providing insights for today's hvac system designer

Engineers Newsletter

volume 49–1

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

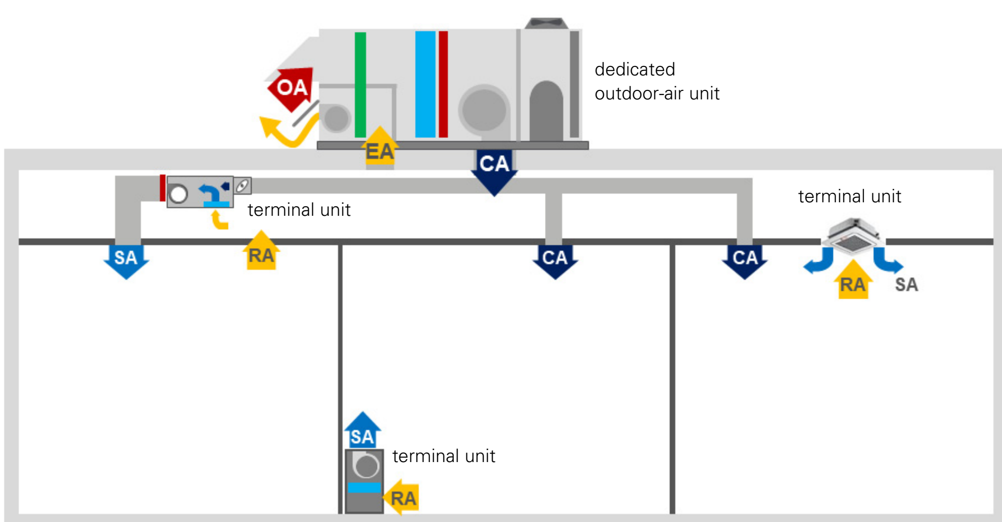
In buildings that use zone-level terminal units for cooling and heating (such as VRF, water-source heat pumps, fan-coils, small DX split systems, PTACs, chilled beams, or sensible-cooling terminal units), a separate dedicated outdoor-air system is often used to provide ventilation.

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.

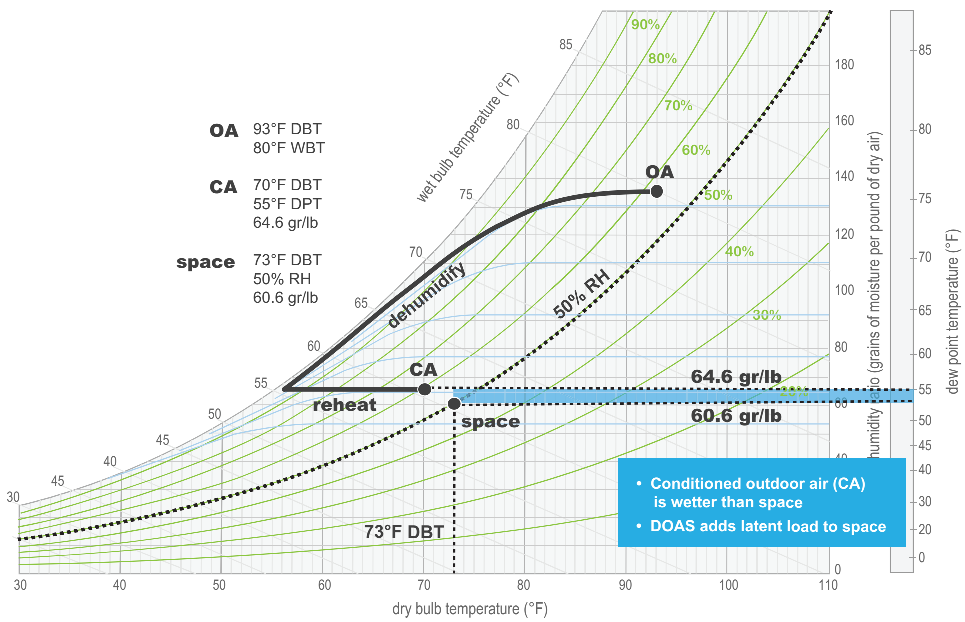


Table 1. Example of K-12 classroom.

	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

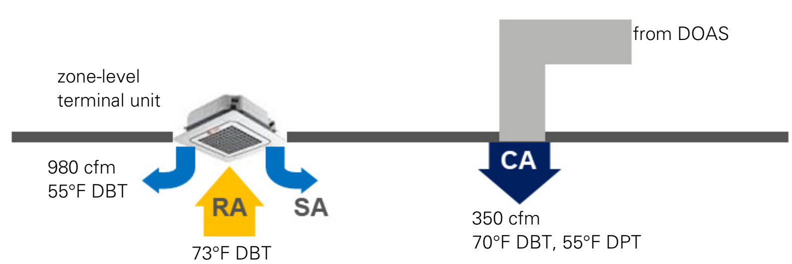
[1] Determined using load calculation software

[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}).

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

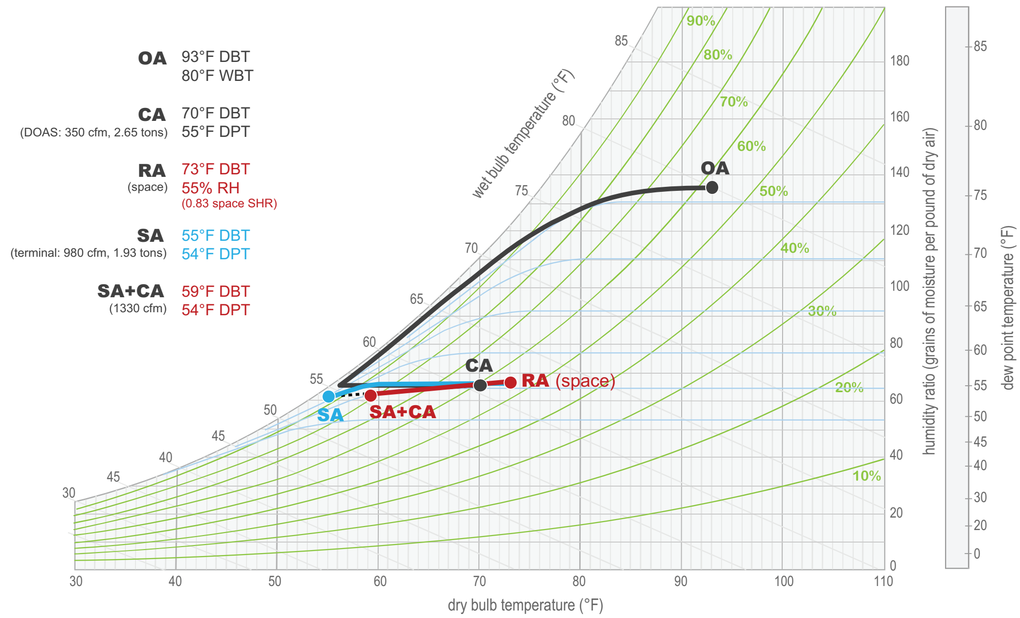
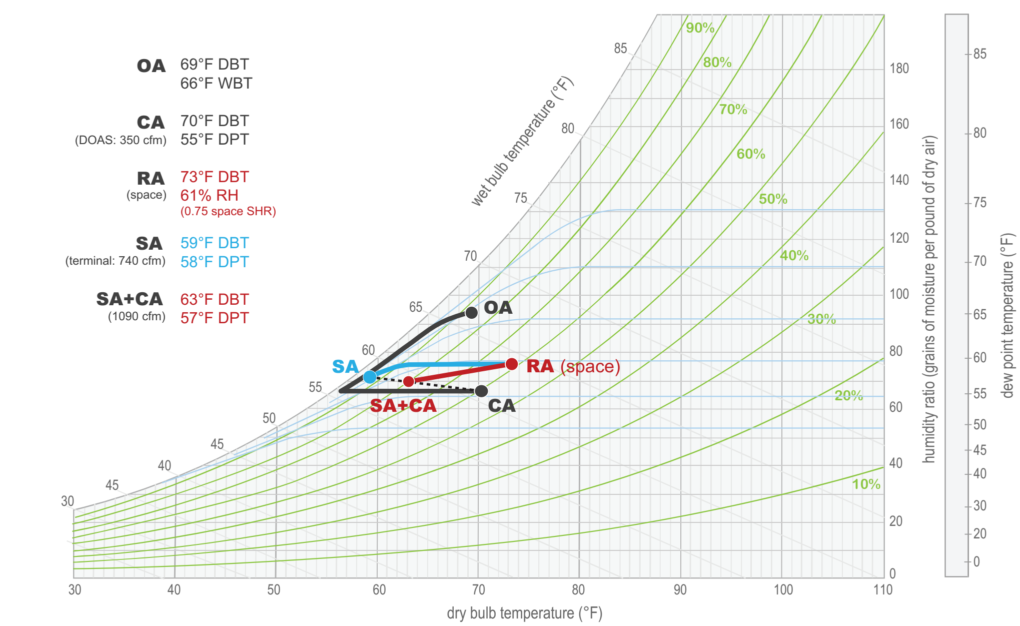


Figure 5. DOAS supplying 55°F dew point at part-load conditions.



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.

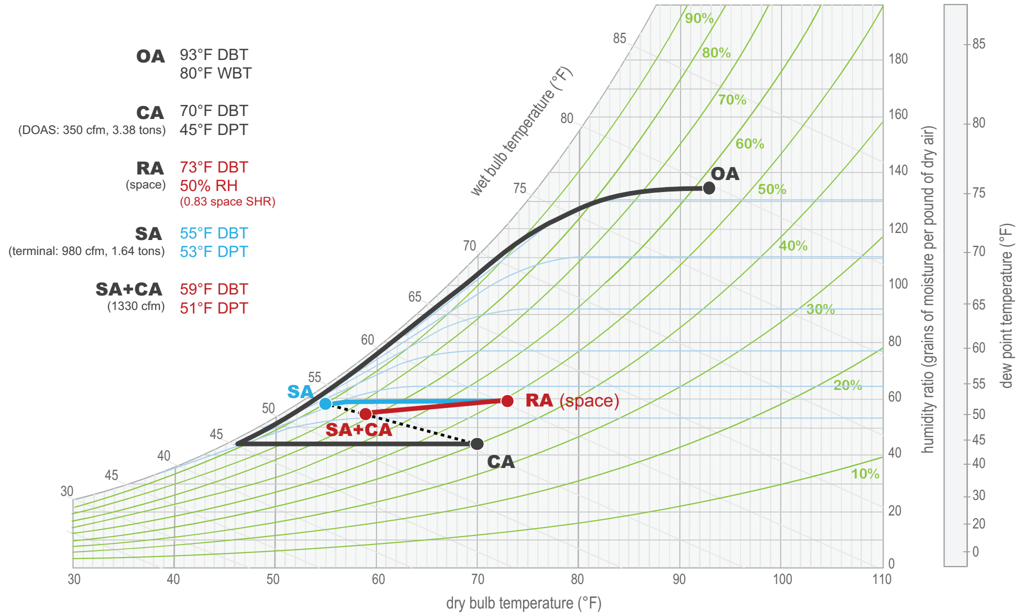
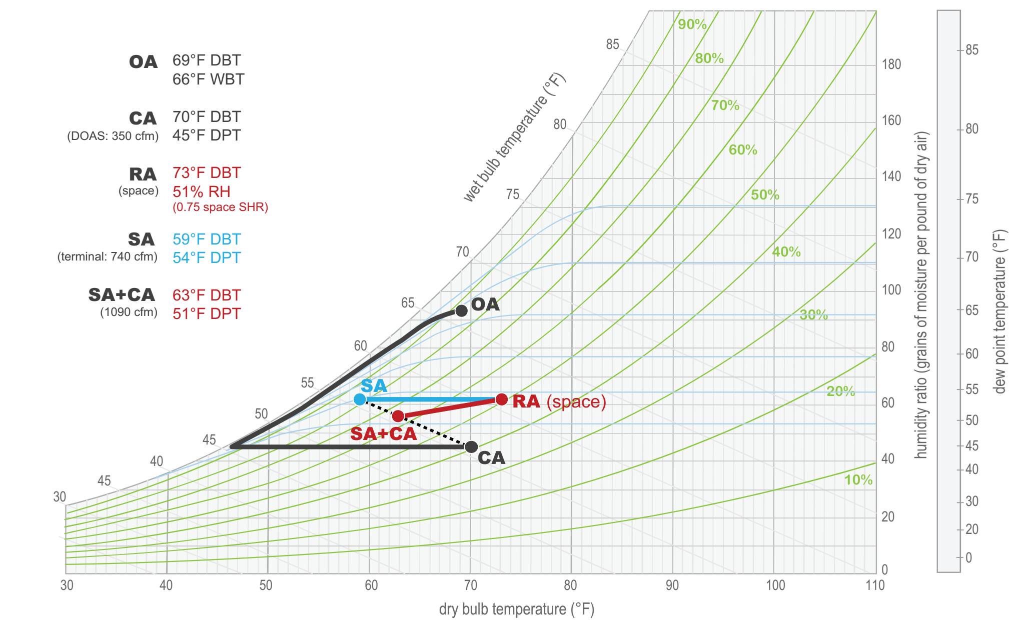


Figure 7. DOAS supplying 45°F dew point at part-load conditions.



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.

The following latent load equation can be used to determine the required humidity ratio (W_{ca}) of the conditioned outdoor air supplied by the DOAS:

$$Q_{space,latent} = 0.69 \times V_{oz} \times (W_{space} - W_{ca})$$

where:

- $Q_{space,latent}$ = latent load in the space, Btu/hr
- V_{oz} = outdoor airflow delivered to the space, cfm
- W_{space} = desired humidity ratio in the space, gr/lb
- W_{ca} = required humidity ratio of the conditioned outdoor air supplied by the DOAS, gr/lb

The value 0.69 in this equation 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.

For the example K-12 classroom described earlier, the space latent load is estimated to be 4030 Btu/hr and the required outdoor airflow is 350 cfm (see Table 1). If the desired space conditions are 73°F dry bulb and 50 percent RH, this equates to a humidity ratio of 60.6 gr/lb in the space. Using the previous equation, in order to remove the entire space latent load, this 350 cfm of outdoor air must be dehumidified to 43.9 gr/lb, which equates to approximately 45°F dew point.

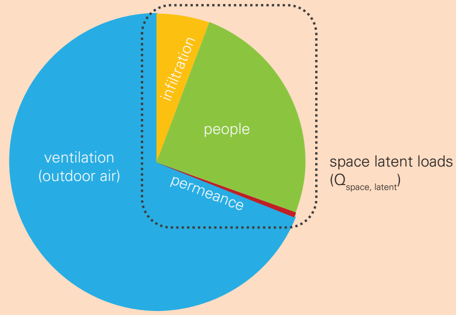
$$4030 \text{ Btu/hr} = 0.69 \times 350 \text{ cfm} \times (60.6 \text{ gr/lb} - W_{ca}), \text{ so } W_{ca} = 43.9 \text{ gr/lb}$$

Space Latent Load ($Q_{space,latent}$)

The pie chart in Figure 8 depicts the sources of latent load for an example K-12 classroom. In this case, the ventilation (outdoor) air accounts for almost two-thirds of the total latent load. The remaining one-third are the space latent loads, which occur within the boundaries of the conditioned space (mostly due to people and infiltration).

The latent load due to the outdoor air affects the dehumidification capacity of the equipment, but only the space portion of the latent load ($Q_{space,latent}$) dictates how dry the supply air must be.These loads will vary for different space types, based on activity level and occupant density.

Figure 8. Latent loads for example K-12 classroom.



Source: *Humidity Control Design Guide*, ASHRAE© 2001, p. 278

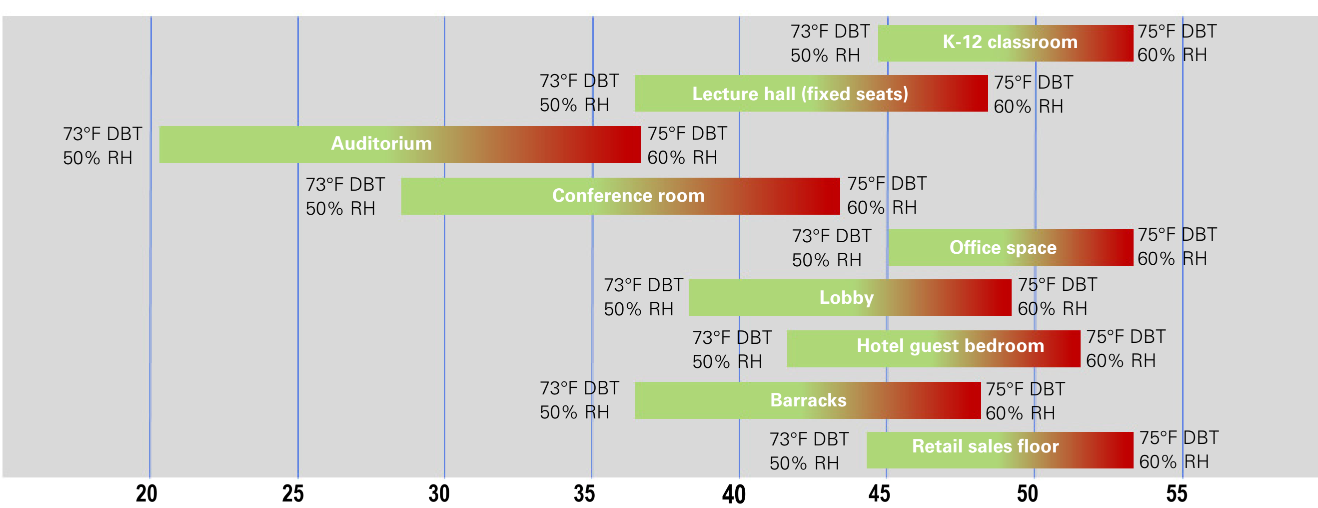
This is dry enough to offset the entire space latent load and keep the space humidity ratio (W_{space}) at the desired level.

The required DOAS supply-air dew point varies for other space types and other space conditions. Figure 9 compares this required dew point for several different space types, assuming minimum ventilation rates and default occupant densities per ASHRAE Standard 62.1.

For the example K-12 classroom (top bar in the chart), if the desired space conditions are 73°F dry bulb and 50 percent RH, the conditioned outdoor air needs to be dehumidified to approximately a 45°F dew point (as calculated previously). But if a higher dew point is supplied by the DOAS, space humidity will be higher.

For spaces with lower ventilation rates, such as an auditorium or conference room, a very low dew point may be required; or the DOAS might be designed to deliver more-than-minimum ventilation airflow to sufficiently dehumidify these spaces.

Figure 9. Comparison of DOAS supply-air dew points for various space types and conditions.



Required dew point temperature of conditioned OA to maintain desired space humidity level, °F

(This assumes default occupant density, 155 or 200 Btu/h/person space latent load, and sea level elevation.)

Neutral versus cold air

Some HVAC engineers design the dedicated outdoor-air system to reheat the dehumidified outdoor air to a "neutral" dry-bulb temperature (70°F for example). But this wastes energy when the zones also require cooling.

When a chilled-water or refrigerant coil is used to dehumidify the outdoor air (OA to CC in Figure 10), a byproduct of that dehumidification process is sensible cooling. The dry-bulb temperature of the air leaving the cooling coil (CC) is colder than the space. If this dehumidified air is then reheated to neutral (CC to CA), some of the sensible cooling performed by the dedicated OA unit is wasted.

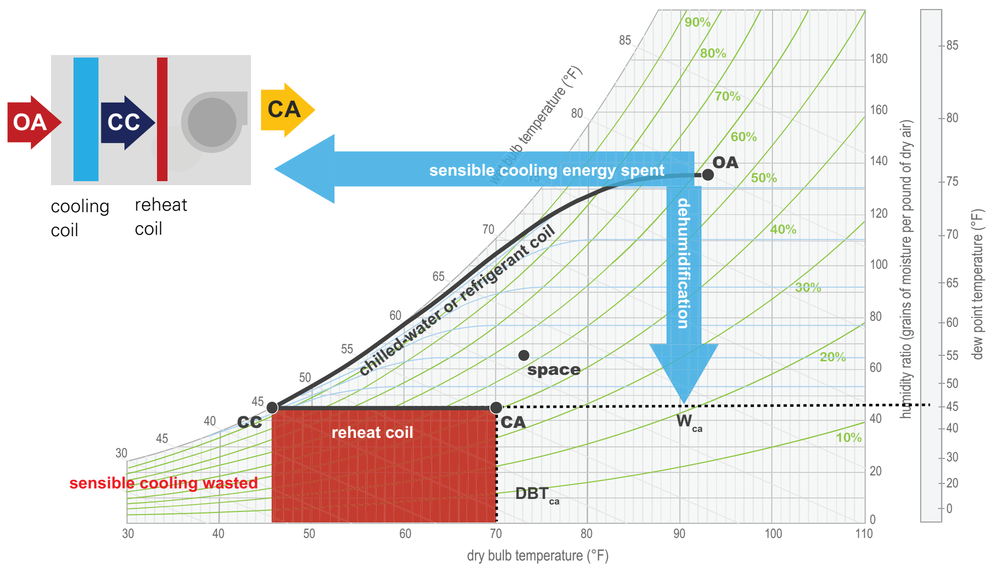
If the zones need cooling, why not make use of the sensible cooling already done by the dedicated OA unit? Otherwise the terminal units will be forced to offset more of the space sensible cooling load, increasing their energy use.

To address the inefficiency of neutral-temperature air, the 2016 version of ASHRAE Standard 90.1 added a new requirement that prohibits a DOAS from reheating the dehumidified outdoor air to any warmer than 60°F dry bulb, whenever the majority of zones require cooling. This applies even if recovered heat is used.

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 when representative building loads or outdoor air temperature indicate that the majority of zones require cooling.

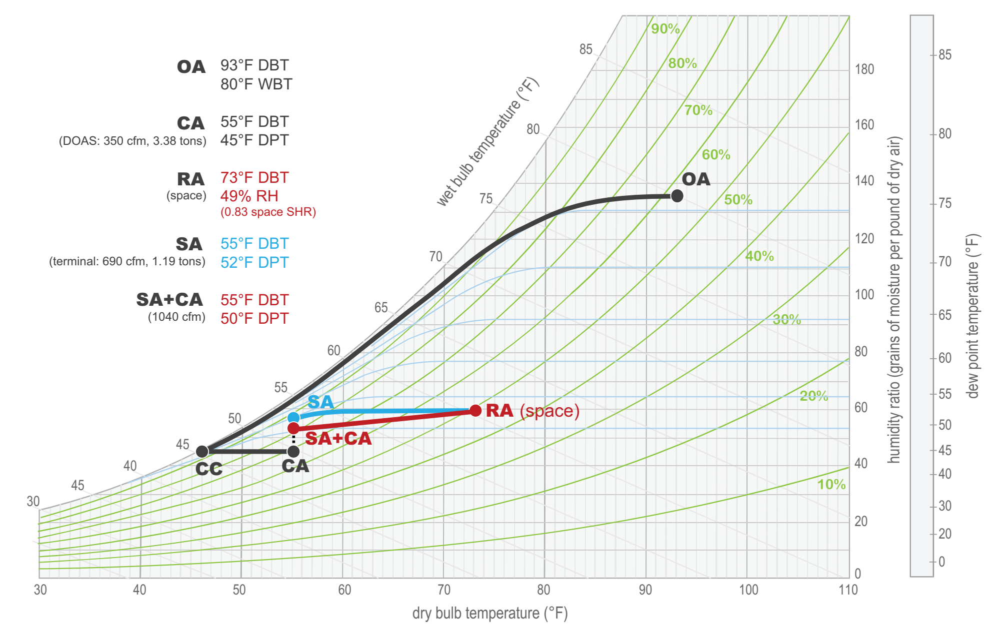
ASHRAE 90.1-2016

Figure 10. Sensible cooling energy wasted when dehumidified outdoor air is reheated to "neutral".



However, there may be times when reheating is needed to avoid over-cooling a space. Or, depending on what dew point is required, some engineers might not be comfortable with delivering the air that cold. In Figure 11, the outdoor air is dehumidified to 45°F dew point, which corresponds to a dry-bulb temperature of about 45°F (CC). Rather than reheating this air to neutral, consider reheating it just enough (CC to CA) to avoid delivering colder-than-desired air to the spaces (to 55°F in this example). This takes advantage of the sensible cooling effect of this cold, dehumidified air, thereby reducing the need for re-cooling at the zone-level terminal units.

Figure 11. DOAS supplying 45°F dew point and 55°F dry bulb at design (full-load) conditions.



Final Thoughts

In most climates, the dedicated OA unit should be sized to dehumidify the outdoor air to a dew point that is low enough to offset the space latent load and maintain space humidity level at or below the desired limit. Arbitrarily specifying a unit to deliver 55°F dew point may not result in the space humidity level desired, especially at part load. A lower dew point is often needed to achieve the desired results.

The "right" 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.

Note that a dedicated OA unit sized to dehumidify to a lower dew point will require more capacity than one sized to dehumidify to a higher dew point (Table 2). For the example classroom described in this EN, dehumidifying 350 cfm from design outdoor air conditions to 55°F dew point requires 2.65 tons of capacity (Figure 4), whereas dehumidifying this air to 45°F dew point requires 3.38 tons (Figure 6).

However, when the space is drier, the enthalpy of the recirculated air entering the terminal unit is lower, reducing its required cooling capacity. When the DOAS delivers 55°F dew point air, the terminal unit requires 1.93 tons to cool 980 cfm of recirculated air to 55°F dry bulb (Figure 4). But when the DOAS delivers 45°F dew point air, the terminal unit requires only 1.64 tons of capacity (Figure 6).

Finally, if the DOAS is designed to dehumidify the OA to 45°F dew point, but then reheat it to only 55°F dry bulb, this cool air often allows the terminal unit to be downsized even further. In this example, the design airflow of the terminal unit is reduced to 690 cfm and its cooling capacity is reduced to 1.19 tons (Figure 11).

The impact on the cost of the overall system depend on the relative installation costs of the DOAS versus the terminal system.

By John Murphy, Trane. To subscribe or view previous issues of the *Engineers Newsletter* visit trane.com/EN. Send comments to ENL@trane.com.

Table 2. Impact of DOAS supply-air conditions (example K-12 classroom).

	DOAS supply-air conditions		
	70°F DBT 55°F DPT (Figure 4)	**70°F DBT 45°F DPT (Figure 6)**	**55°F DBT 45°F DPT (Figure 11)**
dedicated OA unit capacity	2.65 tons	3.38 tons	3.38 tons
terminal unit capacity	1.93 tons	1.64 tons	1.19 tons
terminal unit design airflow	980 cfm	980 cfm	690 cfm

Resources

[1] ANSI/ASHRAE, Standard 62.1-2016, *Ventilation for Acceptable Indoor Air Quality*. Atlanta: ASHRAE. 2016.

[2] ANSI/ASHRAE/IES, Standard 90.1-2016, *Energy Standard for Buildings Except Low-Rise Residential Buildings*. Atlanta: ASHRAE. 2016.

[3] ASHRAE, Inc. *Design Guide for Dedicated Outdoor Air Systems*. Atlanta: ASHRAE, 2017.

[4] ASHRAE, Inc. *ASHRAE Handbook – Fundamentals*. Atlanta: ASHRAE, 2017.

[5] Murphy, J. 2018. "Common Pitfalls in the Design and Operation of DOAS." *ASHRAE Journal* (September): 10-17.

[6] Trane®. *CoolSense® Integrated Outdoor Air* system catalog. SYS-APG004*-EN. 2019.

[7] Trane. *Dedicated Outdoor Air Systems* application guide. SYS-APG001*-EN. 2019.

[8] Trane. *Variable Refrigerant Flow Systems* system catalog. SYS-APG007*-EN. 2020.

[9] Trane. "Horizon® Dedicated Outdoor Air Systems." https://www.trane.com/Horizon.

Join your local Trane office for the 2020 Engineers Newsletter LIVE!

Mark your calendar!

Impact of DOAS Dew Point on Space Humidity. Dedicated outdoor-air systems (DOAS) are used in a variety of building types to provide ventilation; and when the outdoor air is dehumidified, a DOAS can help prevent high space humidity levels. But many of the systems designed and installed today are not dehumidifying adequately. This ENL will demonstrate how space humidity levels are affected by the DOAS discharge-air conditions, at both full load and part load.

Indoor Agriculture: HVAC System Design Considerations. Conditioning spaces for plants instead of humans introduces new challenges. This ENL will discuss plants, the dehumidification challenges they pose, and how precision cooling for indoor agriculture is different compared to comfort cooling. We'll also discuss common air- and waterside system configurations used to maintain space conditions to ensure a healthy crop.

Applying VRF for a Complete Building Solution. This ENL builds upon the 2014 VRF program "Applying Variable Refrigerant Flow" with detailed discussions on several considerations. Topics will include: when to use heat recovery instead of heat pump configurations, how to scale VRF systems to include other building systems, ventilation delivery, humidity management and more.

Decarbonize HVAC Systems. Many municipalities are taking action to reduce their carbon emissions which includes the reduction, or removal, of natural gas for heating. The HVAC industry will face the challenge of heating buildings with electric heat. This ENL will cover the motivation to electrify, areas currently effected by this trend, and potential systems to meet electrification needs.

Contact your local Trane office for dates and details.

For more information, contact your local Trane office or e-mail us at comfort@trane.com

This newsletter is for informational purposes only and does not constitute legal advice. Trane believes the facts and suggestions presented here to be accurate. However, final design and application decisions are your responsibility. Trane disclaims any responsibility for actions taken on the material presented.