Calculating Dew-Point Design for DOAS

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The combination of warm–humid climates and building tightness pose a unique challenge in designing systems. Significant ventilation air quantities are necessary to maintain indoor air quality and meet code requirements, making it difficult to properly manage relative humidity. When internal building latent loads (such as high density occupancy) are combined with the high latent loads of infiltration and code-required ventilation air in warm–humid climates, thermostatic control alone cannot maintain appropriate levels of relative humidity (RH).

Dedicated outdoor air systems (DOAS) have become popular as a delivery mechanism in that they provide a practical means for decoupling latent and sensible loads and can be set up to control dew-point temperature, ensuring control of and maintenance of appropriate relative humidity levels within the conditioned space. Developing design and operating criteria for DOAS can be simplified using a grains/pound (g/kg) reduction calculation to determine the proper dew-point temperature of the DOAS supply airstream to specify.

Background

Humidity control in warm and humid climates is imperative. Using DOAS as the primary means of moisture removal has consistently gained greater acceptance worldwide. Providing a comfortable and healthy indoor environment has been a difficult task for many commercial applications where ventilation rates are high; these buildings include, but are not limited to, hospitals, schools, theaters, retail stores, hotels, restaurants, and nursing homes.[1]

The International Building Code[2] requires mechanical construction to be in compliance with the International Mechanical Code,[3] that in turn requires heating and cooling system design loads be determined per ASHRAE/ACCA Standard 183-2007, *Peak Cooling and Heating Load Calculations in Buildings Except Low-Rise Residential Buildings,* that in turn requires loads to account for capacity required to accomplish psychrometric processes that includes dehumidification.[4]

The *2013 ASHRAE Handbook—Fundamentals* contains specifics for design conditions, ventilation, and humidity

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control.[5] ASHRAE recognizes that too high or too low relative humidity affects indoor air quality and human comfort and addresses it in ASHRAE Standard 62.1[6] and numerous other publications.

Humidity and Comfort

The combination of temperature and relative humidity is the primary component of perceived human comfort.[7] Increased levels of humidity weaken the water vapor partial pressure gradient between the skin surface and the air, restricting the evaporative processes.[8] The result of increasing humidity on human comfort is an amplifying effect of perceived air temperature increases, resulting in a decrease in human thermal comfort, or conversely, an increase in discomfort.

As shown in *Figure 1*, the apparent or perceived air temperatures increase dramatically with increasing relative humidity. These perceived temperature increases result in increased sweating and increased blood flow to the skin, the body's largest organ, and with the corresponding reductions in evaporation and increased blood flow, the temperature of the skin increases. The body responds by increasing sweat rates in an attempt to facilitate evaporative cooling, which is inhibited. The results within the range of temperatures found within a building are slight increases in body temperature, increased skin wetness, and decreased thermal comfort.[9]

Most people prefer indoor temperature generally in the low- to mid-70s (°F) (mid-20s [°C]). As relative humidity increases, the temperature must be reduced to maintain the same level of comfort and vice versa. As shown in *Figure 1*, an actual temperature of 75°F (23.9°C) feels like 76°F (24.4°C) at 60% RH, and feels like 78°F (26.6°C) at 80% RH. These small variations in perceived temperature have a negative effect on human perceptions of thermal comfort and the ability of the human body to adequately maintain comfort.

Standards

The *2013 ASHRAE Handbook—Fundamentals* provides three design conditions that the designer should evaluate in the design of HVAC systems: cooling (DB/MCWB), evaporation (WB/MCDB), and dehumidification (DP/HR/MCDB). Design conditions for warm and humid climates should be based on dew-point temperatures, as dew-point temperatures are directly related to extremes

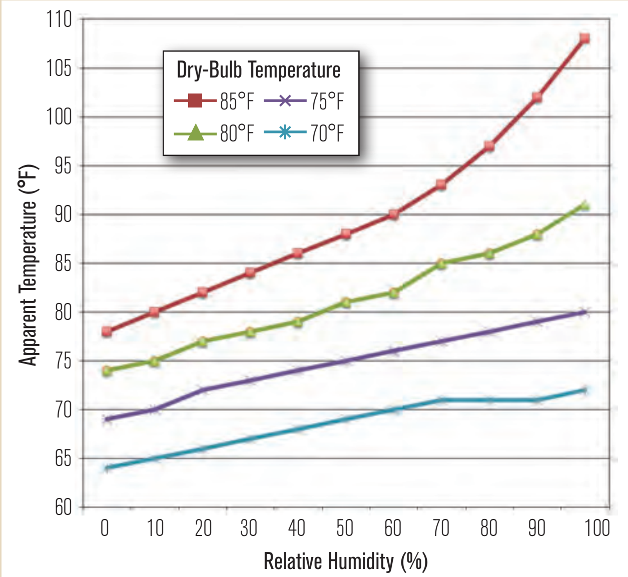

FIGURE 1 Apparent (perceived) temperatures associated with changes in relative humidity. Lines represent perceived temperature at various actual temperatures. (Adapted from: Larrañaga and Bernard.[9])

of humidity because of weather. The dehumidification condition provides the appropriate design basis for humidity control applications because of the high humidity ratios associated with warm and humid climates. This is especially useful in designing desiccant dehumidification, cooling-based dehumidification, and outdoor air ventilation systems.[5]

ASHRAE Journal has published many articles concerning design of HVAC systems and the application of DOAS in warm and humid climates,[1,10–13] as well as guide books for buildings in hot and humid climates.[14,15] The general consensus is that decoupling of the sensible and latent loads by applying DOAS is a practical and reliable way to deal with buildings that have a significant ventilation component in warm and humid climates.[14,15]

All too often design engineers rely too heavily on computer-generated load designs to provide design information that should result in satisfactory indoor conditions. Adequate humidity control for an air-conditioned space with a significant ventilation and/or infiltration component in a warm and humid climate can be a challenge.

A review of HVAC equipment schedules for 25 schools designed in the last decade for south Texas projects showed that the design in 96% of the cases did not adequately compensate for the total latent load, and none of these designs considered infiltration to be a significant variable impacting total latent load.

Building Latent Load

Calculations for determining building latent load for a typical school classroom in climates ranging from warm and humid climate to a dry climate are provided as examples in this paper for comparison. Example cities include McAllen, Texas, on the United States/Mexico border 70 miles (113 km) inland from the Gulf of Mexico; Amarillo, Texas, in the north part of the panhandle of west Texas; as well as other cities such as Dallas, New Orleans and Sarasota, Fla. (*Table 1*).

TABLE 1 ASHRAE 1% design conditions[16] and 2012 IECC Climate Zone classifications.

	ASHRAE 1% DESIGN CONDITIONS			
LOCATION	COOLING (DB/MCWB)	EVAPORATION (WB/MCDB)	DEHUMIDIFICATION (DP/HR/MCDB)	IECC CLIMATE ZONE
McAllen, Texas	99/76.5	79.8/89.7	77.7/144.5/82.2	2A – Warm–Humid, Moist
Amarillo, Texas	94.7/66.3	70.2/85.3	66.1/110.1/74.4	4B – Dry

ASHRAE Standard 62.1-2013 (Table 6.2.2.1) prescribes a minimum ventilation rate comprised of a People Outdoor Rate (cfm/person [L/s·person]) plus an Area Outdoor Air Rate (cfm/ft^2 [L/s·m]). Optionally, Occupant Density and a Combined Outdoor Air Rate based on default occupant density can be used. The default value for ventilation rate in this paper is 13 cfm (6 L/s) per occupant in the breathing zone in a school classroom (age 9 plus).[6]

The Texas Education Code requires classrooms in public school buildings to be constructed for 25 students (occupants). Therefore, 25 occupants per classroom is used in the calculations. Determining the actual occupant count is key to actual energy use, and the reader is so advised to carefully select the occupant load when designing the system and controls.

The basic principle is that air must be dehumidified to sufficiently sorb all latent loads to achieve and maintain required conditions in the occupied space (<65% RH). The source of various latent loads must be recognized and evaluated in a design. See *Figure 2*.

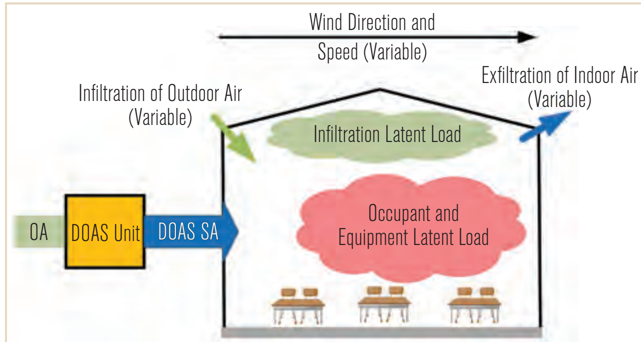

FIGURE 2 The origin of the three basic sources of latent loads.

Calculations

The magnitude of various latent loads must be determined.

Infiltration latent loads, internal latent loads, and the latent load to remove moisture from the ventilation air all add up as the total latent load the HVAC system must handle to provide acceptable indoor environmental conditions and acceptable indoor air quality at dehumidification design conditions. The calculations presented in the sidebars ("Calculations: Latent Loads From Occupants" and "Calculations: Latent Load From Infiltration") illustrate how to apply the gr/lb (g/kg) method to determine dew point of air required to sorb total latent loads in a building.

Applications

The purpose of ventilation air is to dilute pollutants. DOAS applications are generally applied in two basic formats:

1. Conditioned outdoor air is distributed directly to the occupied space. When this application is used, several variations can be used consisting of constant volume and variable volume applications.
2. Conditioned outdoor air is mixed with the return airstream of air handler equipment supplying conditioned air to the occupied space.

For DOAS applications where conditioned air is distributed directly to the occupied space, the equipment must be selected to handle the latent load of ventilation air and infiltration. If the designer so chooses, the occupant latent load can be handled by the air handler equipment providing conditioned air to the occupied space provided that the air can be continuously maintained at a dew point sufficient to sorb the latent load at all times.

For DOAS applications where conditioned outdoor air is mixed with the return airstream of air handler equipment, the latent and sensible loads, for all practical purposes, are decoupled with the DOAS doing all the latent work and the air handler coil doing the sensible work.

Calculations: Latent Loads From Occupants

Constants and Conversion Factors

1,076 Btu/h of latent work = 1 lb/h water vapor removal

7,000 grains water vapor (gr) = 1 lb of water

1 gr/lb = 0.143 g/kg

Latent Heat Conditional Constant[17]

$$[60 \text{ min/h/specific volume } (70°\text{F}, 50\% \text{ RH})] \times [\text{specific heat } (\Delta h_{vap})/7{,}000 \text{ gr/lb}] = 0.68$$

$$(60/13.5) \times (1{,}076/7{,}000) = 0.68$$

Latent load produced by typical occupant = 200 Btu/h·person = Q_L

Indoor Air Conditions (IDA): 75°F/65% RH = 84.8 gr/lb, 62°F DP

People Latent Load

The equation below illustrates the gr/lb of moisture due to a single occupant that must be sorbed by the ventilation air (13 cfm/person) over the period of an hour.

$$\Delta \text{ gr/lb} = (Q_L)/(0.68 \times \text{ventilation rate}_{cfm})$$

$$\Delta \text{ gr/lb} = (200 \text{ Btu/h/person})/(0.68 \times 13 \text{ cfm/person}) = 22.6 \text{ gr/lb}$$

Δ gr/lb = 22.6 gr/lb, which represents the hourly latent moisture introduction by a single occupant that must be removed by the 13 cfm/person of ventilation air.

$$\text{IDA} - \Delta \text{ gr/lb} = 84.8 \text{ gr/lb} - 22.6 \text{ gr/lb} = 62.2 \text{ gr/lb}$$

Occupant Latent Load Calculation

25 occupants per classroom

$$Q_L = 25 \times 200 \text{ Btu/h} = 5{,}000 \text{ Btu/h}$$

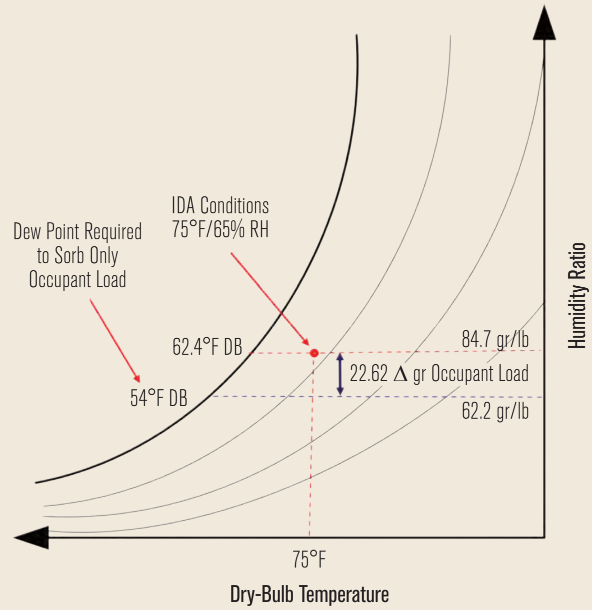

FIGURE 3 The internal load created by occupants requires the dew point of the air in the space to be ≤54°F continuously to provide sufficient sorption of moisture emitted by occupants only to maintain the indoor relative humidity <65% RH at 75°F DBT.

Other internal loads are created by appliances and infiltration. As examples: 1) appliance—in the case of a science laboratory, Bunsen burners give off heat as well as water vapor (latent heat) from combustion; 2) infiltration—the combination of wind pressures and openings in the building enclosure, which allow unconditioned air to find its way into the building bringing humidity (latent load) with it.

It is intuitive for one to think that low dew-point air delivered from a DOAS is only applicable for hot and humid climates. However, peak dehumidification design loads occur when outdoor humidity is high and dry bulb temperatures are moderate (i.e., warm and humid) and peak dehumidification design conditions are not the only time where maintaining a target relative humidity in a classroom is difficult.[15,23] Considering that the occupant component is not related to the outdoor conditions and that the outdoor humidity ratio above indoor conditions is additive to infiltration, one only needs to observe the weather history to realize that the need for dehumidification is not occasional. *Figure 5,* (Page 30) from weatherspark.com (a subscription service of weather data based on NOAA's data, McAllen Miller International Airport) documents dew-point history for 2014 plotted against dew point. The 75°F, 65°F, 55°F and 45°F (24°F, 18°F, 13°C and 7°C) scale lines were added for clarity. These data clearly illustrate high dew-point conditions exist throughout much of the year in this warm and humid climate.

Table 2 illustrates calculated results for typical school classrooms in five different cities indicating that DOAS-delivered air conditions are lower in the warm and humid climates than the dry climates due to infiltration, but never above 54°F (12°C) dew point due to the occupant latent load with an indoor relative humidity target of 65% at 75°F (24°C) dry bulb.

Calculations: Latent Load From Infiltration

Infiltration Latent Load

Standard 62.1 requires outdoor air for ventilation and building pressurization. All buildings have some leakage. Many design professionals mistakenly assume that if ventilation airflow is greater than the exhaust airflow then any leakage will be from inside to outdoors, eliminating the leakage moisture load. That assumption is patently false, as not a single field investigation validates that optimistic assumption. In fact, quite the reverse.[18–20] All buildings pull in some outdoor air at some times—even when the overall average internal air pressure is positive.[15]

To understand the issue, one only needs to recognize that a positive building pressure of 0.01 in. w.c. (2.5 Pa) is only capable of resisting wind velocity pressure <0.01 in. w.c. (2.5 Pa) (5 mph [8 km/h]). The velocity pressure for a 15 mph (24 km/h) gust is 0.11 in. w.c. (27.5 Pa), which is 11 times greater than a positive building pressure of 2.5 Pa (0.01 in. w.c.).

Not only will wind push unconditioned outdoor air into the building (infiltration) on the windward side it will evacuate (exfiltration) conditioned air from the building on the leeward side. Both positive pressure and a reasonably tight building are critical to adequately managing indoor environmental conditions and indoor air quality.

Estimating infiltration leakage rate is difficult. The U.S. Department of Energy (DOE) has provided useful methodology for energy modeling using a reference wind speed of 10 mph (16 km/h) and a reasonable infiltration rate for an average building. For the purpose of illustration in this paper, two infiltration rates were applied:

1. 1.8 cfm/ft^2 at 0.30 in. w.c. (9 L/s·m^2 at 75 Pa) is the "average" leakage rate for exterior walls in commercial buildings referenced in Chapter 25, Ventilation and Infiltration, of the *1997 ASHRAE Handbook—Fundamentals* and other ASHRAE publications.[14,15]

Pacific Northwest National Laboratory (PNNL) published "Infiltration Modeling Guidelines for Commercial Building Energy Analysis," which provides a methodology for modeling air infiltration and recommends the DOE-2 model, which uses an infiltration rate of 0.2016 cfm/ft^2 (1 L/s) of exterior wall area, assuming that uncontrolled air leakage through the building envelope can be specified by a baseline leakage rate of 1.8 cfm/ft^2 at 0.30 in. w.c. (9 L/s·m^2 at 75 Pa) of exterior above-grade envelope area.[21] This value represents the typical leakage rate for commercial buildings with a tightness rating of average.[14,15]

2. Section C402.4.1.2.3 of the 2012 International Energy Conservation Code requires the complete building to be tested and the air leakage rate of the building envelope not exceed 0.40 cfm/ft^2 at 0.30 in. w.c. (2 L/s·m^2 at 75 Pa). This is a new code requirement and anticipates a building rated as tight. Using the same procedure used in the PNNL report to determine the value of 0.2016 cfm/ft^2 (1 L/s) of exterior wall surface for the baseline leakage rate of 1.8 cfm/ft^2 at 0.30 in. w.c. (9 L/s·m^2 at 75 Pa), the IECC test value of 0.4 cfm/ft^2 at 0.30 in. w.c. (2 L/s·m^2 at 75 Pa) yielded

Considering that the DOE-2 criteria is classified as an "average" building and the IECC criteria is classified as a "tight" building, *Figure 6* shows that operation of a building to maintain a target relative humidity in the mid-50s requires a lower dew point than most engineers conventionally consider in their designs for the occupied space to sorb occupant, infiltration, and ventilation loads in their designs.

Further, *Figure 6* (Page 30) shows the DOAS supply air dew point necessary to achieve and maintain a target relative humidity indoors with both average and tight building enclosures. It also shows that infiltration is the most variable element in driving the supply air dew point down below the occupant component at any target indoor relative humidity below the 65% RH required by Standard 62.1. Building tightness presents a significant challenge to achieving and maintaining the designer's target indoor relative humidity.

Energy

To create and maintain acceptable indoor conditions, humidity must be controlled requiring that moisture must be removed from the air in most climates, especially in warm and humid climates. Removing moisture an infiltration value of 0.0448 cfm/ft^2 (0.02 L/s·m^2) of exterior wall surface.[22]

Infiltration Latent Load Calculation (McAllen, Texas)

Square foot area of outside wall × leakage rate[14,15] = infiltration air (cfm).

Using a 0.2016 cfm/ft^2 of wall Leakage Rate × 300 ft^2 = 60.5 cfm infiltration

$$\Delta \text{ gr/lb} = (\text{Infiltration air gr/lb} - \text{IDA gr/lb}) = (144.5 \text{ gr/lb} - 84.8 \text{ gr/lb}) = 59.7\ \Delta \text{ gr/lb}$$

$$Q_L = 0.68 \times \text{cfm} \times \Delta \text{ gr/lb} = 0.68 \times 60.5 \times 59.7 = 2{,}456 \text{ Btu/h}$$

Outdoor Air (OA) for Ventilation Latent Load Calculations (McAllen, Texas)

13 cfm/person × 25 students/classroom = 325 cfm

$$325 \text{ cfm} \times 0.68 \times \Delta \text{ gr/lb} = \text{Btu/h}$$

$$\Delta \text{ gr/lb} = (\text{OA gr/lb} - \text{IDA gr/lb}) = (144.5 \text{ gr/lb} - 84.8 \text{ gr/lb}) = 59.7\ \Delta \text{ gr/lb}$$

$$Q_L = 0.68 \times \text{cfm} \times \Delta \text{ gr/lb} = 0.68 \times 325 \times 59.7 = 13{,}194 \text{ Btu/h}$$

Combined Latent Load (McAllen, Texas)

Occupants + Infiltration + Ventilation = 20,650 Btu/h

DOAS Supply Air gr/lb and Resultant Dew Point (McAllen, Texas)

The following calculation determines the gr/lb and resultant dew point of conditioned OA required for sorption of all latent loads (ventilation, infiltration, and occupants) where DOAS is mixed with return air for distribution by the air handler unit:

$$Q_L = 0.68 \times \text{cfm} \times \Delta \text{ gr/lb}$$

$$20{,}650 \text{ Btu/h} = 0.68 \times 325 \text{ cfm} \times \Delta \text{ gr/lb}$$

$$\Delta \text{ gr/lb} = 20{,}880/(0.68 \times 325) = 93.4 \text{ gr/lb}$$

$$\Delta \text{ OA gr/lb required for sorption} = (144.5 \text{ gr/lb} - 93.4 \text{ gr/lb}) = 51.1 \text{ gr/lb}\ \Delta\ 49°\text{F DP}$$

Figure 4 illustrates the relationship of the above calculations on a psychrometric chart for 65% RH (at 75°F [24°C] dry-bulb temperature) and DOE-2 infiltration model in McAllen, Texas. The occupant component will always be below the dew point of the indoor conditions selected by the engineer, and the infiltration component will be additive to the occupant component. The ventilation component for reduction of humidity ratio of outdoor air will be between the ASHRAE-defined dehumidification design condition humidity ratio and the indoor conditions selected by the engineer. The sum of all three will define the moisture removal capacity required of the DOAS.

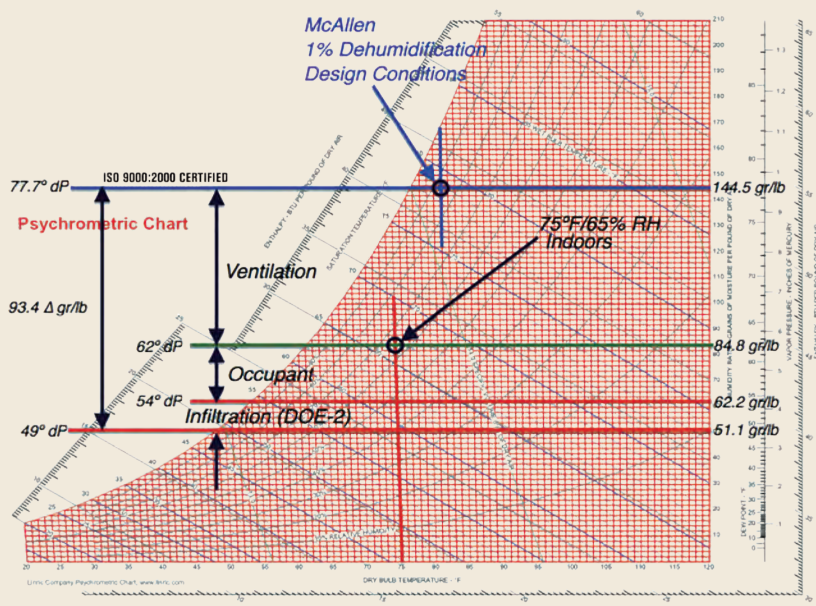

FIGURE 4 Psychrometric chart. (Source: Munters Corporation, printed with permission.)

from air requires an expenditure of energy regardless of the technology used for removal. Many perceive that DOAS is expensive to operate. This perception can either be true or untrue depending on the technology applied. For example, reheat is more expensive to operate than desiccant technologies.

When practical, the application of heat recovery technologies such as heat wheels to recover energy from exhaust is a worthwhile consideration. However, the designer must keep in mind that in order to maintain positive building pressure there will be unbalanced flow impacting heat recovery performance. Dependence on total energy passive heat wheels to sufficiently treat OA will not perform adequately in warm and humid climates. The application of total energy heat wheel technology to recover some energy and precondition OA prior to applying mechanical and/or desiccant means and methods of moisture removal with DOAS have been proven to work well in warm and humid climates.

When using active desiccant technologies, selecting the energy source for reactivation of the desiccant is the key element in energy conservation. Mechanical designers should perform a cost-benefit analysis considering first cost, operating complexity, maintenance cost, and energy use over the service life of the equipment. Careful analysis of these data will likely support a strategy for decoupling the latent and sensible loads with DOAS.

Conclusion

Achieving and maintaining an acceptable indoor relative humidity in most climates is a function of control by providing low enough dew-point air in the right place to sorb latent loads.

Considering that most computerized load programs used by the engineering community generally default to cooling conditions (DB/MCWB), designs are commonly insufficient for humidity control unless designers consider the dehumidification design conditions (DP/HR/MCDB) and make allowances for all the latent loads in their calculations, applications, and equipment selections. Supply air dew points in the low- to mid-50s °F (mid-10s °C) will not handle the latent loads of occupants plus infiltration in typical school classrooms of average tightness at dehumidification design conditions in most parts of the country.

The infiltration component of the latent load in average tightness or loose buildings contributes a significant part of the load requiring a lower dew point than most HVAC engineers anticipate. Even buildings that pass the 2012 IECC tightness test have a lower dew-point requirement than traditional cooling coil applications can achieve to maintain a desirable design relative humidity target in the occupied space.

Designers must no longer ignore or discount the impact infiltration can have on the ability of an HVAC system to control latent load. Engineers should pay particular attention to the as-built condition of the building enclosure in the construction phase of projects to ensure predicted performance.

Where ventilation is a significant part of the latent load and/or a low dew point is necessary to achieve results, DOAS is a proven method of managing indoor relative humidity, especially for warm and humid conditions. Moving latent load from the air handler cooling coil to DOAS by decoupling the sensible and latent loads is a reliable and practical solution.

The gr/lb (g/kg) reduction method of determining the dew point necessary to sorb the components and/or combination of latent loads is a simple and reliable method for determining the supply air dew-point condition at dehumidification design conditions.

Alternatively, this procedure provides a reliable quality control method to verify computer load program outputs for sufficient latent load removal capacity.

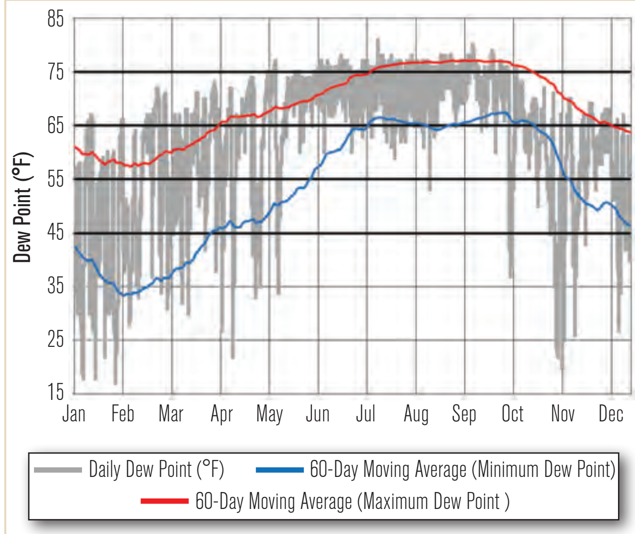

FIGURE 5 2014 daily dew-point history for McAllen-Miller International Airport (Texas). (Source: weatherspark.com, printed with permission.)

TABLE 2 DOAS supply air dew point at ASHRAE 1% dehumidification design conditions.[16]

	SUPPLY AIR DEW POINT FOR DOAS TO MAINTAIN 65% RH				
Infiltration	McAllen, Texas	Dallas	Amarillo, Texas	New Orleans	Sarasota, Fla.
DOE-2	49	50	52	49	48
IECC	53	53	53	53	53

Note: Data is for an indoor relative humidity target of 65% at 75°F DB. A lower indoor relative humidity target, or a lower indoor dry-bulb temperature, will result in a lower supply air dew point.

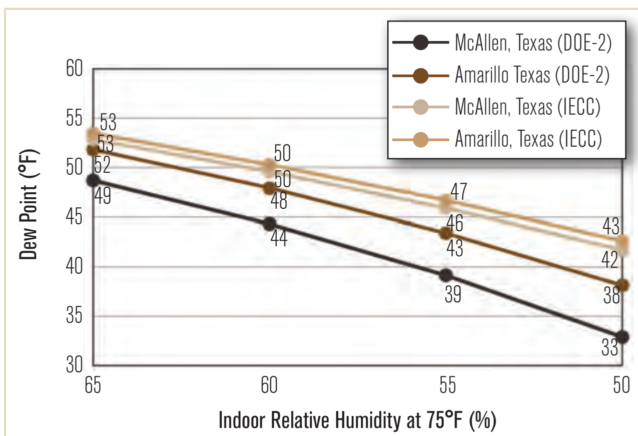

FIGURE 6 Supply air dew-point values required to achieve indoor relative humidity targets for both the average (DOE-2) and tight (IECC) building for dehumidification design conditions.

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