive requirements, Section 6.5.3.1.1 of Standard 90.1-2016 defines a limit on the power used by any "fan system" that exceeds 5 hp (3.7 kW). The standard defines fan system power as "the sum of the (power) of all fans that are required to operate at fan system design conditions to supply air from the heating or cooling source to the conditioned spaces and return it to the source or exhaust it to the outdoors."

As explained in example 6-EEE in the *Standard 90.1-2016 User's Manual,* each zone-level terminal unit (such as a fan-coil, water-source heat pump, VRF terminal, sensible-cooling terminal unit, etc.) served by the DOAS is considered a separate "fan system" because each has a heating or cooling source. However, the "fan system" includes not only the fan inside the terminal unit, but also the fan(s) inside the dedicated OA unit. This example demonstrates how the fan power of the dedicated OA unit must be allocated to each terminal unit, on an airflow-weighted basis.[11]

In the *User's Manual* example referenced above, a wing of an elementary school building contains eight classrooms. Each classroom is served by a separate water-source heat pump, each equipped with a 0.75 hp (0.56 kW) fan motor. The dedicated OA system delivers 500 cfm (240 L/s) of conditioned outdoor air—for a total of 4,000 cfm (1920 L/s)—directly to each classroom. The dedicated OA unit is equipped with a 5 hp (3.7 kW) supply fan motor and a 1 hp (0.75 kW) exhaust fan motor.

As explained previously, in this configuration each WSHP is considered a separate fan system, and the power of the two fans in the dedicated OA unit must be allocated to each heat pump on an airflow-weighted basis. That is, for each classroom, 12.5% [500/4,000 cfm (240/1920 L/s)] of the dedicated OA unit fan power must be added to the fan power of the WSHP:

$$0.75 \text{ hp} + (0.125 \times 5 \text{ hp}) + (0.125 \times 1 \text{ hp}) = 1.5 \text{ hp}$$

$$[0.56 \text{ kW} + (0.125 \times 3.7 \text{ kW}) + (0.125 \times 0.75 \text{ kW}) = 1.1 \text{ kW})]$$

For this example, even with the dedicated OA unit fan power allocated to each WSHP, the total fan motor nameplate power for each "fan system" is 1.5 hp (1.1 kW), which is less than the 5 hp (4 kW) threshold. Therefore, this system does not need to comply with the maximum allowable fan power defined by Section 6.5.3.1.1.

Economizer

As part of its prescriptive requirements, Section 6.5.1 of Standard 90.1-2016 requires each cooling system to be equipped with either an air or fluid economizer. If the DOAS is sized to deliver only the minimum outdoor airflow required for ventilation, it is typically not capable of airside economizing.

Some systems can be designed to use a fluid economizer, while others may be exempt due to one of the many exceptions listed in the standard.

The most notable exception related to DOAS (exception A) exempts individual fan cooling units smaller than 4.5 tons (16 kW). The *Standard 90.1-2016 User's Manual* clarifies that "the requirement is based on the fan-coil unit and not the capacity of a central chilled-water plant or VRF system condensing unit capacity."[11]

Ventilation Air Heating Control

As mentioned previously, a new prescriptive requirement (Section 6.5.2.6) in Standard 90.1-2016 prohibits reheating (or heating) ventilation air above 60°F (16°C) whenever most zones require cooling, even if using recovered energy for the reheat.

Summary

- Dehumidify the OA to a dew point that is drier than the zone.
- Deliver the conditioned OA at a cold (not "neutral") temperature, whenever possible.
- Implement demand-controlled ventilation and use recovered energy to reheat the dehumidified air in the dedicated OA unit, but only when necessary to prevent over-cooling at low-load conditions.
- Deliver conditioned OA directly to each zone or use flow-measuring dampers to ensure proper ventilation as operating conditions change.
- Centralize exhaust to better balance airflows and maximize exhaust-air energy recovery.
- Use Moisture Removal Capacity (MRC) and ISMRE, tested in accordance with AHRI Standard 920, when specifying the required dehumidification performance of a DX dedicated OA unit.

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