Designing HVAC For Humid vs. Arid Climates

BY ERIC DESPLINTER, P.E., ASSOCIATE MEMBER ASHRAE; JEFF BOLDT, P.E., HBDP, FELLOW ASHRAE

Most mechanical engineers design primarily in one climate zone. They know the intricacies of that climate but can overlook important differences when designing in a climate with which they are not familiar. Whether your project climate is humid or arid, hot or cold impacts how you design an HVAC system.

Eric started his career in St. Louis, Mo., in ASHRAE Climate Zone 4A (Mixed-Humid) with summer 0.4% design temperatures of 96.2°F dry bulb/76.7°F wet bulb (35.7°C DB/24.8°C WB) and a winter 99.6% design temperature of 7.5°F (–13.6°C) DB. As the company expanded across the nation, he transferred to our Ontario, Calif., office. Ontario is a starkly different climate, as it is in ASHRAE Climate Zone 3B (Dry) with summer 0.4% design temperatures of 100.2°F DB/69.7°F WB (37.9°C DB/20.9°C WB) and a winter 99.6% design temperature of 38.4°F (3.6°C). How does designing an HVAC system differ in these two climates?

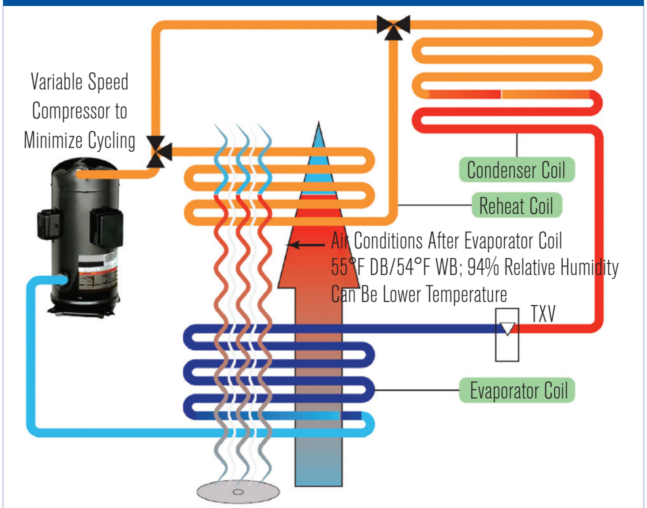
When designing in a humid climate such as St. Louis, it is critical to consistently dehumidify to keep occupiable spaces from becoming too humid and minimize the risk of mold. This is commonly done with a modulating chilled water valve, but using a modulating chilled water valve can be more difficult when designing a direct expansion (DX) system. In DX systems, using a modulating compressor is a good way to provide capacity control and ensure your cooling coil stays active so your system dehumidifies.

Cycling compressors off lets water on the cooling coil evaporate into the airstream and increases humidity. If variable-speed compressors are not an option, providing a multistage compressor is better than a single-stage compressor. The on/off limited controllability of nonmodulating DX systems can result in the system satisfying space temperature and the compressors turning off before enough moisture is removed from the air. Therefore, with DX systems, it is very important to not add a large safety factor to your load calculation, which would oversize your system and satisfy space temperature faster, turn off the compressor and introduce air to spaces that have not been dehumidified.

When a space is in a low-load condition, a single-stage DX coil cycles off after quickly satisfying the space demand. Once the compressor turns off, the ability to dehumidify is lost. Water also reevaporates off the cooling coil before it reaches the drain pan, which then requires more dehumidification. And you cannot just command the compressor to stay on, because the coil will frost without enough load on the system. Fortunately, you can use several methods beyond modulating or multistage compressors that provide improved dehumidification capabilities.

Eric DeSplinter, P.E., was an operations manager, and Jeff Boldt, P.E., is a managing principal at IMEG Corp. Boldt is SSPC 90.1 mechanical subcommittee chair.

FIGURE 1 Hot gas reheat. Reheating the air allows the cooling coil to stay active and continue to dehumidify without overcooling the space.



COURTESY GREENHECK AND BRIJESH PANCHAL (IMEG)

FIGURE 2 Wrap-around heat pipe. This is an effective way of removing more moisture than a standard system without using any extra energy or moving parts.

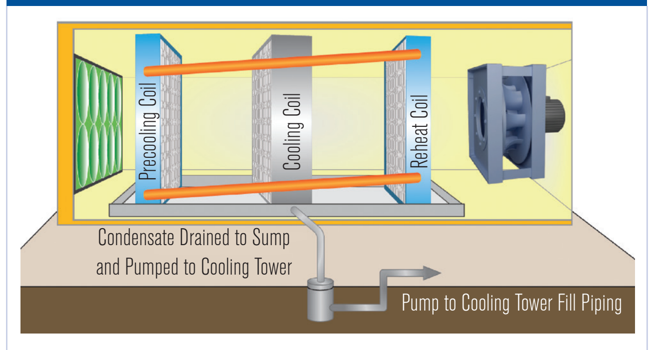


IMAGE COURTESY OF HEATPIPE.COM, ARMIN HAUER (EBM-PAPST) AND BRIJESH PANCHAL (IMEG CORP.)

FIGURE 3 Dual wheel system. This system operates on similar principles to the wrap-around heat pipe.

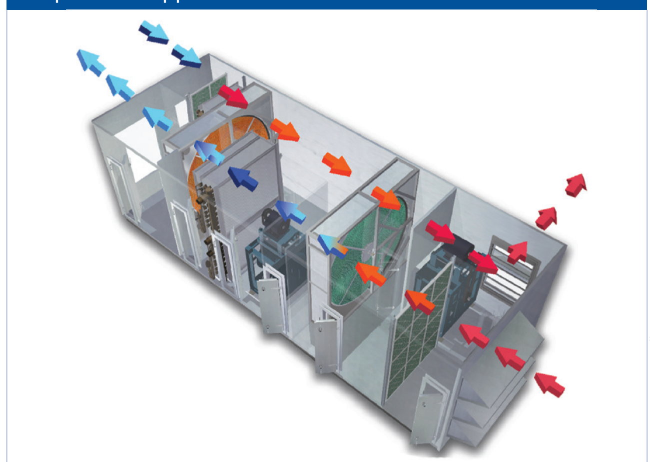


IMAGE COURTESY OF FLÄKTGROUP SEMCO

Methods to Improve Dehumidification Capabilities

The cheapest method to improve dehumidification for DX systems is to add an electric reheat coil that is controlled based on space relative humidity (RH) or dew point (DP). This forces the compressor to run. However, it is not energy efficient. (Think of it as a car speed control system in which you run the engine fully and modulate the brakes to control speed.) This method is not recommended unless you are in a climate in which dehumidification is very rarely needed. Another method, hot gas bypass, is similarly inefficient and is restricted or prohibited in many situations by energy codes such as ASHRAE Standard 90.1 and the International Energy Conservation Code (IECC).

On the other hand, one option we often use is hot gas reheat (*Figure 1*). This option diverts some or all hot refrigerant vapor from the condensing unit to a reheat coil. Hot gas reheat can be modulating or nonmodulating. Reheating the air allows the cooling coil to stay active and continue to dehumidify without overcooling the space.

Another option is a wrap-around heat pipe (*Figure 2*). One coil is located upstream of the cooling coil and another downstream. The coils are connected by refrigerant tubing. When hot air flows over the upstream heat pipe coil, the refrigerant inside the coil evaporates and flows to the downstream heat pipe coil. The upstream coil precools the entering air, reducing the load on the cooling coil. When the cooled and dehumidified air exits the cooling coil, the downstream heat pipe coil reheats the air with the warm refrigerant from the upstream coil. The refrigerant inside the heat pipe condenses and flows back to the upstream coil by gravity flow. Normally, no pump is needed. This is an effective way of removing more moisture than a standard system without using any extra energy or moving parts. A variation of this is to add an outdoor air enthalpy recovery device.

A third option is a dual wheel system (*Figure 3*), which operates on similar principles to the wrap-around heat pipe. It consists of a sensible heat wheel that exchanges heat between the return or exhaust air and the supply air. This provides reheat to the dehumidified supply air and precools the return air. It also has an enthalpy wheel that uses the cooled return or exhaust air to precool the outdoor air, which allows the cooling coil to save energy when cooling the supply air.

When Is it Appropriate to Use One of These Methods?

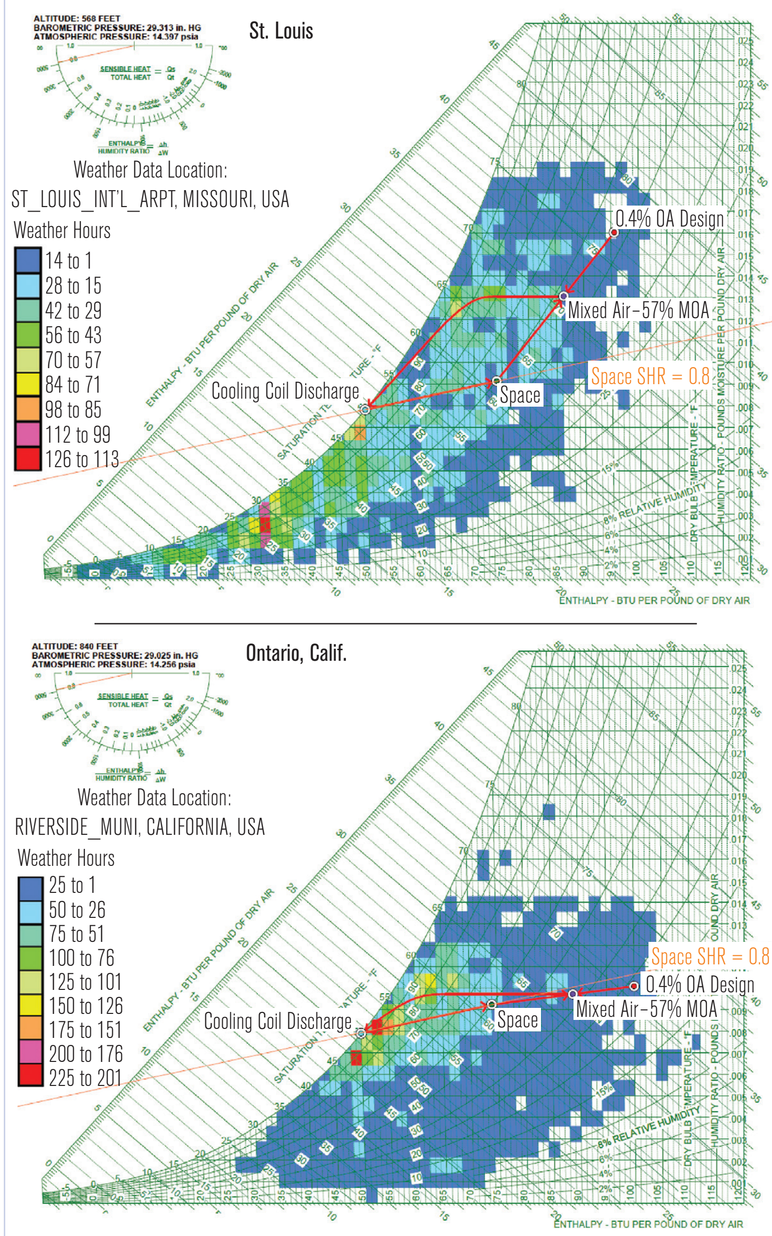
Many resources describe these methods and others in more detail, but how do you know if you should use one of the part-load capacity control systems for your project? You do not want to unnecessarily add cost to your project if extra dehumidification measures are not required, but you also do not want to cut corners and leave them out when they really are needed to maintain proper space conditions.

ASHRAE Standard 62.1 has for many years limited the maximum indoor dew point to 60°F (16°C). The most recent edition of Standard 62.1 also adds a dew-point limit. Generally, 30% to 60% indoor relative humidity is considered ideal for occupant comfort. Some specialty spaces will require more strict relative humidity control.

Knowing when it is appropriate to use an extra dehumidification strategy is a tricky judgment call. You need to assess how critical it is to maintain certain space conditions. An office building owner may have a higher tolerance for having high space humidity for a few hours on a rare sticky summer day than would the owner of a museum storing rare paintings. Both building owners will have a very low tolerance if those sticky summer days occur all summer long every year.

Since we care about how often we would expect the space conditions to be out of tolerance, it is valuable to review the bin data for the project location. Doing so allows you to determine how many hours per year you expect the system to maintain design space conditions. Viewing the bin data on a psychrometric chart can help you visualize your climate conditions.

FIGURE 4 Weather bin data for Saint Louis (Top) and Ontario, Calif. (Bottom).



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Looking at *Figure 4* you can see that there are far more hours above a 60°F (16°C) dew point in St. Louis (top) than there are in Ontario, Calif. (bottom). To determine if your equipment can meet your dehumidification needs, you can calculate the coil sensible heat ratio (SHR) and compare it to the capacity of your proposed cooling coil. If your proposed coil cannot satisfy the calculated latent load on more days than you are willing to accept, consider an additional dehumidification method.

Let's look at a 30 ft × 30 ft (9 m × 9 m) high school classroom served by a standard single-zone packaged rooftop unit (RTU) located in St. Louis and one located in Ontario, Calif., and assess how the dehumidification needs differ. In both cases we use the 0.4% design outside air conditions. Keeping all aspects of the load calculations the same (envelope properties, ventilation rate, internal setpoints, etc.) except for the location and associated outside air conditions, we get these summer mixed-air conditions:

- Saint Louis, Mo.: 86.3°F DB/70.1°F WB/62.4°F DP (30.2°C DB/21.2°C WB/16.9°C DP)
- Ontario, Calif.: 86.9°F DB/66.7°F WB/55.5°F DP (30.5°C DB/19.3°C WB/13.1°C DP)

Yes, Missouri has different ventilation requirements than California, but we are keeping them the same, so we are only analyzing climate impact on the results.

The calculated coil SHR is 0.60 and 0.79 for St. Louis and Ontario, respectively. In reviewing the coil performance for a packaged RTU at a standard 400 cfm/ton (54.7 L/s·kW), the coil SHR is about 0.78 for the St. Louis conditions and 0.94 for the Ontario conditions. Those capacities are not very close to meeting the required coil SHR based on the load calculation. But if you lower the airflow to 300 cfm/ton (40.3 L/s·kW), your latent capacity increases, and the coil SHRs become about 0.71 for St. Louis and 0.83 for Ontario, respectively.

The coil capacities are different for the two locations because they depend on the entering air conditions and ambient temperature. With the St. Louis wet bulb entering air conditions being about 3.5°F (1.9°C) higher than Ontario, the St. Louis coil has more latent capacity. So, at 300 cfm/ton (40.3 L/s·kW), the Ontario coil is pretty close to meeting the calculated load, while the St. Louis coil still is not.

When we look, the number of hours the Ontario coil would not provide the calculated latent capacity are so few that the space does not have time to get out of control in terms of humidity. But there are sustained hours in St. Louis in which the coil does not have adequate latent capacity, which will result in an unacceptably high relative humidity classroom environment.

The example above only describes one specific situation. It is not intended to conclude that humidity control is never warranted in arid climate classrooms. For example, shortly after arriving in Ontario, I was asked to assess some science classrooms that were experiencing high humidity. Teachers in the science classrooms were reporting moisture on countertops and papers curling from the high relative humidity. When I went to the site and took space readings, I found that some of the spaces were as high as 70°F (21°C) DB and 78% RH. These higher humidity conditions only occur for a few days per year, but they do occur.

Looking at the classroom from our previous example and analyzing the SHR at the 0.4% design dehumidification conditions, we get a much different story than in St. Louis. In Ontario, the coil SHR would be 0.50, which is far below the previously determined coil SHR of 0.83. The science classrooms I had been asked to assess were served by standard packaged rooftop units with single-stage compressors. Since they were science classrooms, they were bringing in a higher amount of outdoor air than a standard classroom, which exacerbated the humidity issues on those few days a year they do have high outside dew point.

A science classroom located in Ontario could have a coil SHR as low as 0.44 on the design dehumidification day. The higher outdoor air percentages of the science classrooms increased the number of hours the coil could not meet the required latent load, which resulted in high humidity levels in the classrooms.

If you analyze the system performance throughout the entire year, you can make an educated decision about the best path forward for your project based on how many hours per year you expect those more extreme conditions and whether your design will comply with the applicable editions of Standards 55 and 62.1.

FIGURE 5 Window with condensation and frost.

IMAGE COURTESY OF IMEG CORP.

What About Humidification Needs?

Like the impact of the climate on dehumidification needs, the climate is also important to consider when you humidify a building. Certain buildings such as hospitals and museums require minimum humidity levels inside. When you humidify the building, the dew point increases. This can be especially problematic in cold climates, as it can lead to condensation on glass if not properly mitigated (*Figure 5*).

In a climate like St. Louis, the summers are hot and humid, while the winters are cold and have low dew-point temperatures. A future column will provide a more in-depth analysis of HVAC design in cold vs. hot climates. Below is an example contrasting design in different climates.

A few years ago, we were designing St. Louis's Museum of Westward Expansion. It is the museum located beneath the Gateway Arch. We had to humidify the museum in the winter to maintain high enough humidity levels to prevent the museum displays from drying out and degrading.

The architect had designed an entrance that consisted of glass walls and glass roof (*Figure 6*). The glass was a laminated monolithic structural blast-resistant glass. But this had the trade-off of having subpar thermal performance. The design winter outside air temperature in St. Louis was 7.5°F (–13.6°C). We wanted to maintain inside conditions of 72°F (22°C) at 50% RH to protect the museum displays. That is a dew point of 52.4°F (11.3°C). We chose to be conservative and assess the glass surface temperature with an outside air temperature of 2°F (–17°C).

FIGURE 6 Museum of Westward Expansion.

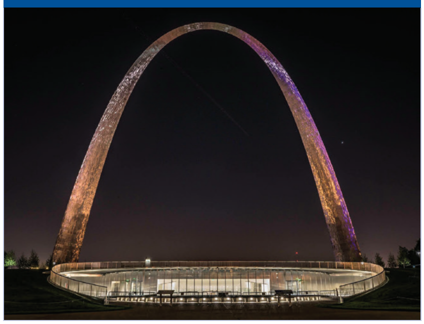

IMAGE COURTESY OF IMEG CORP.

We found that the glass inside surface temperature would be below 30°F (–1.1°C). This was obviously a problem since our dew point was 52.4°F (11.3°C). We determined that condensate would form on the glass when the outside air temperature dropped below 20°F (–6.7°C). There are a lot of hours below 20°F (–6.7°C) each year in St. Louis. Condensation and frost would form and obstruct the view. Condensation could also drip on visitors from the glass roof. We had to find a way to mitigate the condensation and frost.

After assessing several options, including infrared heaters and electric heat wire embedded in the glass, we decided on a two-prong approach. First, we could lower the inside dew point by placing the most sensitive artifacts inside sealed enclosures with their climate controlled by microclimate generators. This allowed us to lower the RH in the museum in the winter. Now we would maintain a minimum inside RH of 36% at 72°F (22°C). That lowered the dew point to 43.6°F (6.4°C). Then we chose to deliver hot air to the glass surface via floor grilles.

Upon CFD analysis we found that the glass inside surface temperature would be raised enough to prevent condensation in all but the most extreme weather conditions. We had mitigated the risk of condensation enough that the project team was willing to accept the minimal risk.

If this same building were located in Ontario, we would not have needed to provide hot air at the glass or add the climate-controlled enclosures because of Ontario's winter design temperature of 38.4°F (3.6°C).

Summary

Designers should pay close attention to a project's climate. The amount of moisture in the air depends on the project location and time of year. Designers must consider outside conditions during all seasons and their impact on the operation of the building and HVAC systems. What is appropriate in one climate does not necessarily make sense in other climates. Especially for firms that share work between offices in different climate zones, it is important to educate employees about this.

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