Dry climate evaporative cooling with refrigeration backup

This unique design concept offers energy savings, improved indoor air quality and a low first-cost per cfm

By C. Mike Scofield, P.E., and Elia Sterling
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Is it possible to provide an industrial owner with an energy efficient evaporative cooling system when his building cannot tolerate the supply air temperature drift if left to the total mercy of the summer ambient wet bulb temperature? This was the question facing the consultant engineer on an electronics plant design for a facility located in Colorado Springs, Colorado.[1]

The weather data for this 6,000 ft (1,829 m) elevation plant site shows a summer design condition of 91°F (33°C) dry bulb and 58°F (14°C) wet bulb. It predicts that 50% of the average summer hours (June through September) will have outdoor wet bulb temperatures low enough to generate a 55°F (13°C) supply air temperature *without mechanical cooling*.

During a typical year, there are some *2,729 hours* with mean coincident wet bulb temperatures *lower than 54°F* (12°C) when the dry bulb temperature ranges between 55° to 75°F (13° to 24°C; see *Figure 1*).

About the authors

C. Mike Scofield is a senior consultant for CFM Company, Englewood, Corlorado. He received his degrees in mechanical engineering and business administration from the University of Colorado-Boulder. Scofield has served as president of both the San Jose, California, and Denver, Colorado, chapters of ASHRAE and as a member of the Society Nominating Committee.

Elia M. Sterling is the president of Theodor D. Sterling and Associates Ltd., Vancouver, British Columbia, Canada. He received his bachelors degree from Washington University-St. Louis and his architectural degree from the University of British Columbia. Sterling is chairman of ASHRAE TG 4.CCD (Cold Climate Design), vice-chairman of TC 9.9 (Building Commissioning) and vice-chairman of GPC-1R (HVAC Commissioning Process). Sterling has also served as president of the British Columbia Chapter.

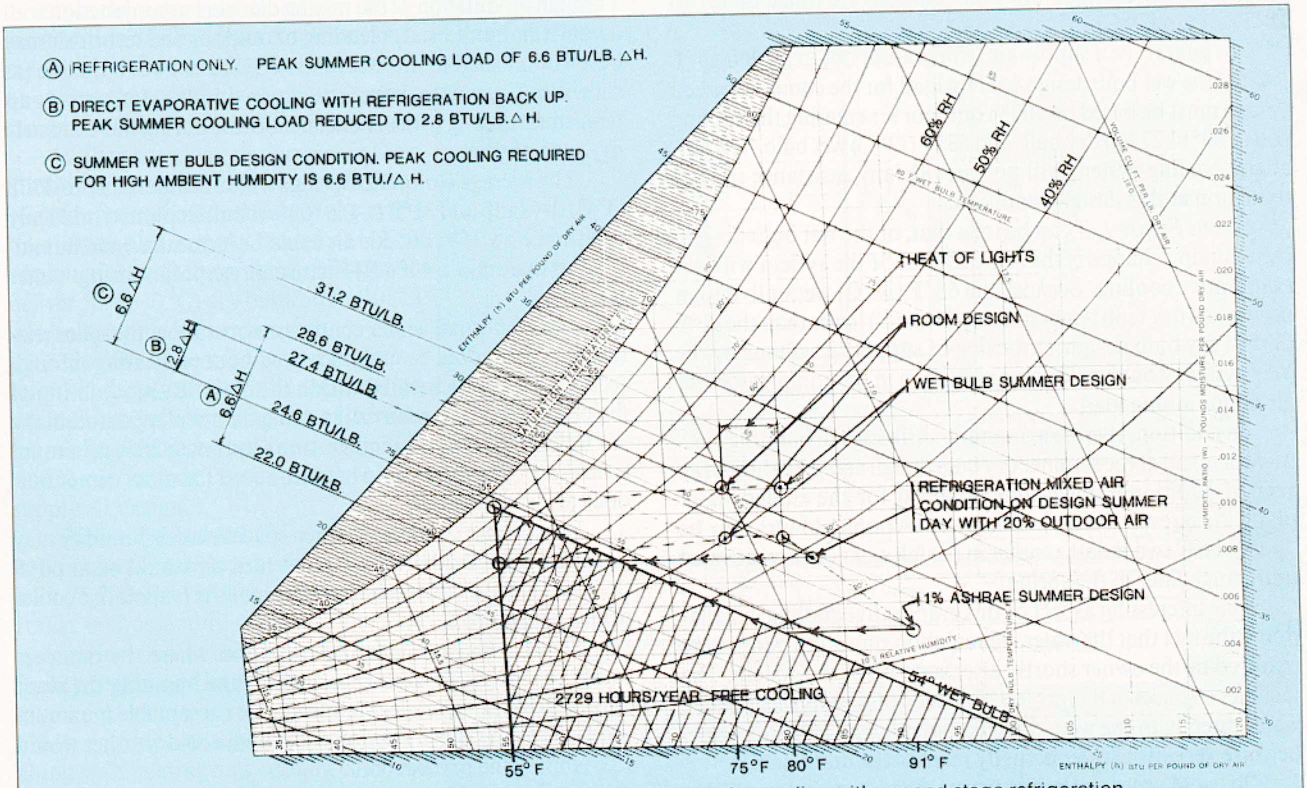

Figure 1. Summer cooling process, refrigeration versus direct evaporative cooling with second stage refrigeration.

Normally, vapor compression air conditioning would be required to meet the 55 °F (13 °C) supply air design temperature in this envelope of hours. However, a 96% saturation efficiency evaporative cooling system could be used to keep the chiller off the line in this dry climate.

The consultant engineer's design coupled the economy of evaporative cooling with the insurance of compressorized air conditioning to guarantee a 55 °F (13 °C) supply air condition for the plant's 300,000 cfm (141,600 L/s) variable air volume (VAV) system. With a wetted media type evaporative cooler as the first stage of cooling, the refrigeration system would be locked out until the supply air temperature started to rise above the required 55 °F (13 °C) setpoint.

During the cooling cycle, with outside air dry bulb temperatures above 55 °F (13 °C), the economizer dampers and evaporative cooling system would operate with 100% outdoor air and 100% return air exhausted. This cooling mode provides a higher degree of indoor air quality (IAQ) while exhausting the heat generated by lights, people and the return air fan that otherwise would be part of the cooling load.

Figure 1 shows the economy of combining a chilled water dry cooler with an adiabatic cooling device in dry climates. Compared to a typical mechanical cooling system using 80% return air and 20% outdoor air, this two-stage cooling technique would reduce the building peak summer demand tonnage by 57%.

Design considerations

A word of caution to aspiring wet bulb wizards. There are, even in dry climates, design wet bulb weather conditions that must be examined. For example, in Colorado Springs, the 1% (30 hours per year) wet bulb design condition of 63 °F (17 °C) is most likely to occur during a dry bulb temperature range of 75° to 80 °F (24° to 27 °C).

To guarantee a supply air temperature not to exceed 55 °F (13 °C), the wet bulb design cooling load for the compressorized system must be based on 100% outdoor air entering the cooling coil at 80 °F (27 °C) dry bulb and 63 °F (17 °C) wet bulb. The adiabatic cooling system will not provide any assistance in load reduction at this design condition.

From *Figure 1*, it can be seen that, at the wet bulb design, the enthalpy change is the same as that of the system without evaporative cooling. Because the 63 °F (17 °C) wet bulb design occurs at a dry bulb temperature 11 °F (6 °C) lower than the 91 °F (33 °C) dry bulb design, throttling of supply air volume by the VAV system would yield some reduction in the compressorized air-conditioning load.

In addition, there is a dramatic shrinkage in building electrical demand at the summer dry bulb design and 2,729 hours per year of additional down-time obtained for the chilled water plant. As a result, refrigeration system cooling towers may be "put to bed" two months earlier in the fall and not be reactivated until much later in the spring.

One interesting aspect of this evaporative cooling installation is the fact that the water recirculation sump and pump were removed by the owner shortly after start-up of the system. The facilities engineer at this project explained that supplying potable water directly to the wetted media did not lead to water waste, because very little water actually reaches the drain pan.

To avoid water waste, a moisture sensor may be installed in the drain from the evaporative cooler to close the supply water valve. The elimination of standing water in a sump greatly reduces any opportunity for breeding harmful bacteria or fungi. In cold weather climates, potential sump freeze-up problems are also avoided.

By separately metering the supply water to a once-through evaporative cooling system, the owner may often avoid the sewer fee portion of his water bill. Waste lines from the evaporative coolers must be connected to storm drains rather than the sanitary sewer system.

Before specifying a once-through evaporative cooling system, the design engineer should carefully evaluate the quality of supply water in his area. This design concept should not be pursued in a hard water area or the life of the wetted media could be short. A periodic, full flow, water flush cycle should be incorporated to purge dissolved solids that accumulate on the wetted surface.

Pay special attention to the provision of good room air motion in the occupied zones supplied by these two-stage cooling systems. In arid climates, the higher room relative humidity levels produced by the direct evaporative cooling component will lead to comfort complaints by the "desert rats" who are accustomed to the normal 20% to 30% RH levels produced by mechanical cooling systems.

This is especially true when the VAV system is turned down to meet reduced cooling loads. The perception of high humidity increases as VAV systems throttle back and room air motion diminishes.

Evaporative coolers as humidifiers

With the placement of the wetted media evaporative cooler downstream of the outside air and return air mixing plenum, it can be employed for humidification in the winter months. Through modulation of the mixing dampers in conjunction with a return air humidistat, blending of outdoor and return air may be accomplished to intersect the 54 °F (12 °C) wet bulb line (as shown in *Figure 2*) to deliver the required 55 °F (13 °C) supply air temperature along with beneficial moisture during this extremely dry time of year.

The average Colorado Springs winter temperature of 37 °F (3 °C) dry bulb and 31 °F (−0.6 °C) wet bulb is plotted on *Figure 2* to show how 35% outside air could be introduced and humidified to maintain a 42% RH return air setpoint during winter operation.

During normal winter continuous operation, the space relative humidity could be maintained without preheating through recovery of heat generated within the plant. By modulation of the outdoor air and return air mixing dampers to maintain the 55 °F (13 °C) supply air temperature setpoint, a 20% minimum outdoor air quantity could be introduced for most winter outdoor air temperatures.

For start-up conditions, when space relative humidity may be in the 10% to 20% range and return air would be at 60 °F (16 °C), a preheat coil located upstream of the evaporative cooler would be required.

Figure 2 shows a start-up condition where the dampers would be set for 100% return air to heat and humidify the plant during the warm-up cycle. Maintenance of acceptable minimum ventilation rates during periods of extreme cold weather would also employ the preheat coil.

At ambient temperatures below 32 °F (0 °C), an indoor humidity level reset to 30% to 35% RH is best for cold climates

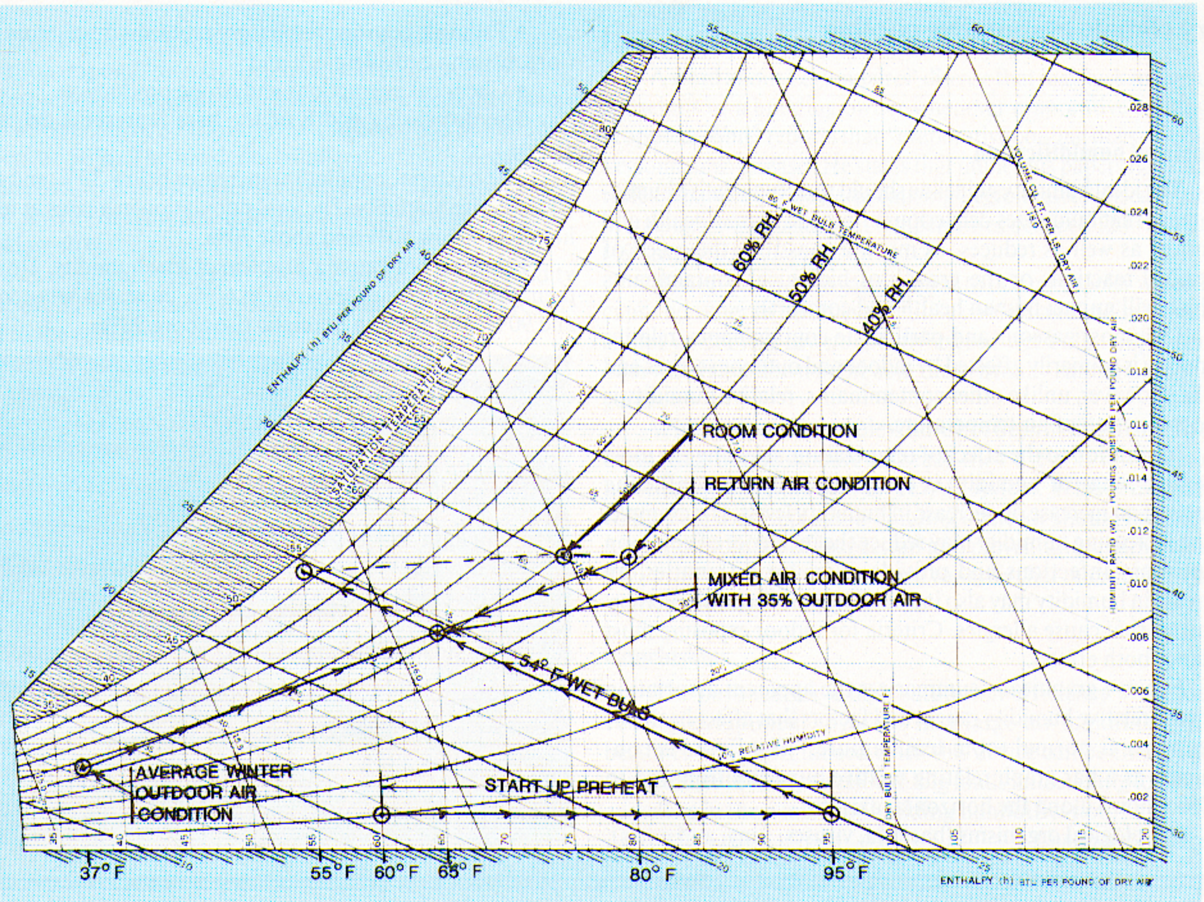

Figure 2. Direct evaporative cooler used for humidification in winter.

to reduce the risk of structural and building moisture damage from condensation.[2] A lower indoor humidity level would require that the supply air wet bulb temperature be reduced to 50°F (10°C) or lower.

With the outdoor air dampers in their minimum position, this can be accomplished through modulation of the evaporative cooler supply water valve in conjunction with the preheat coil valve to deliver 55°F (13°C) dry bulb/50°F (10°C) wet bulb supply air to the space.

The VAV system is tolerant to fluctuations in supply air dry bulb temperature caused by the efficiency inertia of the wetted media evaporative cooling system. This would not be true in a constant volume supply air design.

Figure 3 shows the physical size and component arrangement for a factory-built 75,000 cfm (35,400 L/s) evaporative cooler coil assembly similar to the four systems installed at the Colorado Springs facility. Significant installed cost and space savings can be realized when the evaporative cooler can be pre-assembled along with heating and cooling coils and shipped in one piece from the manufacturing plant.

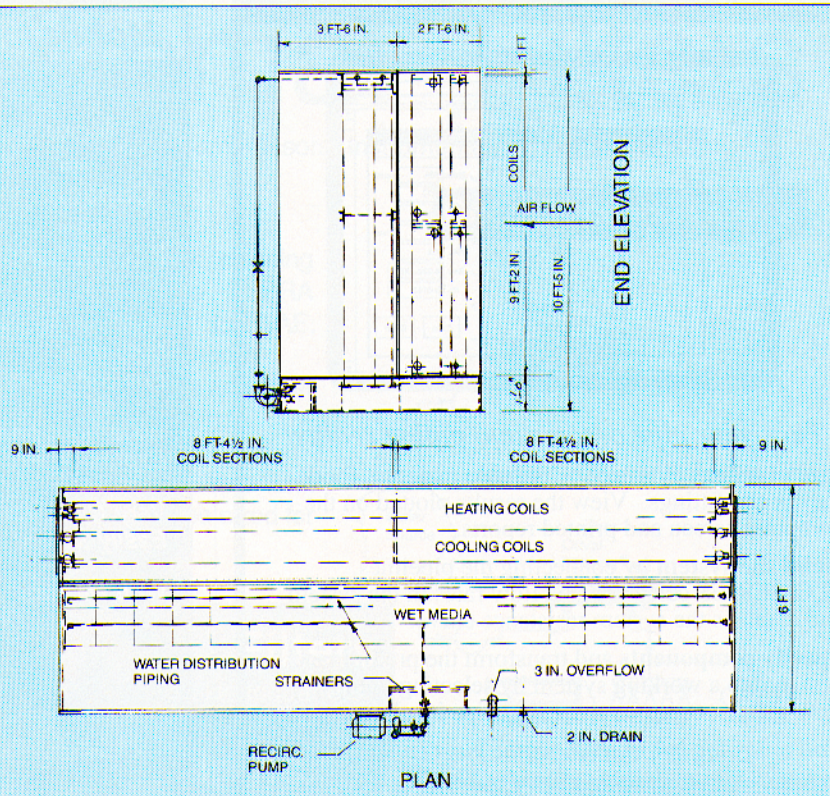

Figure 3. Evaporative cooler coil assembly.

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Beneficial humidification

The ideal humidity guideline should specify a relative humidity range that minimizes deleterious effects on human health and comfort as well as reduces, as much as possible, the speed of chemical reactions or the growth of biological contaminants (which will impact human health and comfort).

Like most gaseous and particulate contaminants, relative humidity is primarily affected by indoor and outdoor sources and sinks. However, unlike other contaminants, relative humidity is also a function of air temperature.

In addition to the effect of temperature, selecting the most desirable range of humidity is complicated by the conflicting effects of an increase or decrease in humidity levels. For example, while increasing humidity may reduce the incidence of common respiratory infections and provide relief for asthmatics, an increase in humidity may also increase the prevalence of microorganisms that cause allergies. Criteria for indoor exposure must balance both effects.

Figure 4 graphically summarizes the apparent association between relative humidity ranges and factors that affect health of occupants at normal room temperature.[3] The figure is constructed as a bar graph relating relative humidity from 0% to 100% (shown along the horizontal axis) to:

- Biological organisms (bacteria, viruses, fungi and mites);
- Pathogens causing respiratory problems (respiratory infections, asthma and allergies); and
- Chemical interactions and ozone production.

The decreasing width of the bars represents decreasing effects.

The bacterial population increases below 30% and above 60% relative humidity. The viral population increases at relative humidity below 50% and above 70%.

Fungi do not cause a problem at low humidity. However, their growth becomes apparent at 60%, increases between 80% and 90%, and shows a dramatic rise above 90%.

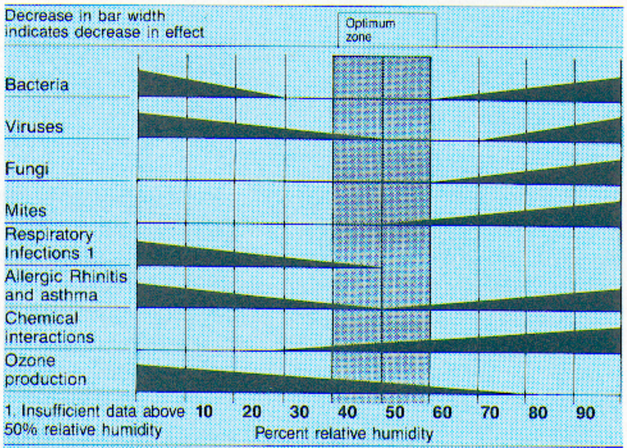

Figure 4. Optimum relative humidity ranges for health.

Mites require humidity for survival. Growth in the mite population responds directly to humidity levels in excess of 50%.

Respiratory infections increase at relative humidity below 40%. However, there is little information on effects of humidity in excess of 50%. The incidence of allergic rhinitis because of exposure to allergens increases at relative humidities above 60% and the severity of asthmatic reactions increases at relative humidities below 40%.

Most chemical interactions increase as the relative humidity rises above 30% although ozone production is inversely proportional to the relative humidity.

The evidence suggests that the optimal conditions to enhance human health by minimizing the growth of biological organisms and the speed of chemical interactions occur in the narrow range between 40% and 60% relative humidity at normal room temperature. That narrow range is represented by the optimum zone in the shaded region of the graph.

Although keeping indoor humidity levels within this region will minimize health problems, there is probably no level of humidity at which some biological or chemical factor that affects health negatively does not flourish.

ASHRAE standards have long provided guidance for engineers on controlling humidity to achieve comfortable conditions. *ASHRAE Standard 62-1989*, recognizing that the provisions of optimal relative humidity must be tempered by requirements for ventilation and energy efficiency, recommends maintaining levels between 30% and 60% relative humidity in habitable spaces and below 70% in low velocity ducts and plenums.[4]

Legionnaires' disease

A word about Legionnaires' disease that may lurk in the subconscious of a building owner whenever evaporative cooling is proposed. Watt[5] reports that "evaporative cooling does not carry Legionnaires' disease." He states that numerous papers have made it clear that the key bacteria causing Legionnaires' disease breed in stagnant water usually at temperatures between 120° to 140°F (49° to 60°C).

He also states, "No cases appear to occur in cooler water. Because no water in direct evaporative coolers and indirect system sumps or cooling towers greatly exceed wet bulb temperatures even when shut down in the sun, they virtually cannot harbor the germs."

While evaporative coolers have not been associated with the occurrence of Legionnaires' disease, their use does introduce the presence of moisture into an HVAC system. When evaporative coolers are used, a prudent building operator should implement a regular program of inspection and preventative maintenance. This program should be designed to verify that microbial growth is not occurring.

Conclusions

The payback for a high saturation efficiency evaporative cooling component installed in a large air handling system becomes very short indeed when this device can be employed in the summer to reduce peak cooling loads and cooling ton-hours as well as in the winter for building humidification.

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At a first-cost of approximately 0.30 dollars per cfm, a high efficiency wetted media type evaporative cooler could easily be the most cost effective component installed in an air-conditioning system for dry climates. The static pressure penalty to air flow at 500 fpm (2.5 m/s) face velocity is only 0.3 in. wg and the fractional pump horsepower penalty is small in comparison to the large cooling effect produced.

The health and environmental benefits of humidification in occupied areas for arid climates is of great importance. Not many design concepts offer both *energy savings* coupled with *improved indoor air quality*.

Acknowledgments

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