TRANE

engineers newsletter

volume 35–3 • providing insights for today's hvac system designer

maintaining a comfortable environment in Places of Assembly

Designing comfort systems for places of assembly (auditoriums, gymnasiums, arenas, houses of worship) presents some vexing challenges. Such facilities often have acoustical requirements that place limits on equipment location and air distribution design. Many places of assembly experience extremely diverse loads and occupancy schedules, complicating part-load system control. But perhaps the biggest challenge is occupancy itself, and its impact on ventilation and humidity control. Design guidelines that are commonly applied in commercial office space mayget us into trouble here.

A simple example can illustrate some of these issues: a school gymnasium during a band concert. As this is a good high school band, both the bleachers and the floor are full. Occupancy is at the fire marshal's rated seating capacity. The 18,000 ft^2 gymnasium is designed for 1200 people, including use of the gym floor. A load calculation reveals the following space loads:

Roof	69,600 Btu/hr
Wall	43,000 Btu/hr
Glass	10,500 Btu/hr
Lights	122,900 Btu/hr
People	300,000 Btu/hr (sensible)
	240,000 Btu/hr (latent)
Totals	546,000 Btu/hr (sensible)
	240,000 Btu/hr (latent)

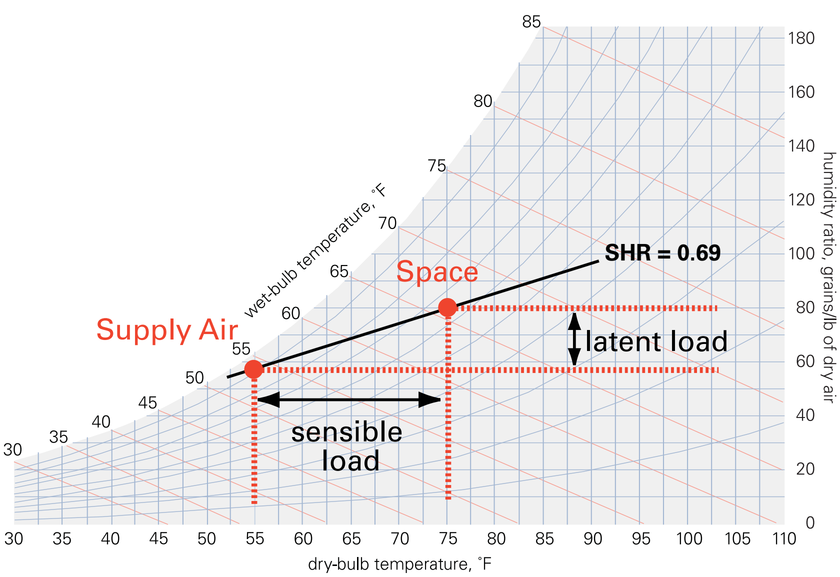
Occupancy a Major Factor

People constitute a significant portion of the space sensible cooling load, over 50 percent. However, it is the impact on humidity that makes occupancy a difficult load to manage. The space sensible heat ratio for this example is only 0.69 (Figure 1).

If the target space comfort condition is 75°F and 50 percent relative humidity (RH), and the air distribution system is designed for 55°F supply air, the required supply airflow is over 4 cfm per square foot of floor area -- a huge amount! How did that happen?

Humidity ratio tells the story. Humidity ratio is grains of water vapor per pound of air. The humidity ratio at 75°F dry bulb and 50 percent RH, the desired space condition, is 64.7 grains of water vapor per pound of air. The 55°F supply air has a humidity ratio of 60.4 gr/lb. At these conditions, each pound of supply air we introduce into the space can remove 4.3 grains of water vapor. If each occupant contributes 200 Btu/hr to the latent load, 1200 people add 230 pounds of water vapor (1,610,000 grains) to the air in the gymnasium. If each pound of supply air can remove only 4.3 grains of water vapor, it will take 374,000 pounds of supply air per hour. This equates to approximately 83,000 cfm, or 4.6 cfm/ft^2. That's a lot of air!

Figure 1. Design sensible and latent loads



Humidity is the Driver.

In this example, 83,000 cfm is required to handle humidity, but only 18,000 cfm of this must be outdoor air for ventilation (assuming 15 cfm of OA per person*). With a space sensible cooling load of 546,000 Btu/hr and a supply-air temperature of 55°F; approximately 25,000 cfm is required to maintain the space temperature at 75°F. In this case, 72% of the supply air must be outdoor air. While this is a high fraction of outdoor air, it is manageable. Ventilation air is not the culprit.

This 25,000 cfm of supply air equates to 1.4 cfm/ft^2. This is a large, but manageable supply air quantity. But we still need 83,000 cfm of supply air to control humidity. How do we better equip the supply air to handle the high latent load associated with this many people? Obviously the supply air needs to be drier. The drier the supply air (the lower the dew point), the more water vapor it will remove from the space. What supply air dew point is required to handle the space latent load?

*While many local codes may still require 15 cfm/person for ventilation, the most recent version of ASHRAE Standard 62.1-2004 has revised the minimum required ventilation rates for places of assembly.

Calculating Specific Humidity. The key is another humidity measurement called specific humidity. Specific humidity is expressed as pounds of water vapor per pound of air. Suppose we choose to design the air distribution system for our example gymnasium for 25,000 cfm (114,000 pounds per hour). The 1200 people generate 227 pounds of water vapor each hour. Removing 227 pounds of water vapor with 114,000 pounds of air requires that the specific humidity of the supply air be 0.0020 lbw/lba drier than the space. The specific humidity at 75°F and 50 percent RH is 0.0092 lbw/lba. So the specific humidity of the supply air must be 0.0072 lbw/lba to offset the latent load of the people. This corresponds to a supply air dew point of about 48°F.

So how do we create this 48°F dew point supply air? One common method is to cool all the supply air to a dry-bulb temperature of about 49°F to 50°F. This should dehumidify the supply air to the 48°F dew point required to offset the latent load due to people.

Supplying 50°F air to the gymnasium provides additional benefits. It reduces the required airflow needed to offset the space sensible cooling load from 25,000 cfm to only 20,000 cfm (1.1 cfm/ft^2). This concept is called cold air distribution, and is a common design approach when aggressive humidity control is required or the design team is seeking ways to reduce fan power or air handler footprint.[1] All of these benefits may be attractive when designing for places of assembly. Cold air also requires careful diffuser selection, careful temperature control, and reliable control of building pressure. In addition, supply-air temperatures below 50°F may preclude the use of conventional, direct expansion (DX) equipment.

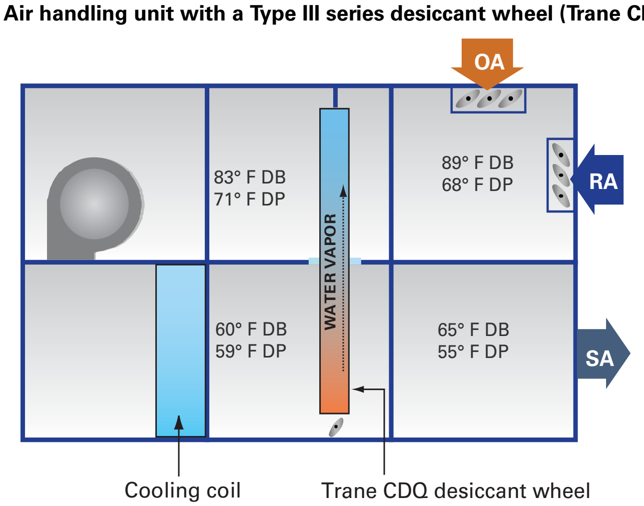
The Desiccant Approach. But do we need colder air, or do we need drier air? The truth is, we don't need air that is colder; we only need air that is drier. Recent research in desiccants has resulted in a Type III desiccant wheel that is able to regenerate at low temperatures, often without the need to add heat. This allows the wheel to be configured in series with a cooling coil. This activated alumina desiccant wheel is available in a Trane system called CDQ™ (Cool, Dry, Quiet).[2] The addition of the CDQ wheel allows the system to deliver supply air at 48°F dew point, while the cooling coil only needs to cool the air to 54°F. With CDQ there is no need to design a cold air distribution system. Since there is no need to produce 50°F supply air, the required capacity of the cooling load is substantially reduced.

With the cold air system, the required cooling coil capacity is about 150 tons (based on 1200 people and 18,000 cfm of outdoor air) and supply fan power is only 10 kW. The CDQ system reduces cooling coil capacity to about 140 tons, but increases fan power to 16 kW because of the higher airflow and additional static pressure from the desiccant wheel. Both are viable options. It is noteworthy that CDQ may be an excellent means to achieve low supply air dew points with conventional DX equipment.

Don't Forget Part Load Situations.

Places of assembly often experience very diverse loads. It would be wise to evaluate the performance of these

Figure 2. Air handling unit with a Type III series desiccant wheel (Trane CDQ)



systems at part load. There are two part-load conditions we should evaluate. One is quite obvious, which is what happens when most of the people leave. Perhaps the remaining occupancy is only 40 people instead of 1200. This is an easy part-load condition to accommodate. The sensible loads drop to 256,000 Btu/hr and the latent load due to people drops to only 8000 Btu/hr. The resulting space sensible heat ratio increases to 0.97.

If we supply air at 50°F with the cold air system, the required supply airflow is only 9400 cfm. This system is called "single zone VAV." Supply airflow is reduced to match the reduced sensible cooling load in the space. Single zone VAV is easy to control. The supply fan airflow is modulated based on space temperature. The 9400 cfm of 50°F air will remove the 256,000 Btu/hr of sensible heat and has the potential to remove 116 pounds of water vapor. However, at this reduced occupancy, the people add only 7.6 pounds of water vapor. The result is that space humidity is lowered to 40 percent RH. At this condition, the supply-air temperature could be reset upward to save some compressor energy.

Problem with Constant Volume Systems. What happens if the cold air system is a constant volume design rather than VAV? The reduced sensible cooling load requires a warmer supply-air temperature, about 63°F for this example. At this supply-air temperature, the 20,000 cfm of supply air will remove 256,000 Btu/hr of sensible heat, but less than 7.6 pounds of water vapor. Space humidity rises to 65 percent RH, well above our target of 50 percent. Not only does a constant volume system use more fan energy at part load, but it is less adept at removing moisture. By comparison, a single zone VAV system reduces fan energy while adequately removing moisture. Single zone VAV with cold air provides humidity control at most load conditions, while simultaneously saving fan energy.

How does CDQ fare with reduced occupancy? If the supply fan delivers a constant volume of air, the reduced sensible load requires the supply-air temperature to increase to over 65°F. However, the CDQ desiccant wheel can still deliver the supply air at 55°F dew point (Figure 2). The resulting space humidity rises to only 52 percent RH. Constant volume CDQ is certainly adept at controlling space RH at lower occupancy, but the benefits of VAV can be applied to CDQ systems too.

When Sensible Loads are Lighter. There is another part-load condition that can be even more sinister; reduced building-related sensible loads while the space is fully occupied. What happens with full occupancy (1200 people) when there is envelope or glass loads? If the only loads in the space are due to lighting and people, the sensible heat ratio drops to 0.63. If we dim the lights, the situation gets even worse.

This reduction in the space sensible cooling load creates a sensible heat ratio more severe than what the system was originally designed to accommodate. Increasing the supply-air temperature or reducing supply airflow in response to the reduced sensible load will hinder the ability to remove moisture. Reheat can help when the sensible heat ratio is lower than design. Both cold air and CDQ have the ability to reduce sensible cooling capacity while maintaining a lower supply air dew point. Table 1 compares these systems, with and without reheat, when all envelope conduction and solar loads are absent.

Both cold air and CDQ systems perform well at this part-load condition. Single zone VAV results in a slightly elevated space relative humidity, but still well within the comfort zone. This comfortable condition is achieved without reheat and uses less fan energy. Some reheat and additional fan energy may be needed if more precise humidity control is desired.

Each Situation Unique. Well, it was a great concert, but this was a high school concert band, not a rock band. Add smoke from a pyrotechnic display, or moisture from an Olympic size swimming pool, and designing a comfort system for "places of assembly" can be even more challenging. In addition to reheat, cold air distribution and the CDQ desiccant wheel, give us additional tools to deal with high space latent loads. Simple airside control schemes like single zone VAV provide an easy means of adapting to diverse part-load conditions while providing some energy savings.

Table 1. System comparison at part load (no envelope conduction or solar loads)

	Cold Air VAV	Cold Air CV	CDQ VAV	CDQ CV
Supply airflow (cfm)	15,600	20,000	19,500	25,000
Supply air drybulb (°F)	50°	55.5°	55°	59.4°
Supply air dew point (°F)	48.5°	47°	48°	49°
Space relative humidity (%)	56	50	53	50
Reheat (MBh)	0	152	0	123

Article by Don Eppelheimer, applications engineer,Trane. You can find this and previous issues of the *Engineers Newsletter* at www.trane.com/engineersnewsletter. To comment, e-mail comfort@trane.com.

[1] A 2000 Engineers Newsletter (volume 29-2, "Cold Air makes Good $ense") provides more detail on the benefits and design issues related cold air distribution systems.

[2] A 2005 Engineers Newsletter (volume 34-4, "Advances in Desiccant-Based Dehumidification") provides more detail on the series configuration of a Type III desiccant wheel (Trane CDQ).

Trane
A business of American Standard Companies
www.trane.com

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