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Impact of DOAS Supply-Air Dew Point Temperature on Space Humidity

Using a dedicated outdoor-air system (DOAS) to condition outdoor air separately from recirculated air can make it easier to verify that sufficient ventilation airflow reaches each zone. And when that outdoor air is dehumidified, so that it is drier than the space, it can also help prevent high space humidity levels.

But many of the dedicated outdoor-air systems designed and installed today are not dehumidifying adequately. This *Engineers Newsletter* examines one reason why this may be the case.

Introduction

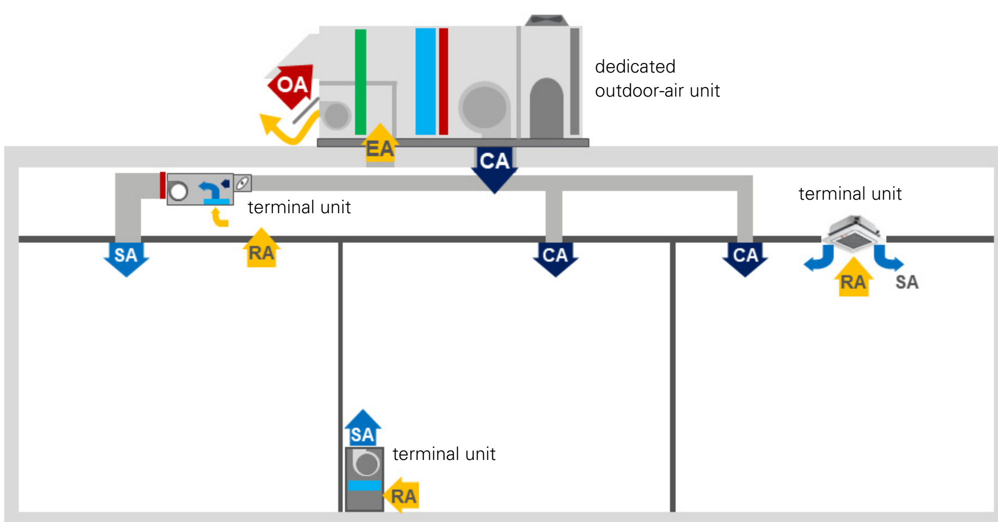
In buildings that use zone-level terminal units for cooling and heating (such as VRF, water-source heat pumps, fan-coils, small DX split systems, PTACs, chilled beams, or sensible-cooling terminal units), a separate dedicated outdoor-air system is often used to provide ventilation.

As the name implies, this system uses a dedicated unit to condition all the outdoor air (OA) being brought in for ventilation (Figure 1). Meanwhile, a terminal unit provides cooling or heating in each zone.

Isn't 55°F Dew Point Good Enough?

For many HVAC design engineers, a common design practice has been to specify the dedicated OA unit to dehumidify the outdoor air to a 55°F dew point temperature (which equates to 64.6 gr/lb at sea level) and then reheat it to a "neutral" dry-bulb temperature (70°F for example).

Figure 1. Dedicated outdoor-air system (DOAS).



However, depending on the desired conditions in the space, this conditioned outdoor air (labeled "CA" in the figures throughout) may actually be wetter than the space. For example, if the desired space conditions are 73°F dry bulb and 50 percent relative humidity (RH), this equates to a humidity ratio of 60.6 gr/lb.

Therefore, the humidity ratio of the conditioned outdoor air (CA) is higher than the desired space humidity ratio, meaning that the DOAS is **adding** latent load to the space. This requires the zone-level terminal unit to have sufficient dehumidification (latent) capacity to remove this added latent load, and the latent load generated in the space (due to people, infiltration, etc.). Otherwise, the space humidity level will rise higher than desired.

To demonstrate this impact, consider the example K-12 classroom detailed in Table 1.

The dedicated OA unit delivers 350 cfm of outdoor air (CA) directly to this classroom at 55°F dew point and 70°F dry bulb (Figure 3). The terminal unit then conditions only recirculated air (RA) to cool or heat the space to the desired dry-bulb temperature.

Because the conditioned outdoor air is slightly cooler than the space, it offsets 1140 Btu/hr of the space sensible cooling load: 1.085 × 350 cfm × (73°F – 70°F). At design cooling load conditions (Table 1), this leaves the remaining 19,160 Btu/hr (20,300 – 1140) of sensible load to be offset by the terminal unit.

Assuming this terminal unit is sized to cool recirculated air to 55°F dry bulb at design conditions, the terminal unit serving this classroom is sized for 980 cfm: 19,160 Btu/hr = 1.085 × V_{sa} × (73°F – 55°F).

Figure 2. Dehumidifying to 55°F dew point may actually *add* latent load to the space.

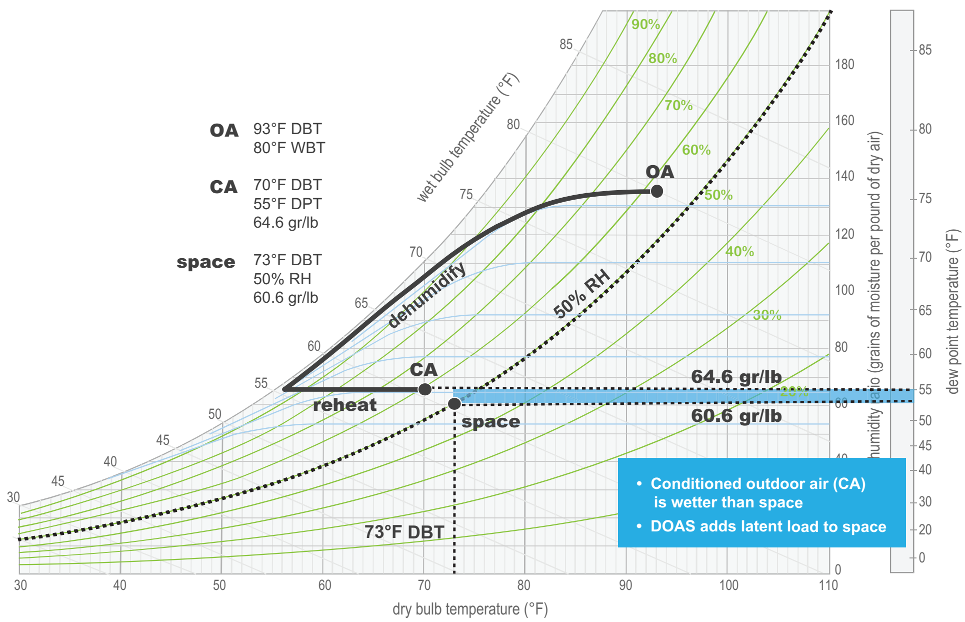


Table 1. Example of K-12 classroom.

	design load	part load
floor area (A_z)	750 ft²	
design population (P_z)	26 people	
space dry-bulb temperature (DBT_{space})	73°F	
space relative humidity desired (RH_{space})	50%	
space sensible cooling load ($Q_{space, sensible}$)[1]	20,300 Btu/hr	12,200 Btu/hr
space latent load ($Q_{space, latent}$)[2]	4030 Btu/hr	4030 Btu/hr
space sensible heat ratio (SHR)[3]	0.83	0.75
zone outdoor airflow (V_{oz})[4]	350 cfm	350 cfm

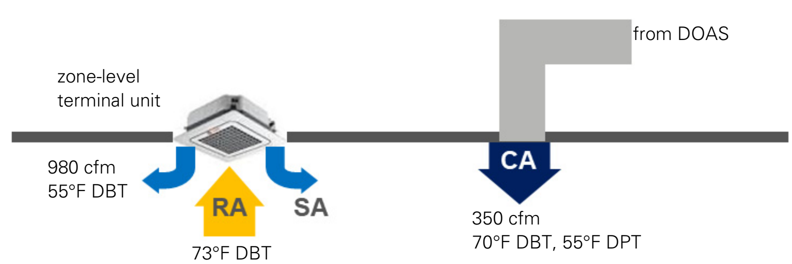
[1] Determined using load calculation software

[2] 26 people × 155 Btu/hr/person (per 2017 ASHRAE® Handbook, page 18.4, assuming "seated, very light work" for the occupant activity level)

[3] Space SHR = $Q_{space,sensible}$ / ($Q_{space,sensible}$ + $Q_{space,latent}$)

[4] V_{bz} = 350 cfm = (26 people x 10 cfm/person) + (750 ft² x 0.12 cfm/ft²), per ASHRAE® Standard 62.1-2016, Table 6.2.2.1. V_{oz} = V_{bz} / E_z = 350 cfm / 1.0 (per ASHRAE® Standard 62.1-2016, Table 6.2.2.2, assuming conditioned OA delivered directly to the space at a temperature cooler than the space)

Figure 3. DOAS and terminal unit serving this example classroom.



Depicted on a psychrometric chart (Figure 4), the dedicated OA unit dehumidifies the warm, humid outdoor air (OA) to a 55°F dew point and then reheats it to 70°F dry bulb (CA). The terminal unit cools 980 cfm of recirculated air (RA) from 73°F to 55°F dry bulb (SA). The classroom receives a total of 1330 cfm (350 cfm of conditioned outdoor air from the DOAS plus 980 cfm of cooled supply air from the terminal unit), depicted as the combined condition labeled "SA+CA" in Figure 4.

Following the 0.83 space SHR line from point SA+CA, the resulting condition in the space (RA) is 73°F dry bulb and 55 percent RH; **higher** than the 50 percent RH desired.

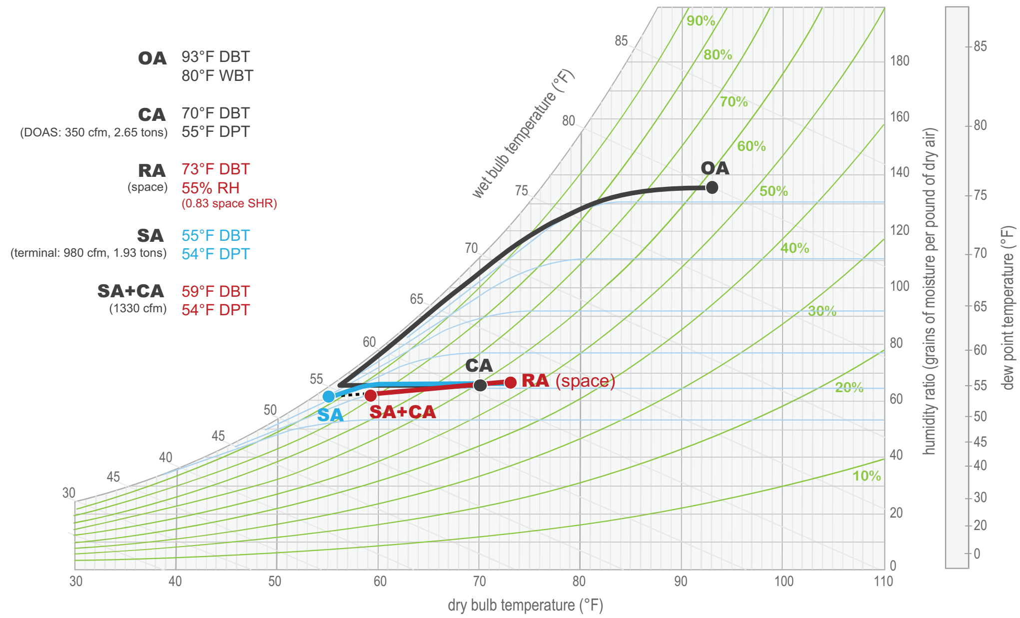
Why is this? The dedicated OA unit dehumidifies the incoming outdoor air to a 55°F dew point, which is at a higher humidity ratio than the desired space humidity ratio. Therefore, the DOAS still adds some latent load to the space, and the terminal unit does not have sufficient dehumidification (latent) capacity to remove this added latent load plus the space latent load.

What happens at part-load conditions? For this example classroom, assume that the space sensible cooling load is only 60 percent of the design load, or 12,200 Btu/hr (Table 1). However, all the students are still present in the classroom, so the space latent load remains unchanged (4030 Btu/hr). This causes a reduction in the space SHR, from 0.83 at design to 0.75 at this part-load condition.

The DOAS continues to deliver 350 cfm of conditioned outdoor air to the classroom at the same conditions, so it still offsets 1140 Btu/hr of the space sensible cooling load. That leaves the remaining 11,060 Btu/hr (12,200 – 1140) of sensible load to be offset by the terminal unit.

Because the sensible load in the space is lower, the terminal unit controller has reduced its fan speed in this example, delivering its minimum airflow of 740 cfm (980 cfm × 0.75, assuming 75 percent minimum fan speed). To offset the remaining 11,060 Btu/h of sensible load in the space, the terminal unit needs to cool this 740 cfm of recirculated air to 59°F dry bulb: 11,060 Btu/hr = 1.085 × 740 cfm × (73°F – DBT_{sa}).

Figure 4. DOAS supplying 55°F dew point at design (full-load) conditions.

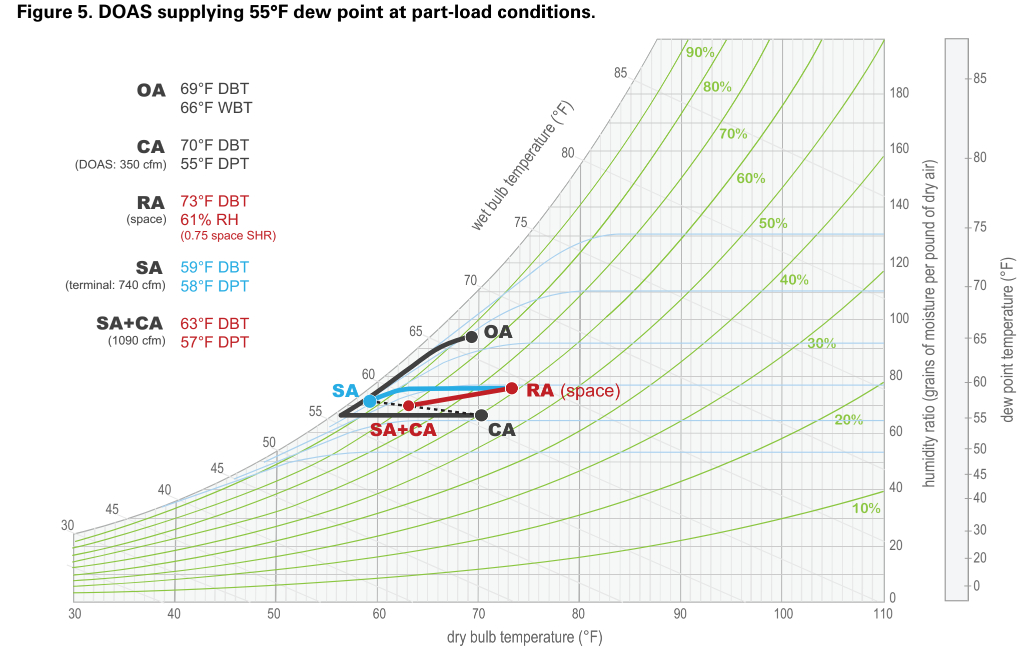


At this part-load condition, the outdoor air (OA) is cooler, but still humid (Figure 5). The dedicated OA unit still dehumidifies it to the same 55°F dew point before reheating it to 70°F dry bulb (CA).

As described, the terminal unit now cools 740 cfm of recirculated air (RA) from 73°F to 59°F dry bulb (SA), so the classroom receives a total of 1090 cfm (SA+CA). Following the part-load 0.75 space SHR line, the resulting condition in the space is 73°F dry bulb and 61 percent RH; **much higher** than the 50 percent RH desired.

For this example classroom, arbitrarily specifying the dedicated OA unit to dehumidify the outdoor air to a 55°F dew point did not result in the desired space humidity level, especially at part load.

Figure 5. DOAS supplying 55°F dew point at part-load conditions.



What if instead of 55°F dew point, the dedicated OA unit is selected for a lower dew point? As depicted in Figure 6, at design (full-load) conditions, the dedicated OA unit now dehumidifies the warm, humid outdoor air (OA) to 45°F dew point, but still reheats it to 70°F dry bulb (CA). Again, the terminal unit cools 980 cfm of recirculated air (RA) from 73°F to 55°F dry bulb (SA).

Because the conditioned outdoor air (CA) is drier, the combined 1330 cfm of air supplied to the space (SA+CA) is drier, and the resulting condition in the space is 73°F dry bulb at the desired 50 percent RH.

At the same example part-load condition (Figure 7), the outdoor air (OA) is still dehumidified to 45°F dew point and reheated to 70°F dry bulb (CA). The terminal unit, now operating at minimum fan speed, cools 740 cfm of recirculated air (RA) from 73°F to 59°F dry bulb (SA). The resulting condition in the space is 73°F dry bulb and 51 percent RH.

As mentioned, arbitrarily specifying the dedicated OA unit to deliver 55°F dew point may not result in the space humidity level desired, especially at part load. A lower dew point (45°F in this example) may be needed to achieve the desired results.

Determining the Required DOAS Supply-Air Dew Point

The "right" DOAS supply-air dew point for a given application depends on the type of space and the indoor humidity level desired, as well as the type of terminal units used (see sensible-only terminal units sidebar).

Figure 6. DOAS supplying 45°F dew point at design (full-load) conditions.

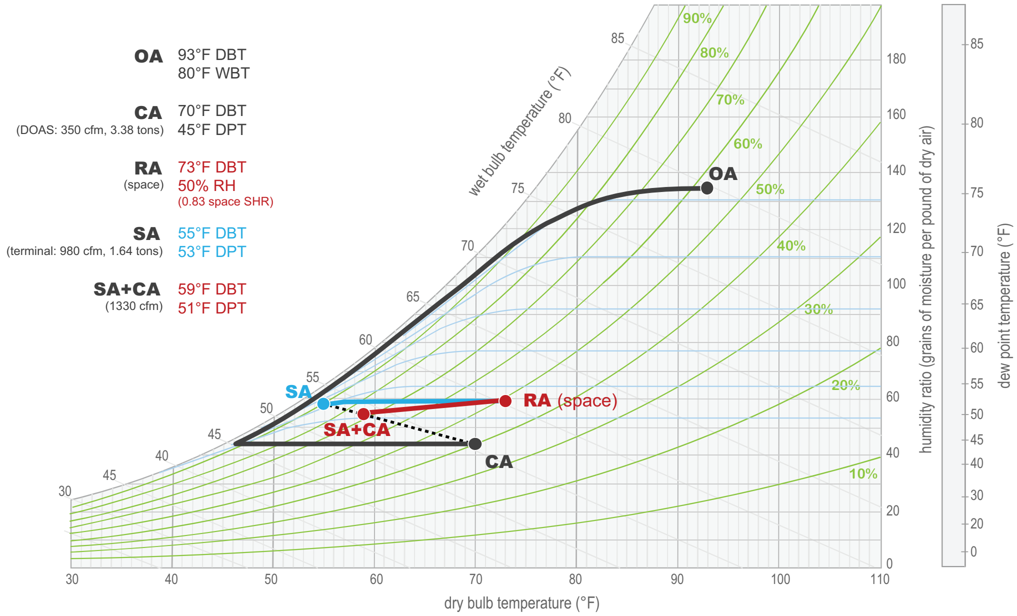
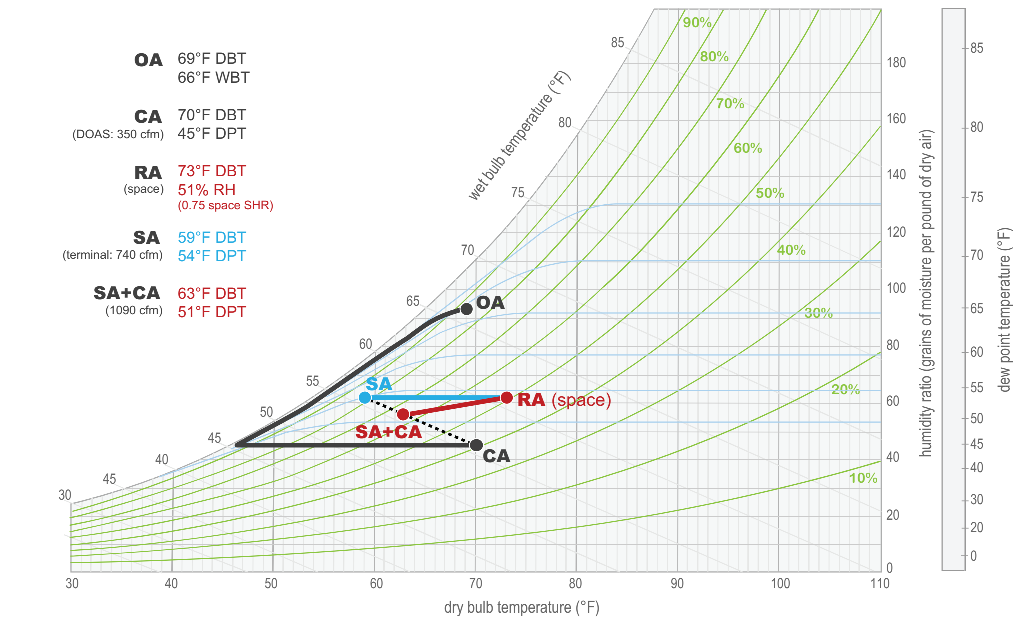


Figure 7. DOAS supplying 45°F dew point at part-load conditions.



Sensible-only terminal units

Some types of terminal units (such as chilled beams, radiant cooling panels, or sensible-cooling terminals) are designed to provide only sensible cooling, no dehumidification. When this type of terminal equipment is used, the DOAS is the **only** source of dehumidification and must be sized to dehumidify the OA dry enough so that it maintains the space dew point low enough to avoid condensation on the sensible-only terminals. For example, if 57°F water is supplied to the terminals, the DOAS might be designed to prevent the space dew point from rising above 55°F, creating a 2°F buffer.