Efficient Space Humidity Control With Active Chilled Beam Systems

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Over the past few years, the use of chilled beams and radiant chilled ceilings has increased rapidly in North America due to the space and energy savings inherent to air-water systems. Although these systems have been widely used in Europe for more than 20 years, North American climates and other building design practices often result in higher space loads and moisture removal requirements than their European counterparts generally encounter.

Space humidity control is essential for the proper application of any HVAC system used in humid climates, but it is particularly critical for these air-water systems. These devices rely largely on their integral tempered chilled water coils (chilled water delivery is maintained at, or above, the room dew-point temperature) for the removal of much of the space sensible heat gains without condensation. Maintenance of space dew-point temperatures at or below the entering chilled water temperature is critical to prevent condensation.

As with most HVAC systems, moisture is removed from a mixture of outdoor and recirculated room air at the central air-handling unit (AHU) coil prior to the mixture's delivery through the distribution ductwork. This process sufficiently lowers the primary air dew-point temperature to allow its removal of all latent heat gains within the space including those moisture gains related to building infiltration. All of the system moisture removal occurs at the central air-handling unit.

Although outdoor humidity has some effect on space latent loads, the effect is minimized in tightly constructed buildings with proper building pressure control. The *2009 ASHRAE Handbook—Fundamentals*[1] classifies building tightness by the leakage rate that results from pressurizing the building to 0.3 in. w.c. (75 Pa). Leakage rates less than 0.1 cfm/ft^2 (0.05 L/s·m^2) of outside wall area are generally considered representative of "tight" buildings. Buildings of "average" construction exhibit leakage rates of around 0.3 cfm/ft^2 (0.14 L/s·m^2) of exterior wall area while leakage rates exceeding this are considered leaky and most likely unsuitable for the

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application of air-water systems. In any case, infiltration usually comprises only a small amount of the total building latent load, and its contribution is offset by the amount of dehumidification provided by the AHU coil.

Active Chilled Beam Systems

Chilled beam systems allow primary airflow rates to be reduced by up to 70%[2] when compared to all-air systems as the majority of the space sensible cooling is accomplished via the beams' conditioning of room air within their integral cooling coils.

Figure 1 illustrates the operation of an active chilled beam. Preconditioned primary air ① ducted to the beam from the air-handling unit is injected through a series of nozzles ②. The relatively high velocity air jets induce room air ③ through the beam's face and across a chilled water coil ④ before mixing it with the primary air and discharging it ⑤ to the room. As the coil is intended to remain dry, the chilled water supply temperature should be maintained at or above the room dew-point temperature. The primary air delivered to the beam is thus the sole source of room dehumidification. As such, it is advisable that design space latent gains are also considered when establishing the space primary airflow rate. In fact, the space primary airflow rate will be established as the maximum required for sensible (CFM_{SENS}) and latent (CFM_{LAT}) design cooling of the space or that which is required to preserve the mandated space ventilation rate (CFM_{VENT}).

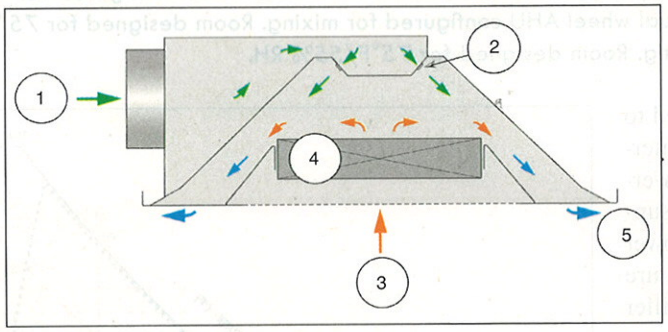

Figure 1: Active chilled beam in operation.

Active chilled beams are typically supplied a fixed volume flow rate of primary air while a zone chilled water valve varies the chilled water flow to their integral coils in response to the space cooling demand, resulting in a constant volume, variable temperature delivery of supply air to the space.

Space Conditioning in Air-Water Systems

To maximize the performance of chilled beam systems in interior spaces, the room to primary air dew-point temperature differential should allow the primary airflow rate for space humidity control (CFM_{LAT}), to closely match the (CFM_{VENT}) required for space ventilation. The ventilation rate for an office space with one occupant per 120 ft^2 (11 m^2) is 11 cfm (5.2 L/s) per person.[3] The latent gain (q_{LAT}) of a person is about 200 Btu/h (59 W) per person. If room design conditions are specified as 75°F (24°C) and 50% RH, the room humidity ratio (W_{ROOM}) is 65 grains (4.2 g). Air-handling units with chilled water coils typically supply air to the space at 55°F (13°C) and a humidity ratio (W_{PA}) of 54 grains (3.5 g). The airflow rate (CFM_{LAT}) required to maintain W_{ROOM} can be calculated as:

$$CFM_{LAT} = q_{LAT} / [0.69 \times (W_{ROOM} - W_{PA})]$$

In this case, the latent airflow requirement is 0.23 cfm/ft^2 (1.16 L/s·m^2) or 23 cfm (13 L/s). Interior area sensible cooling loads are commonly 10 to 12 Btu/h·ft^2 (30 to 38 W/m^2). Under the aforementioned room and primary air conditions, primary airflow rates of 0.18 to 0.23 cfm/ft^2 (0.9 to 1.1 L/s·m^2) would be required for a beam whose chilled water coil provides 60% of its sensible cooling.

Perimeter area cooling loads are often 40–50 Btu/h·ft^2 (125–160 W/m^2). In these areas the primary airflow must both drive the induction function and provide an adequate contribution toward the space sensible cooling. For a space sensible load of 40 Btu/h·ft^2, the beam (whose primary air is supplied 20°F (11°C) below the room temperature and whose chilled water provides 60% of its sensible heat removal) would require a primary airflow rate (CFM_{SENS}) of 0.74 cfm/ft^2 (3.7 L/s·m^2), which is far greater than the space ventilation or latent airflow requirement. Chilled water temperature control and/or flow discontinuation should be used in applications where the perimeter space latent airflow requirement may exceed those for sensible cooling. This includes areas with operable windows and leaky buildings.

Desiccant Moisture Removal

Air-handling units with desiccant moisture removal can be used to produce lower primary air dew-point temperatures to compensate for the primary airflow reduction inherent to air-water systems. *Figure 2* (Scenario B1) illustrates an air-handling unit, which incorporates total energy recovery and a passive desiccant wheel. Outside air transfers heat and moisture to the exhaust airstream by means of the total energy wheel. The entering air is then cooled to saturation at the AHU cooling coil before passing through the passive desiccant wheel. This wheel removes water vapor from the cool air, by converting latent heat in the primary airstream to sensible. Primary air then leaves the unit at an increased dry-bulb temperature of about 64°F (18°C), but a low dew-point temperature of around 47°F (8°C), which corresponds to a humidity ratio of 48 grains (3.1 g).

Balancing Latent Loads and Ventilation

Although the supply of low dew-point primary air in chilled beam systems has obvious advantages, it may also have drawbacks. The cost of a desiccant air-handling unit as described here is considerably higher than that of conventional equipment. In addition, desiccant based dehumidification results in the delivery of primary air at higher dry-bulb temperatures, 68°F to 75°F (20°C to 24°C). This reduces the primary air sensible cooling contribution, increasing the

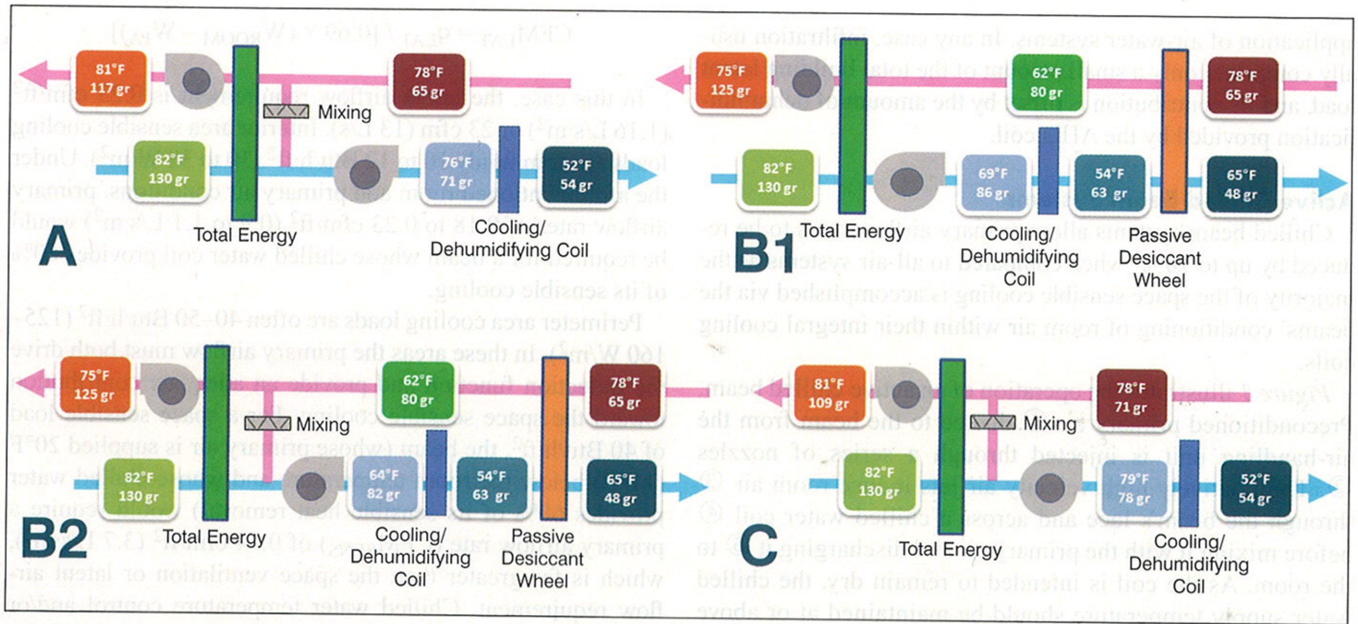

Figure 2: Air-handling units used for various design example scenarios. **Scenario A:** Single wheel AHU configured for mixing. Room designed for 75°F/50% RH. **Scenario B1:** Dual wheel AHU configured for 100% OA. Room designed for 75°F/50% RH. **Scenario B2:** Dual wheel AHU configured for mixing. Room designed for 75°F/50% RH. **Scenario C:** Single wheel AHU configured for mixing. Room designed for 75°F/55% RH.

linear footage of beams required to accommodate the increased water-side cooling requirements. Lowering the primary chilled water supply temperature to achieve a lower primary air dew-point temperature may also have an impact on chiller operating costs.

Raising the Design Dew Point

An alternative to lowering the primary air dew point could involve raising the design dew-point temperature of the space to reduce the latent cooling primary airflow requirements.

Figure 5.2.1.1 of ASHRAE Standard 55-2010, *Thermal Environmental Conditions for Human Occupancy* (shown as *Figure 3*) defines the range of space operative and dew-point temperatures that result in acceptable levels of occupant satisfaction. In this figure, space dew-point temperatures as high as 62°F (17°C) with a dry-bulb temperature of 75°F (24°C) are deemed acceptable.

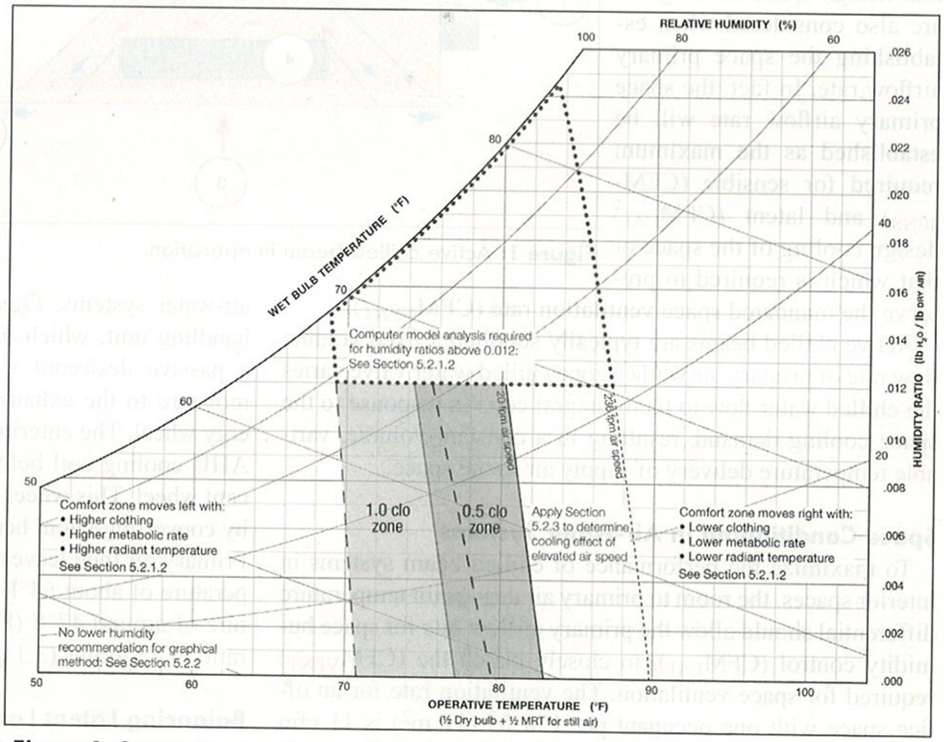

Figure 3: Summer and winter comfort zone chart.[4]

It is not suggested that such high space dew points be considered with chilled beam systems as it would require unacceptably high chilled water temperatures. However, maintaining space dew-point temperatures around 57°F (14°C) fits well within the design range recommended in *Figure 3* and still allows the use of chilled water supply temperatures in the range of 57°F to 58°F (14°C to 14.5°C).

Chilled water coils provide primary air dew-point temperatures in the range of 50°F to 52°F (10°C to 11°C). This coupled with a "relaxed" interior space dew-point temperature of 57°F (14°C) will often allow space latent loads to be removed by a primary airflow rate similar to the space ventilation air requirement.

Design Example

An active chilled beam system is being used in a 100,000 ft^2 (9290 m^2) office building in Atlanta where design dry-bulb/MCWB temperatures are 92°F/74°F (33°C/23°C) and design wet-bulb/MCDB temperatures are 85°F/77°F (31°C/26°C). The room design conditions are 75°F (24°C) with a relative humidity not to exceed 55%.

Tables 1 and *2* define the building occupancy as well as its sensible load design parameters. All latent infiltration gains are assigned to the perimeter areas and are based on an infiltration rate of 0.2 cfm/ft^2 (0.09 L/s·m^2) of façade. Ventilation rates are calculated according to ASHRAE Standard 62.1-2010.

Design Sensible Load and Ventilation Requirements (For 100,000 ft^2 Building in Atlanta)	Perimeter Offices (27,000 ft^2)	Perimeter Conference Rooms (3,000 ft^2)	Interior Offices (63,000 ft^2)	Interior Conference Rooms (7,000 ft^2)
Average Zone Size (ft^2)	900	450	2,000	450
Number of Occupants	6	20	20	20
Occupant Density (ft^2/person)	150	22.5	100	22.5
Design Sensible Cooling Load (Btu/h·ft^2)	40.0	52.5	11.0	23.5
Design Latent Cooling (Btu/h·ft^2)	1.3	8.9	2.0	8.9
Occupant OA Rate (R_P) (cfm/person)	5	5	5	5
Area OA Rate (R_A) (cfm/ft^2)	0.06	0.06	0.06	0.06
Zone Ventilation Effectiveness (E_Z)	0.90			
Occupant Diversity Factor (D)	0.70			

Table 1: Description of building and loads used in examples.

HVAC System Design Conditions	
Sensible Outdoor Design Conditions	
Dry-Bulb Temperature (°F)	92
MCWB Temperature (°F)	74
Humidity Ratio (grains)	97
Specific Volume (ft^3/lb_m)	14.21
Latent Outdoor Design Conditions	
Wet-Bulb Temperature (°F)	77
MCDB Temperature (°F)	82
Humidity Ratio (grains)	130
Specific Volume (ft^3/lb_m)	14.06
Chilled Beam Performance Characteristics	
Maximum Primary Airflow Rate (cfm/linear ft)	15
Coil Sensible Heat Removal (Btu/h·cfm) (At a chilled water flow rate of 1.0 gpm and EAT– MWT = 16°F)	34

Note: All air-handling units are provided with total energy recovery wheels with 83% sensible efficiency, 68% total efficiency.

Table 2: Design conditions and chilled beam characteristics.

Chilled beams with a water-side sensible cooling rate of 34 Btu/h·cfm (6 Wh·m^3) of primary air at an entering air to mean water temperature differential of 16°F (9°C) are used in all four scenarios. In each case, the chilled water supply temperature has been established at 1°F (0.6°C) above the room dew-point temperature and the sensible cooling performance of the coil has been adjusted accordingly.

Table 3 details four system scenarios that will be studied. All scenarios are based on local design wet-bulb conditions of 82°F (28°C) dry bulb and 77°F (25°C) wet bulb. The air-handling unit for each scenario is illustrated in *Figure 2*. All air-handling units are equipped with total energy wheels with 83% sensible and 68% total efficiency.

Scenario A is the baseline, which includes a conventional air-handling unit with a chilled water cooling and dehumidification coil. A primary air mixture of outdoor and return air conditioned to a 50°F (10°C) dew point is delivered at 55°F (13°C) to spaces maintained at 75°F (24°C) with design relative humidity levels of 50%.

Scenario B1 assumes a dedicated outdoor air system (DOAS) air-handling unit with a passive desiccant wheel supplying 100% outside air to the building.

Scenario B2 uses the same air-handling unit but delivers a primary air mixture of return and outdoor air. Both scenarios B1 and B2 involve delivery of primary air to the room at 68°F (20°C) dry bulb and a 47°F (8.3°C) dew-point temperature to spaces designed for 75°F (24°C) and 50% relative humidity.

Scenario C uses the same air-handling unit as Scenario A, but the space relative humidity design has been relaxed to 55% (design temperature remains the same).

Table 4 allows comparison of the primary and outside airflow rates and water flow rates required for space ventilation, cooling and dehumidification under each design scenario. It also estimates the energy use of the system at the local design wet-bulb conditions.

The upper section of *Table 4* summarizes the various zone airflow requirements for each scenario. In all cases, the mini-

Design Parameters for Example Building	Scenario A Baseline (Mixing at AHU)	Scenario B1 Reduced Primary Air Dew Point (100% OA)	Scenario B2 Reduced Primary Air Dew Point (Mixing at AHU)	Scenario C Increased Design Room Dew Point (Mixing at AHU)
Room Design Conditions				
Design Dry-Bulb Temperature (°F)	75.0	75.0	75.0	75.0
Design Relative Humidity (%)	50.0	50.0	50.0	54.6
Design Humidity Ratio (grains)	65	65	65	71
Design Space Dew Point Temperature (°F)	55.1	55.1	55.1	57.6
Primary Air Design Conditions				
Dry-Bulb Temperature Leaving AHU (°F)	52.0	65.0	65.0	52.0
Dry-Bulb Temperature Entering Room (°F)	55.0	68.0	68.0	55.0
Design Humidity Ratio (grains)	54	48	48	54
Percent of Primary That is Outside Air	28.3%	100.0%	26.7%	35.0%
Beam Performance Parameters				
Chilled Water Supply Temperature (°F)	56.0	56.0	56.0	58.5
Coil Sensible Cooling (Btu/h·cfm)	34.0	34.0	34.0	29.5
Primary Air Sensible Cooling (Btu/h·cfm)	21.7	7.6	7.6	21.7
Total Sensible Heat Removal (Btu/h·cfm)	55.7	41.6	41.6	51.2

Table 3: Design parameters for comparison example.

mum zone primary airflow rate will provide the required outdoor airflow and sensible cooling to the space at design sensible load conditions while maintaining the space relative humidity level. The values denoted in bold represent the (sensible, latent cooling or ventilation) function that determines the airflow requirement. The middle sections of the table summarize the air handler cooling capacities, mixed/outside air ratios, and the number of feet of chilled beams required in each scenario. They also identify the chilled water system capacity requirements.

Finally, the lower section of *Table 4* summarizes the HVAC system energy requirements of the various scenarios of design conditions.

Discussion of Results

Scenarios B1 and B2 require about 8% more primary air than the baseline (Scenario A) due to their elevated primary air temperature. Coupled with the pressure loss associated with the desiccant wheel, the fan energy is about 35% greater than that of the baseline Scenario A.

By comparison, the primary air requirement and fan energy use of Scenario C is 10% less than the baseline due to its relaxed space relative humidity requirement. This is attributable to the reduced latent cooling requirements associated with the higher design humidity levels in the conference and interior office areas. The airflow requirements in perimeter office areas is slightly higher than that of the baseline scenario due to the higher chilled water supply temperature associated with the relaxed humidity design of Scenario C.

Water-side cooling capacity comparisons in *Table 4* indicate that AHU cooling coil requirements for each scenario are basically the same when the air handler delivers a mixture of outdoor and return air. Scenario B1 exhibits higher AHU cooling requirements due to its delivery of 100% outside air.

The total length of chilled beams required under Scenarios B1 and B2 is almost 25% greater than that required by the base scenario because the warm primary air results in the beams' water coils having to provide a far greater contribution toward the space sensible cooling. The length of the beams required in Scenario C is also about 10% greater than the baseline due to the increased chilled water supply temperature used in the relaxed humidity strategy.

In conclusion, the total energy requirement for the baseline (Scenario A) is actually lower than any of the other scenarios studied. Although the primary airflow requirement exceeds that of Scenario C, its chiller and pumping energy is less. The

Airflow Calculations at Design Conditions						
		Perimeter Office Areas	Perimeter Conference Areas	Interior Office Areas	Interior Conference Areas	Weighted Average
Primary Airflow to Provide q_{SENS} CFM_{SENS}/ft^2	Scenario A	0.72	0.92	0.20	0.40	0.38
	Scenarios B1 & B2	0.97	1.23	0.27	0.53	0.51
	Scenario C	0.79	1.00	0.22	0.43	0.41
Primary Airflow to Provide q_{LAT} CFM_{LAT}/ft^2	Scenario A	0.18	1.17	0.27	1.17	0.34
	Scenarios B1 & B2	0.12	0.76	0.18	0.76	0.22
	Scenario C	0.12	0.76	0.18	0.76	0.22
Zone Outdoor Air Rate (V_{OZ}) cfm/ft^2	All Scenarios	0.10	0.31	0.12	0.31	0.14
Total Zone Primary airflow (V_{PZ}) cfm/ft^2	Scenario A	0.72	1.17	0.27	1.17	0.48
	Scenarios B1 & B2	0.97	1.23	0.27	0.76	0.52
	Scenario C	0.79	1.00	0.22	0.76	0.43

	Design Scenario	System Ventilation Efficiency (E_{VZ})	Minimum Outdoor Air (V_{OT})	Total Primary Airflow Rate	Mixing Ratio at AHU (% OA)	Beams Required (Linear Feet)
Primary Air & Chilled Beam Requirements Coincident With Zone Design Conditions	Scenario A	0.76	13,619 cfm	48,183 cfm	28.3%	3,174 ft
	Scenario B1	1.00	52,117 cfm	52,117 cfm	100.0%	3,932 ft
	Scenario B2	0.75	13,915 cfm	52,117 cfm	26.7%	3,932 ft
	Scenario C	0.68	15,196 cfm	43,420 cfm	35.0%	3,516 ft

	Design Scenario	AHU Cooling & Dehumidification	Beam Water Side Cooling	Total System Cooling Dehumidification	Beam System Water Flow Rate[a]	AHU Coil Chilled Water Flow Rate[b]
Water Side Cooling/ Dehumidification Requirements Coincident With Zone Design Conditions	Scenario A	160 tons	88 tons	249 tons	354 gpm	321 gpm
	Scenario B1	162 tons	140 tons	302 tons	561 gpm	324 gpm
	Scenario B2	126 tons	140 tons	267 tons	561 gpm	252 gpm
	Scenario C	170 tons	95 tons	265 tons	380 gpm	340 gpm

	Design Scenario	Fan Energy[c]	Total Chilled Water Flow Rate	Pump Energy[d]	Chiller Energy[e]	Total Energy Requirement
Energy Requirements At Specified Design Conditions for Various Scenarios	Scenario A	48.5 kW	674 gpm	14.6 kW	149 kW	212 kW
	Scenario B1	65.5 kW	886 gpm	19.2 kW	182 kW	266 kW
	Scenario B2	65.5 kW	814 gpm	17.6 kW	160 kW	243 kW
	Scenario C	43.7 kW	720 gpm	15.6 kW	159 kW	218 kW

a. Chilled beam system water flow rates are based on an average 6°F (3.3°C) temperature difference between entering and leaving chilled water.
b. AHU coil water flow rates are based on an average 12°F (6.6°C) temperature difference between entering and leaving chilled water.
c. Fan energy calculation assumes the following fan static pressures:
 –Single wheel AHU (Scenarios A and C): Supply fan static pressure of 3.5 in. w.g. (872 Pa), exhaust fan static pressure of 2.5 in. w.g. (623 Pa)
 –Dual wheel AHU (Scenarios B1 and B2): Supply fan static pressure of 4.25 in. w.g. (1058 Pa), exhaust fan static pressure of 3.25 in. w.g. (809 Pa)
d. Pump energy calculation assumes 60 ft of head pressure (179 kPa) for all scenarios.
e. Chiller energy calculation assumes efficiency of 0.6 kW/ton (0.17 kW/kW) for all scenarios.

Table 4: Capacity comparisons for example scenarios.

chiller energy reduction is due to a reduction in the outside airflow rate attributable to its higher system ventilation efficiency. The pumping energy reduction is due to this and a lower beam water flow rate (coincident with its lower beam chilled water supply temperature).

Summary

Although low dew-point primary air temperatures are often associated with chilled beam systems, they are often not a necessity. An alternative is to relax space design humidity levels, which may offer savings in fan and chiller energy, as well as reducing the required number of beams. The design consultant is urged to weigh the cost/benefit of the various solutions on a project-by-project basis, understanding the humidity relationships between air-water systems and space latent control.

A slight risk of overcooling may exist in perimeter offices and conference areas where the space sensible and/or latent loads vary significantly. The use of a warmer primary air temperature will reduce the reheat requirements as the chilled beam coil provides a greater share of the space sensible cooling, but again this increases the number of beams required. Perimeter office areas might be better served by using a variable air volume terminal with a sensible heating coil to feed the beams. The heating coil also could be used to warm the primary air during periods requiring perimeter heating, eliminating the need for four-pipe beams and the associated costs of piping hot water to every beam. Conference areas with wide swings in occupancy might be more efficiently served by variable air volume terminals, in lieu of active beams.

Applications in humid climates, such as classrooms and assembly areas, where space sensible heat gain ratios are below 0.8, may be better served by the use of some type of secondary moisture removal. In all cases, the designer should be aware of the impact of delivering warmer primary air and attempt to research alternate remedies before making a decision regarding the air-handling unit configuration.

References

1. *2009 ASHRAE Handbook—Fundamentals.*
2. REHVA. 2004. *Chilled Beam Application Guidebook.* Federation of European Heating and Air-conditioning Associates, Brussels.
3. ANSI/ASHRAE Standard 62.1-2010, *Ventilation for Acceptable Indoor Air Quality.*
4. ANSI/ASHRAE Standard 55-2010, *Thermal Environmental Conditions for Human Occupancy.* ■