Selecting DOAS Equipment with Reserve Capacity

By John Murphy, Member ASHRAE

Safety factors are often used by engineers when designing various types of HVAC systems. However, excessive use of safety factors can result in larger-than-necessary equipment, inflated installed costs, and sometimes excessive energy use. This is especially true when safety factors are used during several steps of the design process, which compounds their impact along the way.

However, many of the decisions made during the design process are based on incomplete information, assumptions that may turn out to be invalid, or valid assumptions that may no longer be valid one, five, or 10 years after the system or equipment is installed. Therefore, safety factors are important tools for design engineers allowing the HVAC system to accommodate unexpected loads and the need for increased airflow or dehumidification capacity.

This article discusses a slightly different approach to use when designing dedicated outdoor air systems (DOAS). The recommendation of this author is to avoid applying safety factors when calculating design outdoor airflow and design dehumidification capacity, and instead select the dedicated outdoor air (OA) equipment with reserve capacity.

Dedicated Outdoor Air Systems

A dedicated outdoor air system uses a separate piece of equipment to condition (filter, heat, cool, humidify, dehumidify) all of the outdoor air brought into a building for ventilation. This conditioned outdoor air is then delivered either directly to each occupied space or to local HVAC equipment serving those spaces. Meanwhile, the local equipment (such as fan-coils, water-source heat pumps, packaged terminal air conditioners (PTACs), small packaged units, VAV terminals, chilled ceiling panels, or chilled beams) located in or near each space provide cooling and/or heating to maintain space temperature.[1,2]

Treating the outdoor air separately can make it easier to verify that sufficient ventilation airflow reaches each occupied space and can help avoid high indoor humidity levels. The latter is accomplished by dehumidifying the outdoor air to remove the entire ventilation latent load and most (or all) of the space latent loads, leaving the local HVAC equipment to primarily handle space sensible cooling

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loads. Some types of local equipment, such as chilled ceiling panels or chilled beams, must operate dry and avoid condensation, which limits their ability to handling sensible loads only.

Figure 1 shows several example DOAS configurations. Some deliver the conditioned outdoor air (CA) directly to each zone,[3] while other configurations deliver the air to the intakes of local, single-zone equipment (such as fan-coils, water-source heat pumps, dual-duct VAV terminals, small packaged rooftop units, or single-zone air handlers) or to centralized, multiple-zone equipment (such as floor-by-floor VAV air handlers or self-contained units).

In addition, many types of dedicated outdoor air equipment are available (*Figure 2*). Dehumidification is usually provided by direct-expansion (DX) refrigeration, a chilled-water coil, a desiccant-based dehumidification device, or some combination of these technologies. Often, the dedicated outdoor air unit includes an exhaust-air energy recovery device (such as a total-energy wheel, fixed-plate heat exchanger, coil runaround loop, or heat pipe), which can reduce energy use and allow for downsizing of the cooling and heating equipment. In fact, ASHRAE Standard 90.1 requires the use of an exhaust-air energy recovery device for many DOAS applications.[4]

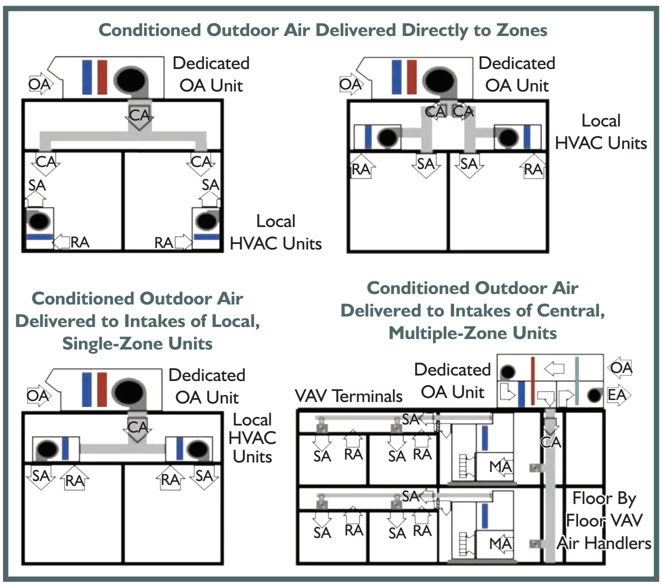

Figure 1: Example dedicated outdoor air system configurations.

Determining Design Airflow

In most applications, the design airflow for a dedicated outdoor air unit is dictated by the amount of ventilation air required by industry standard or local code.[5] In some cases, the owner or design team may choose to deliver more than code-minimum ventilation airflow to improve indoor air quality or to earn the "Increased Ventilation" credit when certifying a project using LEED. Finally, in applications with very low ventilation requirements or very high indoor latent loads, the design engineer may choose to increase the airflow delivered by the dedicated outdoor air unit so that the conditioned outdoor air can be delivered at a higher dew point (not as dry).

Table 6-1 of ASHRAE Standard 62.1-2007[6] prescribes two ventilation rates for each occupancy category: one for people-related sources of contaminants (R_p) and another for building-related sources (R_a). Equation 6-1 from Standard 62.1 is used to determine the minimum outdoor airflow (V_{bz}) that must be delivered to each breathing zone:

$$V_{bz} = R_p \times P_z + R_a \times A_z \qquad (1)$$

where

V_{bz} = outdoor airflow required in the breathing zone of the occupiable space, cfm (L/s)

R_p = outdoor airflow rate required per person, cfm/person (L/s·person)

P_z = largest number of people expected to occupy the zone during typical usage

R_a = outdoor airflow rate required per unit area, cfm/ft^2 (L/s·m^2)

A_z = occupiable floor area of the zone, ft^2 (m^2)

Next, Equation 6-2 and Table 6-2 from Standard 62.1 are used to account for zone air-distribution effectiveness (E_z), and to calculate the design outdoor airflow for the zone (V_{oz}). This is the outdoor airflow that must be provided to the zone by the air distribution system (that is, through the supply-air diffusers).

Finally, for a 100% outdoor air system in which one air handler supplies only outdoor air to one or more zones, Equation 6-4 from Standard 62.1 is used to calculate the system-level outdoor air intake flow (V_{ot}), by summing the zone outdoor airflows of all zones served by the dedicated outdoor air unit:

$$V_{ot} = \Sigma_{all\ zones}\ V_{oz} \qquad (2)$$

In some system configurations, the dedicated outdoor air unit provides conditioned OA to the intakes of local or centralized HVAC equipment, rather than directly to each zone. In these configurations, the dedicated OA unit must be sized to deliver the sum of the outdoor air intake flows (V_{ot}) required by each of the systems being served.

Makeup Air Applications. In some applications, the design airflow for the dedicated OA unit is dictated by the need to replace air that is being exhausted from the building. This is common in laboratories, commercial kitchens, or other applications with large exhaust requirements. In this case, the design airflow is the sum of all exhaust airflows plus any air needed for positive building pressurization.

Safety Factor for Zone Population

The first common use of a safety factor occurs when determining P_z, the number of people expected to occupy the zone. The definition of this term in Standard 62.1 states that this is "the largest number of people expected to occupy the zone *during typical usage*." This is not the largest number of people that could conceivably be in the zone under any special circumstance.

If actual zone population is not known, or if the owner and design team are not comfortable estimating it, Table 6-1 from

Standard 62.1 includes default values for occupant density. A note included under Table 6-1 clarifies that design engineers are not required to use this default, but may choose to if the actual occupant density is not known.

Using an overly conservative estimate for zone population can result in significant overventilation and excessive energy use. It also impacts the calculation of the space latent cooling load, and the resulting dehumidification capacity of the dedicated OA unit.

The recommendation of this author is to avoid applying safety factors to zone population (P_z), instead using the best estimate for expected occupancy during typical usage. Then, select the dedicated OA unit with "reserve" airflow capacity, as will be explained later in this article.

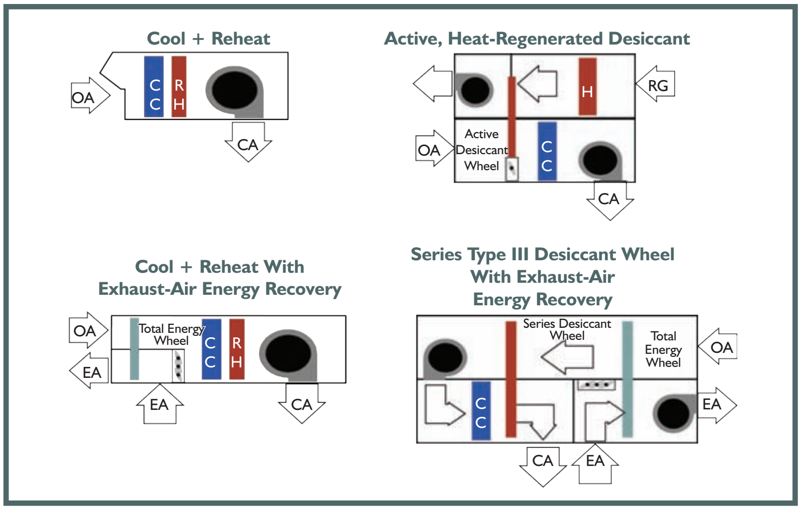

Figure 2: Example dedicated outdoor air equipment types.

Safety Factor for Future Expansion or Change of Use

The other common use of a safety factor when determining the design airflow of the dedicated OA unit occurs when accounting for future expansion or a change in use of the facility. If the facility is expanded in the future, and the dedicated OA unit will be expected to serve the expansion, it would be prudent to select a unit with some amount of reserve (or extra) airflow capacity. Or, if the use of the facility is likely to change in the future (from an office space to a group of meeting rooms or a retail area, for example) the ventilation requirements for the zones served by the dedicated OA unit may change. Again, it would be prudent to select a unit with reserve airflow capacity.

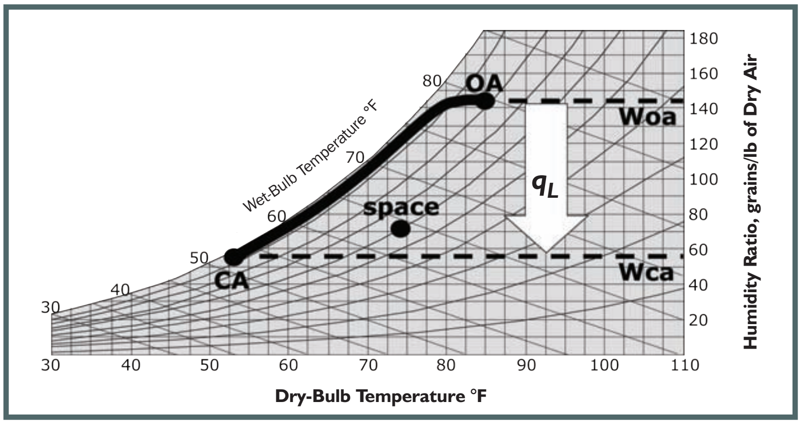

Figure 3: Dehumidification provided by a chilled water or direct-expansion dedicated outdoor air unit.

The recommendation of this author is to calculate the design airflow as accurately as possible, without using safety factors. Then select a dedicated outdoor air unit that has reserve airflow capacity, rather than a unit operating near its maximum allowable airflow. The impact of this will be discussed later in this article.

Determining Design Dehumidification Capacity

The required dehumidification capacity of a dedicated OA unit (*Figure 3, Equation 3*) is dictated by the design airflow (V_{ot}), humidity ratio of the entering outdoor air (W_{oa}), and humidity ratio of the conditioned air leaving the unit (W_{ca}):

$$q_L = 4.5 \times V_{ot} \times (W_{oa} - W_{ca}) / 7000 \text{ gr/lb} \quad (3)$$

where

q_L = required dehumidification capacity, lb/h
V_{ot} = design outdoor airflow, cfm
W_{oa} = humidity ratio of the entering outdoor air, grains/lb
W_{ca} = humidity ratio of the leaving conditioned air, grains/lb

Since the previous section discussed the influence of safety factors on the design airflow (V_{ot}), this section will focus on the remaining two variables in *Equation 3*: the humidity ratio of the entering outdoor air (W_{oa}) and the humidity ratio of the leaving conditioned air (W_{ca}).

Safety Factor for W_{oa} (Humidity Ratio of the Entering Outdoor Air)

ASHRAE Handbook—Fundamentals[7] is a popular source for climatic data that represents the outdoor design conditions for various locations. Historically, design engineers have used the design dry-bulb temperature and mean coincident wet-bulb temperature when calculating the required capacity of cooling systems. However, the highest outdoor humidity ratio does not occur at the same time as the highest outdoor dry-bulb temperature.

Since 1997, the *ASHRAE Handbook* has included the design dew-point temperature and mean coincident dry-bulb temperature for use when calculating the required capacity of dehumidification systems. *Table 1* lists the 0.4% outdoor design conditions for Jacksonville, Fla.[7] Notice that the outdoor humidity ratio is 32% higher at the design

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	Design Dry Bulb, Mean Coincident Wet Bulb	Design Dew Point, Mean Coincident Dry Bulb	Design Wet Bulb, Mean Coincident Dry Bulb
0.4% Design Condition	95°F DBT, 78°F WBT	78°F DPT, 85°F DBT	80°F WBT, 90°F DBT
Humidity Ratio	116 grains/lb	144 grains/lb	141 grains/lb
Enthalpy	41.0 Btu/lb	42.9 Btu/lb	43.8 Btu/lb

Table 1: Comparison of 0.4% outdoor design conditions (Jacksonville, Fla.)

dew-point condition than at the design dry-bulb condition. For this reason, it is important to use the design dew-point condition when determining the required dehumidification capacity of a dedicated outdoor air unit.

	0.4%	1%	2%
Design Condition	78°F DPT, 85°F DBT	77°F DPT, 84°F DBT	76°F DPT, 83°F DBT
Humidity Ratio	144 grains/lb	140 grains/lb	137 grains/lb

Table 2: Comparison of design dew-point conditions (Jacksonville, Fla.)

The first common use of a safety factor when determining the required dehumidification capacity of a dedicated OA unit occurs when selecting the design humidity ratio of the entering outdoor air (W_{oa}).

ASHRAE Handbook includes three sets of design conditions: 0.4%, 1%, and 2%. These indicate the percentage of hours during a year when with outdoor conditions are expected to exceed the tabulated design value. *Table 2* lists the 0.4%, 1%, and 2% design dew-point conditions for Jacksonville, Fla.[7]

Many engineers tend to use the most conservative (0.4%) design condition, but this often results in the selection of larger dedicated OA equipment. The impact of this decision on equip-

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	Classroom 101	Classroom 102	Classroom 103	Classroom 104
W_{sp}	75.2 grains/lb	75.2 grains/lb	75.2 grains/lb	75.2 grains/lb
Space Latent Cooling Load (q_{Lspace})	5250 Btu/h	5465 Btu/h	5697 Btu/h	5250 Btu/h
Zone Outdoor Airflow (V_{oz})	450 cfm	450 cfm	480 cfm	435 cfm
W_{ca}	58.3 grains/lb	57.6 grains/lb	58.0 grains/lb	57.7 grains/lb

Table 3: Example dedicated outdoor air system serving four classrooms.

ment capacity will be demonstrated and discussed later in this article.

Calculating the Humidity Ratio of the Conditioned Air Leaving the Unit (W_{ca})

The process for determining the required humidity ratio of the conditioned air delivered by a dedicated OA unit involves the following steps:[8]

1. Define the target humidity level for the occupied space (W_{sp}). This is the maximum indoor humidity that is considered acceptable. Some

	60% RH	55% RH	50% RH
W_{sp}[1]	75.2 grains/lb	68.8 grains/lb	62.5 grains/lb
W_{ca}	57.6 grains/lb	51.2 grains/lb	44.9 grains/lb
TDP_{ca}	52.0°F	48.9°F	45.4°F
Dedicated OA Unit Dehumidification Capacity (q_L)[2]	100.8 lb/h	108.2 lb/h	115.6 lb/h

[1] Assumes the same 74°F (23°C) dry-bulb temperature setpoint for the zones

[2] Calculated per *Equation 3* using the 0.4% design dew-point condition for Jacksonville, Fla. (included in *Table 2*), the design latent loads and ventilation airflows for each zone (listed in *Table 3*), and $V_{ot} = \Sigma_{all\ zones} V_{oz}$ = 1815 cfm

Table 4: Impact of safety factors on W_{ca} ***and*** q_L.

	Smaller Unit	Larger Unit
Cooling Coil Face Velocity	508 fpm	401 fpm
Cooling Coil Pressure Drop	0.96 in. H_2O	0.60 in. H_2O
Wheel Pressure Drop	1.1 in. H_2O	0.79 in. H_2O
Wheel Total Effectiveness (Cooling)	71%	76%
Wheel Capacity (Cooling)	180 MBh	190 MBh
Supply Fan Brake Horsepower	4.8 hp	3.3 hp
Exhaust Fan Brake Horsepower	3.8 hp	2.5 hp

***Table 5:** Energy-related impacts of equipment sizing.*

engineers consider 60% RH an acceptable upper limit, while others choose 50%, 55%, 65%, or some other limit.

2. Determine the design latent load for each zone (q_{Lspace}). This is the latent load that occurs within the boundaries of the zone. It typically consists of moisture generated by people or other sources within the zone, as well as infiltration or diffusion of humid air from outdoors or from adjacent zones.[9] These loads are often determined with the help of load calculation software.
3. Calculate the required humidity ratio of the conditioned air (W_{ca}) delivered by the dedicated OA unit. If the dedicated OA system is being designed to offset the entire indoor latent load, in addition to the ventilation latent load, then the conditioned outdoor air must be dry enough to offset the latent load in each zone, such that the humidity ratio in every zone is maintained at or below the desired upper limit. *Equation 4* is used to determine the required W_{ca} for each zone. Then, the humidity ratio of the conditioned outdoor air delivered by the unit must be the lowest (or worst-case) of all the zones it serves.

$$W_{ca} = W_{sp} - [q_{Lspace} / (0.69 \times V_{oz})] \tag{4}$$

where

W_{ca} = required humidity ratio of the conditioned air leaving the unit, grains/lb

W_{sp} = desired upper limit for humidity ratio of the occupied space, grains/lb

q_{Lspace} = design latent load in the zone, Btu/h

V_{oz} = design zone outdoor airflow, cfm

To demonstrate this process, consider an example elementary school located in Jacksonville, Fla. A dedicated outdoor air unit is being designed to deliver air directly to four classrooms, removing the entire ventilation latent load *and* the entire indoor latent load.

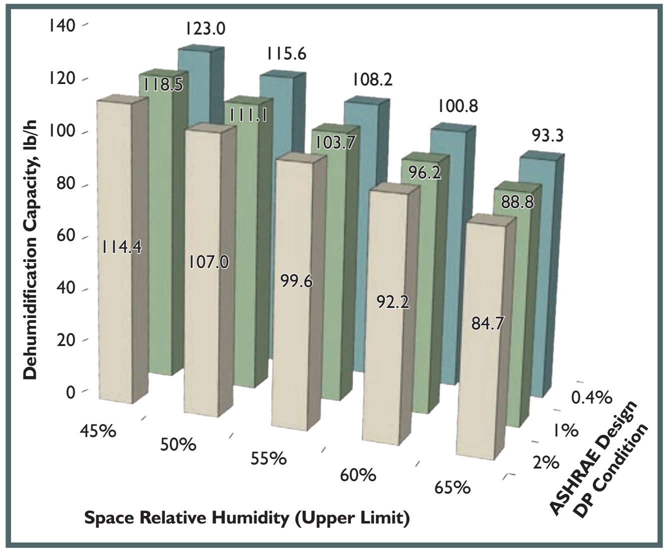

***Figure 4:** Impact of W_{oa} and W_{sp} on required dehumidification capacity (q_L), without total-energy recovery (Jacksonville, Fla.)*

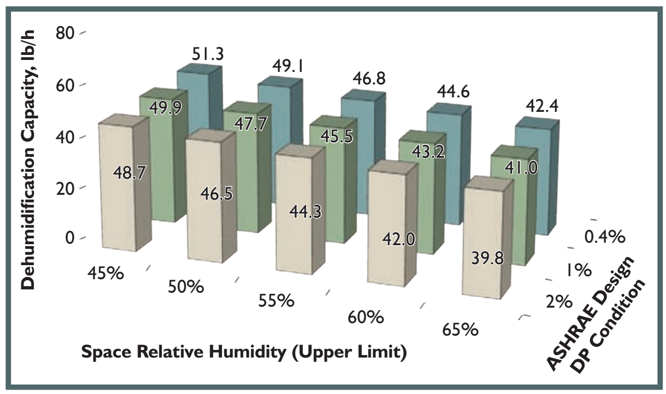

***Figure 5:** Impact of W_{oa} and W_{sp} on required dehumidification capacity (q_L), with total-energy recovery (Jacksonville, Fla.).*

1. For this example, the zone dry-bulb temperature setpoint during the cooling season will be 74°F (23°C) and the acceptable upper limit for indoor humidity is 60% RH. This equates to an indoor humidity ratio (W_{sp}) of 75.2 grains/lb (10.8 g/kg).
2. *Table 3* (see Page 35) lists the design latent loads (q_{Lspace}) and required outdoor airflows (V_{oz}) for each zone.
3. *Equation 4* is used to calculate the required conditioned-air humidity ratio (W_{ca}) for each zone. For example, the required humidity ratio of the conditioned outdoor air delivered to Classroom 102 is 57.6 grains/lb (8.24 g/kg).

$$W_{ca} = 75.2 \text{ grains/lb} - [5465 \text{ Btu/h} / (0.69 \times 450 \text{ cfm})] = 57.6 \text{ grains/lb}$$

As the results in *Table 3* show, Classroom 102 requires the driest air. Therefore, for this example, the dedicated outdoor air unit must be sized to dehumidify the outdoor air to 57.6 grains/lb (8.24 g/kg), which equates to a 52°F (11°C) dew

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point (drier than the space), to remove the entire indoor latent load (*Figure 3*).

Safety Factor for W_{ca} (Humidity Ratio of the Conditioned Air Leaving the Unit)

Target Space Humidity Level (W_{sp}). The first influence of safety factors in determining the required humidity ratio of the conditioned air (W_{ca}) is in the selection of the target humidity level for the occupied space (W_{sp}). For most comfort-cooling applications, engineers will typically design HVAC systems using a target relative humidity (RH) of 50%. However, the allowable upper limit (worst case) for indoor humidity may be higher. Some engineers consider 60% RH an acceptable upper limit, while others use 55%, 65%, or some other limit. The new *ASHRAE Guide for Buildings in Hot and Humid Climates*[10] recommends limiting indoor humidity to no greater than 55°F (13°C) dew point, which equates to about 51% RH at 74°F (23°C) dry bulb.

This decision has a huge impact on the sizing of the dedicated outdoor air unit. Using the same four-classroom example, the humidity ratio of the conditioned air (W_{ca}) and the resulting dehumidification capacity of the dedicated OA unit were determined for various RH limits. As *Table 4* (see Page 35) demonstrates, to maintain a lower RH in the zones, W_{ca} must be lower, resulting in greater dehumidification capacity required.

For this example, designing the system to limit indoor humidity to 50% RH increases the required dehumidification capacity of the dedicated OA unit by 15% compared to allowing indoor humidity to rise to 60% RH at worst-case conditions.

However, be sure to analyze the impact on the entire system. When a chilled-water or DX dedicated OA unit delivers the conditioned air cold (that is, not reheated to "neutral"), a dedicated OA unit that is selected to deliver drier air (lower W_{ca}) delivers the air at a colder dry-bulb temperature. This colder air offsets more of the space sensible cooling load, allowing for the local HVAC units to be smaller and use less energy.[11]

Design Indoor Latent Loads (q_{Lspace}). The second influence of safety factors in determining the required conditioned-air humidity ratio (W_{ca}) is in the calculation of the design latent loads for each zone. As discussed earlier, the estimated zone population (P_z) not only impacts ventilation calculations, it also affects indoor latent load calculations.

Another source of latent load in a zone is infiltration of humid air from outside. Calculating loads due to infiltration is often more of an art than a science because it is highly dependent on the quality of the building envelope construction.[9] Since precise calculations are often difficult, many engineers choose to select the dedicated OA unit with reserve (or extra) dehumidification capacity, rather than attempt to apply safety factors to the calculation of the latent load due to infiltration.

Summary and Discussion

Excessive use of safety factors in the design of a dedicated outdoor air system (DOAS) can result in significant overventilation and larger-than-necessary equipment. This may inflate installed costs and can result in excessive energy use.

To minimize the impact of safety factors on installed cost and energy use, the recommendations of this author are as follows:

1. Avoid applying safety factors to zone population (P_z) when calculating the required outdoor airflow (V_{oz}) and design latent load (q_{Lspace}) for each zone. Estimate population and loads as accurately as possible, then select a dedicated OA unit that has reserve (or extra) airflow and/or dehumidification capacity. This may result in the selection of larger equipment, which can increase installed cost, but there are often energy-related benefits associated with a larger unit. The example in *Table 5* (see Page 36) shows two dedicated OA units selected to deliver the same airflow and provide the same dehumidification capacity. The larger casing size has more reserve capacity to accommodate unexpected loads and the need for increased airflow or dehumidification. The larger unit, when operating at the same design airflow as the smaller unit, has a lower airside pressure drop across both the cooling coil and the total-energy wheel, resulting in less fan power. In addition, the lower face velocity across the wheel results in higher effectiveness and more energy-recovery capacity.

   However, selection of the dedicated OA unit is influenced by the type of dedicated OA equipment being used. For example, DX dehumidification equipment may be less tolerant of being selected with reserve capacity than chilled water or desiccant-based equipment, because of the need to properly match airflow range with the operating envelope of the refrigeration equipment.

2. When defining the design outdoor conditions (W_{oa}) and the target space humidity level (W_{sp}) analyze the impact of these decisions up front and share that information with the owner to assist in making the final decisions. Using the same four-classroom example in Jacksonville, Fla. (*Table 3*), *Figures 4* and *5* (see Page 36) demonstrate the impact of the selected ASHRAE design dew-point condition (0.4%, 1%, or 2%) and the desired upper humidity limit for the occupied space (W_{sp}) on the required dehumidification capacity (q_L) of the dedicated OA unit. This is the capacity needed to offset the entire ventilation latent load plus all of the indoor latent loads. (*Figure 4* shows the required capacity if the unit has no exhaust-air energy recovery, while *Figure 5* shows the required capacity if the unit includes a total-energy recovery device with 70% latent effectiveness.) These figures emphasize the following points:

- Designing the DOAS for a more severe outdoor condition, or for a lower indoor humidity level, both increase the required dehumidification capacity of the dedicated OA unit. In this example, the result of these decisions can impact capacity by as much as 45%.
- Using total-energy recovery can significantly reduce the required capacity of the chilled-water coil, DX refrigeration circuit, or desiccant-based dehumidifier. But, it also dampens the impact of these more-conservative design decisions. Without exhaust-air energy recovery, designing the system for the 0.4% ASHRAE design dew-point condition and 45% indoor RH increases the required dehumidification capacity by about 38.3 lb/h (17.4 kg/h) compared to using the 2% design condition and 65% indoor RH (*Figure 4*). However, with total-energy recovery, the more conservative design decisions increase the required dehumidification capacity by only 11.5 lb/h (5.2 kg/h) (*Figure 5*).

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