engineers newsletter

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understanding
Chilled Beam Systems

Although chilled beam systems have been used in Europe and Australia for many years, they are a new concept to many in the U.S. Those interested in learning more about these systems, as with any new concept, are faced with the task of discerning its true strengths and weaknesses. The goal of this EN is to investigate the common claims about chilled beam systems.

Overview of Chilled Beam Systems

Passive chilled beams (PCB). A PCB consists of a fin-and-tube heat exchanger, contained in a housing (or casing), that is suspended from the ceiling (Figure 1). Chilled water passes through the tubes. Warm air from the space rises toward the ceiling, and the air surrounding the chilled beam is cooled, causing it to descend back toward the floor, creating convective air motion to cool the space. This allows a passive chilled beam to provide space cooling without the use of a fan.

Active chilled beams (ACB). An ACB also consists of a fin-and-tube heat exchanger contained in a housing that is suspended from, or recessed in, the ceiling (Figure 1). The primary difference is that an active chilled beam contains an integral air supply. This primary air passes through nozzles, which induce air from the space up through the cooling coil. This induction process allows an active chilled beam to provide much more cooling capacity than a passive chilled beam. For this reason, active chilled beams are more commonly used, and are the focus of this EN.

Figure 2 includes examples of active chilled beams. They have pipe connections and a primary air connection. Note that either two- or four-pipe designs are available. With the two-pipe design, all zones receive either cold water or hot water. The benefit of the four-pipe design is that some zones can receive cold water for space cooling, while other zones simultaneously receive hot water for space heating.

Primary air system. To comply with most building codes in the U.S., outdoor air must be supplied to each space for ventilation. And since a chilled beam (whether passive or active) typically does not contain a condensate drainage system, the primary air system must also maintain the dew point of the indoor air below the surface temperature of the chilled beam to avoid moisture from condensing on the coil and dripping into the space.

Therefore, the purpose of the primary air system for active chilled beams is to:

1) Deliver at least the required amount of outdoor air to each space for ventilation, ***and***

2) Deliver air which is dry enough to offset the space latent load and maintain the indoor dew point low enough to avoid condensation on the chilled beams, ***and***

3) Deliver enough air to induce sufficient room airflow to offset the space sensible cooling load.

Figure 1. Passive versus active chilled beams

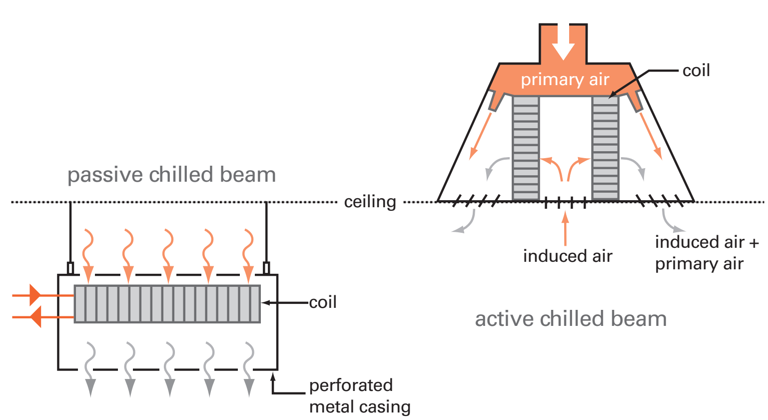


Figure 2. Examples of active chilled beams

Closer Investigation of Advantages

The goal of this section is to assess the common claims about active chilled beam systems.

Claimed advantage 1: An ACB system typically allows for smaller ductwork and smaller air-handling units than a variable-air-volume (VAV) system. Because active chilled beams provide some of the space cooling using induced airflow, primary airflow delivered by the central air-handling unit (AHU) is typically less than the design primary airflow delivered by the AHU in a conventional VAV system. Therefore, ACB systems typically allow for smaller ductwork and AHUs than a conventional VAV system.

Depending on how the ductwork is routed, smaller ducts can sometimes allow for shorter floor-to-floor heights. And because the primary AHUs will likely be smaller, a building with chilled beams may require less floor space for mechanical rooms.

The primary airflow delivered to an active chilled beam must be the *largest* of the following:

1) The minimum outdoor airflow required by ASHRAE Standard 62.1 or the local building code.[1]

2) The dry airflow needed to offset the indoor latent load and maintain the space dew point low enough to avoid condensation on the chilled beams. Of course, the quantity of air needed for this purpose depends on how dry that air is. If the primary air is delivered drier (at a lower dew point), then less airflow is needed.

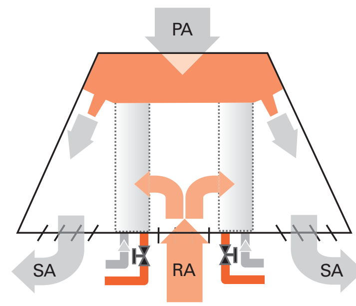
3) The primary airflow (PA) needed to induce sufficient room air (RA) to offset the design space sensible cooling load (Figure 3). If more primary air is delivered through the nozzles, more room air will be induced through the chilled beam coils, resulting in greater cooling capacity.

Table 1 demonstrates the impact of these three functions for an example office space. Using the default occupant density from ASHRAE 62.1-2007, the minimum outdoor airflow required for an office space is 0.085 cfm per square foot of floor area.

Based on typical occupant latent loads in an office space, if primary airflow is only 0.085 cfm/ft^2, it must be dehumidified to 47°F dew point to offset the space latent load and maintain the indoor dew point at 55°F (75°F dry bulb and 50 percent RH). However, if primary airflow is increased, the primary air would not need to be dehumidified as much (Table 1).

Finally, based on typical sensible cooling loads in an office space, catalog performance data from several manufacturers of active chilled beams indicates that between 0.35 and 0.40 cfm/ft^2 of primary air is required to provide the required sensible cooling capacity.

Figure 3. Primary airflow (PA) in an active chilled beam.



For this same example office space, the design airflow delivered by a conventional VAV system would be 0.90 cfm/ft^2. At design cooling conditions, primary airflow required for the ACBs serving this space is 60 percent less than for the conventional VAV system. However, this does not translate to a 60 percent reduction in fan energy use, as will be discussed later in this EN under the *claimed advantage 3* section.

As you can see from this example (Table 1), the primary airflow required for space sensible cooling in the ACB system is four times larger than the minimum outdoor airflow requirement

Table 1. Zone-level primary airflow for an example office space

Minimum outdoor airflow required (per ASHRAE 62.1-2007)[1]	0.085 cfm/ft^2 (for LEED® EQ credit,1.3 x 0.085 = 0.11 cfm/ft^2)
Active chilled-beam system	
Primary airflow required to offset space latent load[2]	0.085 cfm/ft^2 (DPT_{PA}=47°F or W_{PA}=47 gr/lb) 0.11 cfm/ft^2 (DPT_{PA}=49°F or W_{PA}=51 gr/lb) 0.36 cfm/ft^2 (DPT_{PA}=53°F or W_{PA}=60 gr/lb)
Primary airflow needed to induce sufficient room airflow to provide sensible cooling[3]	mfr A: 0.36 cfm/ft^2 (DBT_{PA} = 55°F) mfr B: 0.38 cfm/ft^2 (DBT_{PA} = 55°F) mfr C: 0.35 cfm/ft^2 (DBT_{PA} = 55°F)
Conventional VAV system	
Primary airflow needed to provide sensible cooling[4]	0.90 cfm/ft^2 (DBT_{PA} = 55°F)

[1] For an office space, Table 6-1 of ASHRAE Standard 62.1-2007, *Ventilation for Acceptable Indoor Air Quality*, requires 5 cfm/p (R_p) plus 0.06 cfm/ft^2 (R_a), and suggests a default occupant density of 5 people/1000 ft^2: V_{bz} = (5 cfm/person x 5 people/1000 ft^2) + 0.06 cfm/ft^2 = 0.085 cfm/ft^2.

[2] For moderately active office work, the 2009 *ASHRAE Handbook-Fundamentals* (Table 1, Chapter 18) suggests a latent load of 200 Btu/h/person. Using the same default occupant density (5 people/1000 ft^2): Q_{latent} = (200 Btu/h/person x 5 people/1000 ft^2) = 1.0 Btu/h/ft^2 = 0.69 x V_{PA} x (W_{space} - W_{PA}). Assuming a space dew point target of 55°F (64 gr/lb): V_{PA} = 1.0 Btu/h/ft^2 / (0.69 x (64 - 47 gr/lb) = 0.085 cfm/ft^2, V_{PA} = 1.0 Btu/h/ft^2 / (0.69 x (64 - 51 gr/lb) = 0.11 cfm/ft^2, or V_{PA} = 1.0 Btu/h/ft^2 / (0.69 x (64 - 60 gr/lb) = 0.36 cfm/ft^2.

[3] For an office space, the space sensible cooling load typically ranges from 17 to 24 Btu/h/ft^2 (which equates to about 0.8 to 1.1 cfm/ft^2 if a conventional VAV system is used). Assuming a space sensible cooling load of 19.5 Btu/h/ft^2, a zone cooling setpoint of 75°F, and a primary-air dry-bulb temperature of 55°F, product literature from manufacturer A indicates that four (4) 6-ft long, 4-pipe, 2-way discharge active chilled beams require 0.36 cfm/ft^2 to offset the design space sensible cooling load. With the same type of chilled beam, manufacturers B and C require about 0.38 and 0.35 cfm/ft^2 of primary air, respectively.

[4] Assuming the same space sensible cooling load of 19.5 Btu/h/ft^2, a zone cooling setpoint of 75°F, and a primary-air dry-bulb temperature of 55°F: V_{PA} = 19.5 Btu/h/ft^2 / (1.085 x (75 - 55°F) = 0.90 cfm/ft^2.

of 0.085 cfm/ft². Even if this project was being designed to achieve the "Increased Ventilation" credit of LEED 2009 (which requires 30 percent more outdoor air than required by ASHRAE 62.1-2007), the required outdoor airflow would still be much lower than the primary airflow required for space sensible cooling.

A survey of performance data from various chilled beam manufacturers indicates that the typical primary airflow rate for active chilled beams ranges from 0.30 to 0.70 cfm/ft². This is typically higher than the minimum outdoor airflow required by ASHRAE 62.1-2007 for many applications.

In this case, the primary AHU for an active chilled beam system must be designed to either a) bring in more than the minimum required amount of outdoor air—which will increase energy use in most climates—or b) mix the minimum required outdoor airflow with recirculated air to achieve the necessary primary airflow.

Claimed advantage 2: An ACB system can typically achieve relatively low sound levels. Chilled beams do not have fans or compressors located in (or near) the occupied space, so they have the opportunity to achieve low sound levels. Of course, most VAV systems can also be very quiet when designed and installed properly. Fan-powered VAV terminals do have fans located near the space, so they can be more challenging.

Claimed advantage 3: An ACB system uses significantly less energy than a VAV system, due to 1) significant fan energy savings—because of the reduced primary airflow—2) higher chiller efficiency—because of the warmer water temperature delivered to the chilled beams—and 3) avoiding reheat—because of the zone-level cooling coils.

Is there significant supply-fan energy savings? In some applications, a zone served by active chilled beams may require 60 to 70 percent less primary airflow, at design cooling conditions, than the same zone served by a conventional VAV system (0.36 cfm/ft² versus 0.90 cfm/ft² in the previous office space example). However, the difference in annual fan energy use is likely much less because the VAV system benefits from reduced zone airflow at part load, system load diversity, and unloading of the supply fan.

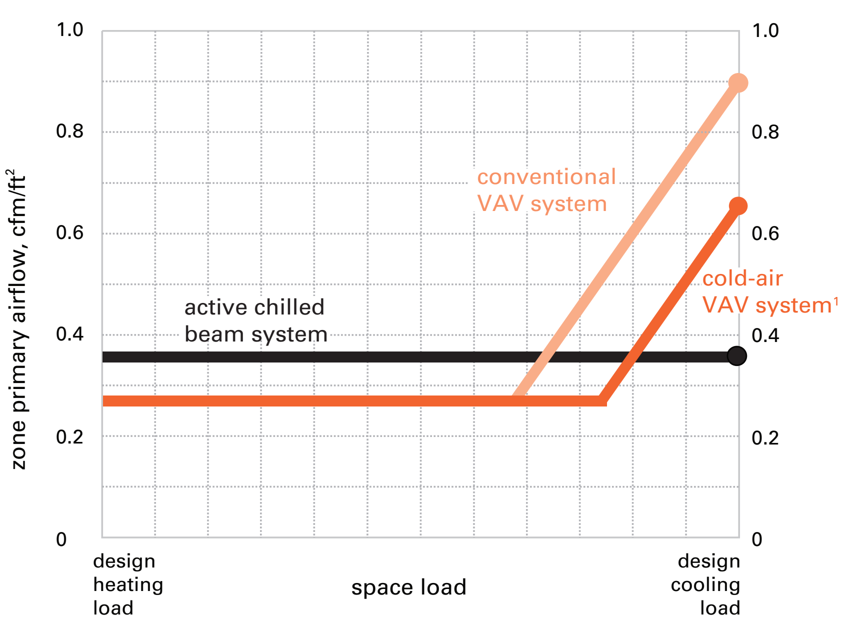
1) **VAV systems benefit from reduced zone airflow at part load.** An active chilled beam relies on primary airflow to induce room air through the coils inside the beam, so the quantity of primary air delivered to the chilled beams is typically constant (not variable). This means that, for this example, primary airflow is 0.36 cfm/ft² at all load conditions (Figure 4).

In a VAV system, however, primary airflow delivered to the zone is reduced at part load. Assuming a 30 percent minimum airflow setting for the VAV terminal, primary airflow to this example office space varies between 0.90 cfm/ft² at design cooling conditions and 0.27 cfm/ft² at minimum airflow (Figure 4).

If a cold-air VAV system (48°F primary air, rather than the conventional 55°F) is used, however, design airflow for this example office space is reduced to 0.67 cfm/ft², which shrinks the difference even further (Figure 4).

2) **VAV systems benefit from load diversity.** Because of load diversity, the central supply fan in a multiple-zone VAV system does not deliver 0.90 cfm/ft² on a building-wide basis. Assuming 80 percent system load diversity for this example, the supply fan only delivers 0.72 cfm/ft² (the "block" airflow), at design cooling conditions.

Figure 4. Zone primary airflow at part load



[1] For a cold air VAV system, assuming a design space sensible cooling load of 19.5 Btu/ft², primary air dry-bulb temperature of 48ºF, and zone cooling setpoint of 75ºF: 19.5 Btu/ft² = 1.085 x V_{PA} x (75ºF-48ºF), so V_{PA}= 0.67 cfm/ft².

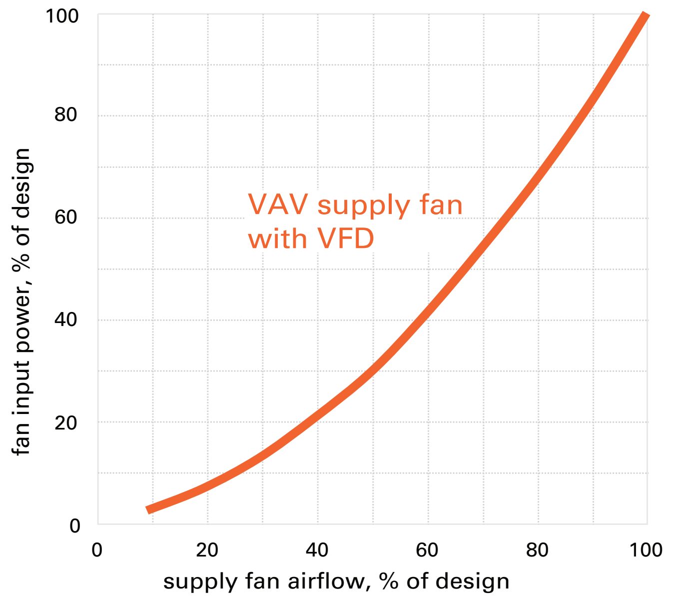
For an ACB system, primary airflow delivered to each zone is typically constant (capacity is adjusted by modulating, or cycling, water flow). Therefore, the fan in the centralized, primary AHU must deliver the sum of the zone primary airflows—the "sum-of-peaks" airflow, rather than the "block" airflow—which is 0.36 cfm/ft^2 for this example.

3) **VAV systems benefit from unloading of the supply fan at part load.** But reduced airflow (cfm) at part load is only part of the story. Fan energy depends on both airflow and pressure. In a VAV system, as the supply fan delivers less airflow, the pressure loss through the components of the air distribution system (ductwork, diffusers and grilles, air-handling unit, etc.) decreases. The result is that the fan power decreases exponentially (not linearly) as airflow is reduced. Figure 5 depicts the part-load performance of the supply fan in a typical VAV system, according to ASHRAE Standard 90.1.[2]

Using this performance curve, Table 2 and Figure 6 demonstrate how fan power decreases as the supply fan unloads for this office space example. Again, because of load diversity, the supply fan in the VAV system only delivers 0.72 cfm/ft^2 at design cooling conditions. For the ACB system, primary airflow (0.36 cfm/ft^2), and therefore fan power, remains constant at all load conditions.

Notice that as soon as the VAV supply fan unloads below 68 percent of design fan airflow, the conventional VAV system is actually using less fan energy than the constant-volume primary AHU fan in the ACB system (Figure 6). For a cold-air VAV system, this threshold increases to 80 percent of design fan airflow (Figure 7).

Figure 5. Part-load performance of a typical VAV system supply fan



Source: Table G3.1.3.15 ("Part-Load Performance for VAV Fan Systems"), Appendix G, ASHRAE Standard 90.1-2007

Note: Because the dampers in the VAV terminals modulate to reduce zone primary airflow at part load, the fan does not follow the "fan laws"-which would suggest that fan power drops off with the cube of airflow reduction-but the power still drops off exponentially, rather than linearly.

Table 2. Example part-load performance of a VAV system supply fan

supply fan airflow, % of design	supply fan airflow, cfm/ft^2	supply fan power[3], bhp/1000 ft^2
100%	0.72[1]	0.76[2]
90%	0.65	0.63
80%	0.58	0.51
70%	0.50	0.40
60%	0.43	0.31
50%	0.36	0.22
40%	0.29	0.15
30%	0.22	0.09
20%	0.14	0.05
10%	0.07	0.01

[1]Assuming 80% system load diversity, design airflow for the supply fan is 0.72 cfm/ft^2 (0.80 x 0.90 cfm/ft^2).

[2]Assuming total static pressure of 4 in. H_2O and 60% fan efficiency: bhp = (0.72 cfm/ft^2 x 4 in. H_2O) / (6356 x 0.60) = 0.00076 bhp/ft^2 or 0.76 bhp/1000 ft^2.

[3]Part-load fan power determined using Table G3.1.3.15 (depicted in Figure 5), Appendix G, ASHRAE/IESNA Standard 90.1-2007, *Energy Standard for Buildings Except Low-Rise Residential Buildings.*

Considering that the central supply fan in a VAV system typically operates at less than design airflow for much of the year, the actual difference in fan energy use between the two systems may be small. And in climates with several months of cold weather, the VAV system might actually use less fan energy than the ACB system over the year.

When operation of the system is considered over the entire year, the difference in fan energy use is much less than the difference in zone primary airflow (at design cooling conditions) might suggest. The actual difference depends on climate, building usage, and design of the air distribution system, so it requires a whole-building energy simulation.

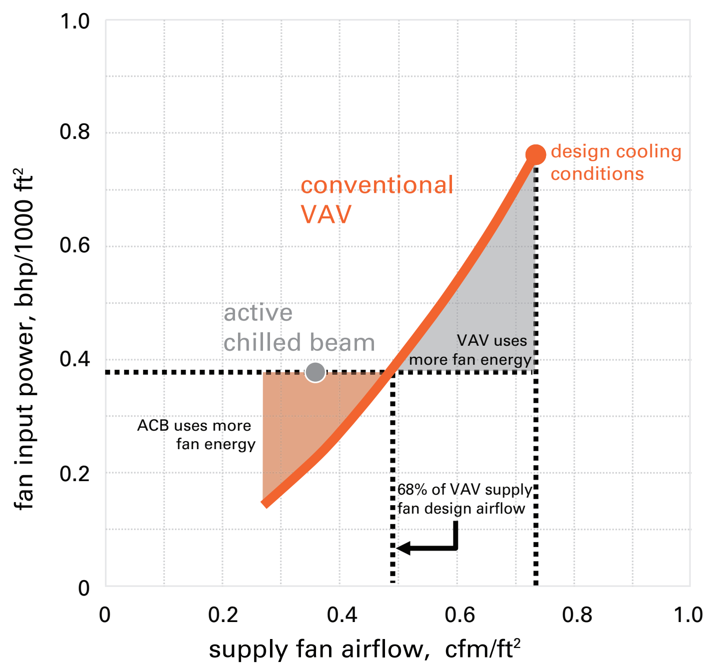
Is chiller efficiency higher? Because the temperature of water supplied to the chilled beams must be relatively warm (typically between 58°F and 60°F) to avoid unwanted condensation, the water chiller can be more efficient (higher COP) than if it was making colder water (typically between 40°F and 44°F) for central VAV air-handling units.

But remember, the primary AHUs in a chilled-beam system must dehumidify the air to a low enough dew point to offset the space latent loads. This typically requires as cold, if not colder, water—offsetting the same space latent load with less primary airflow requires that air to be drier—than is typically used for a conventional VAV system (see Table 1).

Whether or not the warmer water delivered to the chilled beams contributes to a higher chiller COP depends on the configuration of the chiller plant.

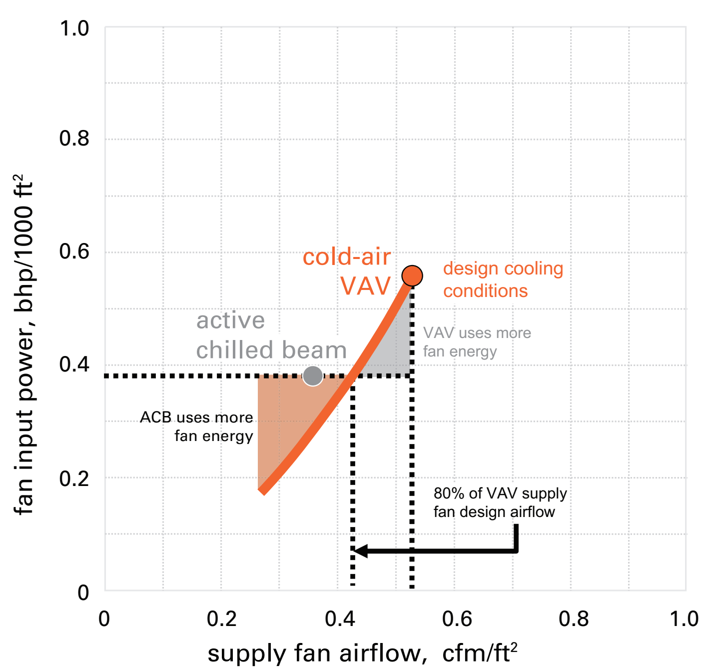
If separate chiller plants are used, the plant serving the primary AHUs can make cold water to sufficiently dehumidify the primary air, while the plant serving the chilled beams can make warmer water and benefit from operating at a higher COP. Of course,

Figure 6. Supply fan power at part load (ACB vs. conventional VAV)



Note: For the ACB system, assumes total static pressure 4 in. H_2O and 60% fan efficiency: bhp = (0.36 cfm/ft^2 x 4 in. H_2O)/(6356 x 0.60) = 0.00038 bhp/ft^2 or 0.38 bhp/1000 ft^2. For the VAV system, see Table 2. The VAV fan input power levels off below 0.27 cfm/ft^2, assuming all VAV terminals are at the minimum airflow setting, 30% of design zone airflow (0.30 x 0.90 cfm/ft^2 = 0.27 cfm/ft^2).

Figure 7. Supply fan power at part load (ACB vs. cold-air VAV)



Note: For the cold air VAV system (48ºF primary air, rather than conventional 55ºF), assumes the ductwork has been downsized for first cost savings—so total static pressure remains the same at 4 in. H_2O—and assumes 60% fan efficiency.

using separate chiller plants increases installed cost, and considering that only a fraction of the annual cooling loads in the building are handled by the chilled beams—the primary AHUs handle the cooling loads of the outdoor air, plus typically provide some of the space sensible cooling—the benefit of a dedicated chiller serving the chilled beams and operating at a higher COP is lessened on an annual basis.

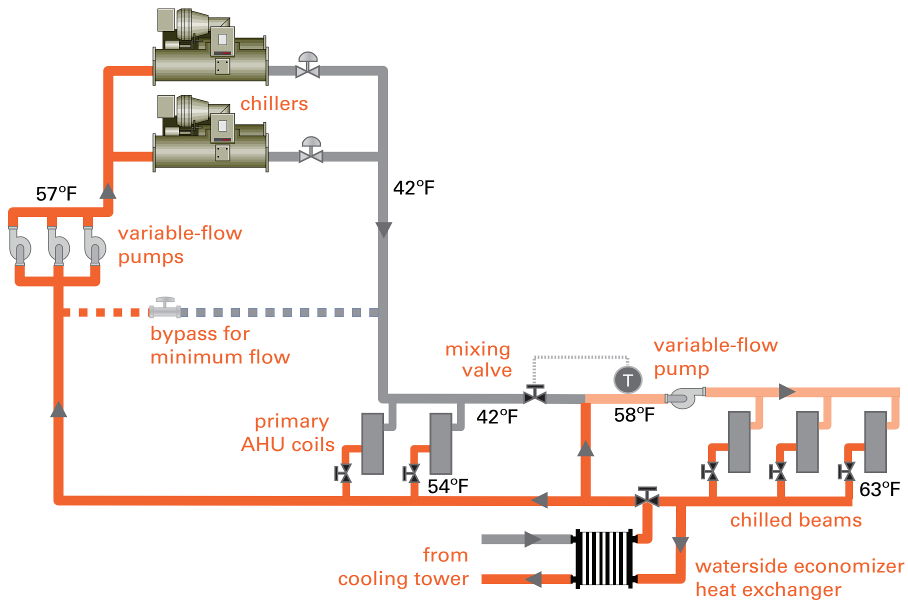
A more common plant configuration is to use one set of chillers to make cold water (42°F in this example) for the entire building (Figure 8). Some of this water is delivered to the primary air-handling units for dehumidification, and the rest is mixed with warm water (63°F) returning from the chilled beams to achieve the desired water temperature (58°F) to deliver to the beams.

The supply water temperature is not the only factor that impacts chiller energy use. Some other factors that impact the difference in chiller energy use between ACB and VAV systems include:

- ACB systems typically do not employ *demand-controlled ventilation* (DCV), so outdoor airflow remains constant. VAV systems, however, are more likely to implement some form of DCV to reduce outdoor airflow and chiller energy use during partial occupancy.[3]
- Since ACB systems are designed to deliver less primary airflow, the primary air system typically has little or no capacity for *airside economizing*. A VAV system, however, can provide up to 100 percent of design supply airflow for "free" cooling, when outdoor conditions permit. Using the previous office space example, the primary air-handling unit in the ACB system can provide up to 0.36 cfm/ft^2 of outdoor air for economizing. The VAV air-handling unit can provide up to 0.72 cfm/ft^2, which offers more capacity to reduce chiller energy use through an airside economizer cycle.

  Of course, chilled beam systems can be equipped with a waterside economizer, such as a plate-and-frame heat exchanger (Figure 8). For most applications, however, a waterside economizer does not provide as much energy savings as an airside economizer.

Figure 8. Shared chilled-water plant with waterside economizer



Note: Other plant configurations have been used—such as putting the primary AHU coils and chilled beams in series or putting chillers in series and diverting some of the warmer water leaving the upstream chiller to serve the chilled beams—and they can provide energy benefits over this example configuration, but they also increase complexity.

However, focusing on the chiller is only part of the story. Since the temperature of water supplied to the chilled beams must be relatively warm (typically between 58°F and 60°F) to prevent condensation, they are often selected with high water flow rates (gpm) to provide the required cooling capacity. Typically, ACBs are selected with a 5°F or 6°F ΔT, compared to the 10°F to 14°F ΔT commonly used for selecting cooling coils in a VAV system.

Therefore, ACB systems will likely use more **pumping energy** than a VAV system because a) the pumps need to move a lot more chilled water (higher gpm) and b) this water needs to be pumped throughout the entire building (to one or more chilled beams installed in every space), rather than pumped only to the mechanical rooms that contain the centralized VAV air-handling units.

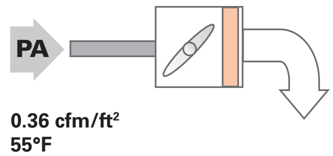
Is reheat energy avoided? Some proponents suggest that since chilled beams provide space cooling using zone-level cooling coils, they avoid the need for reheat that is common in many VAV systems.

A VAV system reduces the airflow delivered to the zone as the sensible cooling load in that zone decreases. Reheat is only activated after primary airflow (PA) has been reduced to the minimum setting for the VAV terminal. Returning to the previous office space example, if the minimum airflow setting for the VAV terminal is 40 percent of design airflow (0.90 cfm/ft^2 x 0.40 = 0.36 cfm/ft^2), reheat is not activated until the space sensible cooling load drops below 40 percent of the design load. Below that point, the 55°F primary air (now 0.36 cfm/ft^2) provides more cooling than the space requires, so the heating coil inside the VAV terminal unit warms the primary air just enough to avoid overcooling the space (Figure 9).

An ACB system reduces the water flow rate through the coil as the sensible cooling load in the zone decreases. When the space sensible cooling load drops below the point when the chilled-water (CHW) valve is completely closed, the primary airflow (0.36 cfm/ft^2 delivered at 55°F, for this example) provides more cooling than the space requires, so a four-pipe ACB will need to open the hot-water valve and add heat to the induced room air (RA) to avoid overcooling the space (Figure 9). (A two-pipe ACB would need to either add heat to the space with a separate heating system, switchover the water-distribution system to deliver warm water to all beams, or just allow the space to overcool.)

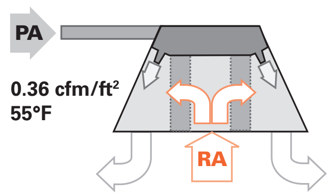
When the space sensible cooling load drops below 40 percent of design load, both the VAV and ACB system are delivering 0.36 cfm/ft^2 of 55°F primary air to the zone, so the same amount of heat would be needed to prevent overcooling the space.

Figure 9. Impact of reheat energy



VAV terminal with 40% minimum airflow setting
primary airflow at design conditions = 0.90 cfm/ft^2
primary airflow when reheat is activated:
= 40% x 0.90 cfm/ft^2
= 0.36 cfm/ft^2
cooling provided when primary airflow is at minimum:
= 0.36 cfm/ft^2 of 55°F primary air

Reheat is needed to avoid overcooling the space when the space sensible cooling load < 40% of design load.



active chilled beam
primary airflow at design conditions = 0.36 cfm/ft^2
primary airflow when CHW valve is fully closed
= 0.36 cfm/ft^2
cooling provided when CHW valve is fully closed:
= 0.36 cfm/ft^2 of 55°F primary air

Heat is needed to avoid overcooling the space when the space sensible cooling load < 40% of design load.

The annual difference in reheat energy depends on the primary airflow delivered to the ACBs, the minimum airflow setting for the VAV terminals, the primary air temperature (see sidebar), whether there is any temperature reset strategy being used for the primary-air in either system, and the number of hours when the zones experience these low cooling loads. (VAV systems can benefit by using parallel fan-powered VAV terminals to draw warm air from the ceiling plenum as the first stage of heating, thereby reducing reheat energy use.)

However, reheat is not the only factor that impacts the difference in heating energy use between ACB and VAV systems. As mentioned earlier, ACB systems typically do not employ *demand-controlled ventilation*, so outdoor airflow remains constant. VAV systems, however, are more likely to implement some form of DCV to reduce outdoor airflow and heating energy use during partial occupancy.[3]

Cold versus neutral primary-air temperature

Some ACB systems are designed to dehumidify the primary air to a low dew point, and then reheat the air to a dry-bulb temperature close to space temperature, sometimes referred to as "neutral" air. Delivering primary air at a neutral temperature avoids the need to add heat to prevent overcooling the space at low cooling loads (Figure 9), but it requires the entire space sensible cooling load to be offset by the chilled beams (the primary air provides no space sensible cooling).

Therefore, designing the primary AHU for neutral-temperature air typically increases installed cost due to the need for more beams, higher primary airflows, and increased water flows. For this same example office space (see Figure 11), if the primary air is delivered at 70°F dry bulb (rather than 55°F), six chilled beams would need to be installed (rather than four), primary airflow would increase to 0.50 cfm/ft^2 (rather than 0.36 cfm/ft^2), and water flow would increase to 9 gpm (rather than 6 gpm).

Also, designing for neutral-temperature primary air likely increases system energy use because more cooling must be provided by the beams (higher primary airflows, increased water flows) and, since the primary air must be dehumidified to a low dew point, the dehumidification performed by the primary AHU is approximately the same, whether the primary air is delivered neutral or cold.

Of course, the actual difference in energy use for a specific building depends on climate, building usage, and system design. Building analysis tools (like TRACE™ 700) can be used to analyze the performance of different HVAC systems and design strategies. However, only a few whole-building energy simulation tools can currently model chilled beam systems, so be sure to understand the capabilities of the software.

To illustrate, TRACE™ 700 was used to compare an active chilled beam system to a chilled-water VAV system in an example office building (Figure 10). The baseline building uses a conventional chilled-water VAV system, modeled according to Appendix G of ASHRAE Standard 90.1-2007.

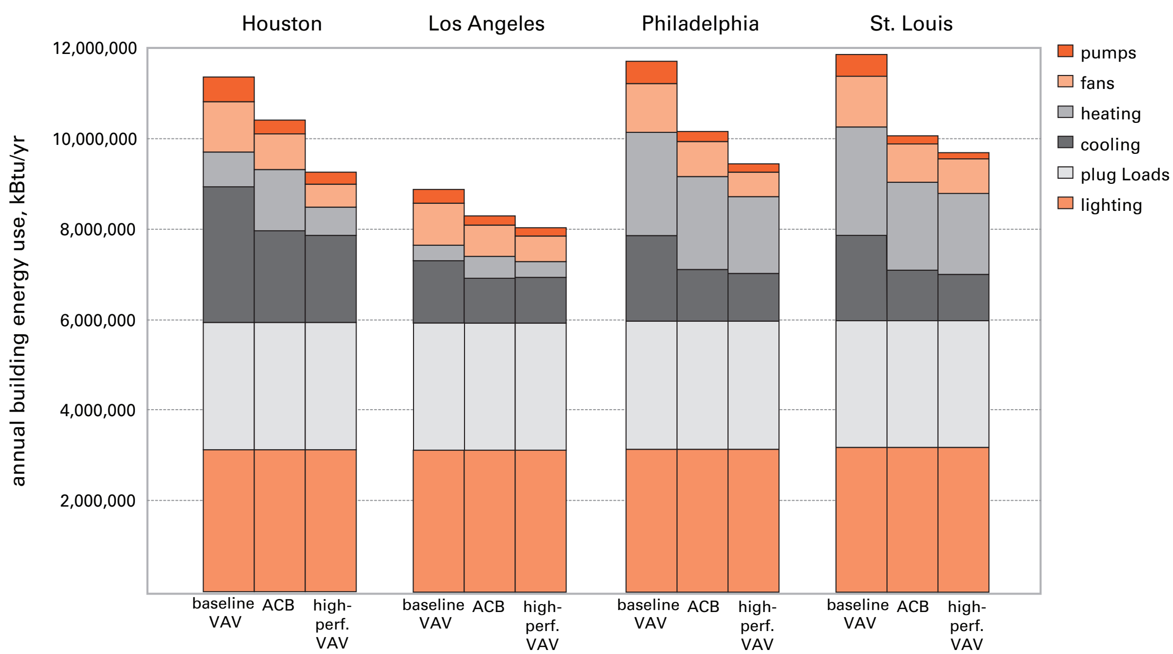
The building with the chilled beam system includes four-pipe ACBs, separate primary AHUs for perimeter versus interior zones, separate chiller plants (one supplying warm water to the chilled beams and the other supplying cold water to the primary AHUs), and an airside economizer on the primary air systems. The chilled-water system serving the primary AHUs is a "low flow" design with high-efficiency chillers, a VFD on the cooling tower fans, and the chiller-tower optimization control strategy.

For this example, the building with the ACB system uses about 8 percent less energy than the conventional chilled-water VAV system in Houston, 7 percent less in Los Angeles, 13 percent less in Philadelphia, and 15 percent less in St. Louis.

However, this analysis also investigated a "high-performance" chilled-water VAV system, which uses 48°F supply air (rather than the conventional 55°F, but kept the same size ductwork), supply-air-temperature reset and ventilation optimization control strategies, and parallel fan-powered VAV terminals. The chilled-water system is a "low flow" design with high-efficiency chillers, a VFD on the cooling tower fans, and the chiller-tower optimization control strategy.

The high-performance chilled-water VAV system uses less energy than either the baseline VAV or the ACB systems: 11 percent less energy than the ACB system in Houston, 3 percent less in Los Angeles, 7 percent less in Philadelphia, and 5 percent less in St. Louis.

Figure 10. Comparison of annual energy use for an example office building



Challenges of Using Chilled Beams

The purpose of this section is to review the challenges associated with applying chilled beams, and some ways to overcome those challenges.

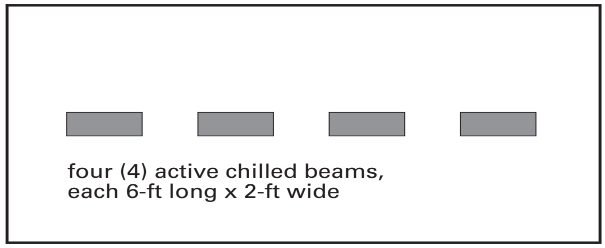
High installed cost. Active chilled beam systems often have a higher installed cost than other system alternatives because the chilled beams have a relatively low cooling capacity. This means that more coil surface area is needed to provide the required space cooling. In addition, piping and control valves must be field-installed to deliver chilled water to the beams installed in every space.

Two factors that significantly limit the cooling capacity of an ACB are warm entering-water temperature and low inlet air pressure.

1) The entering-water temperature must be relatively warm (typically between 58°F and 60°F) to prevent condensation. With a warmer water temperature, a higher water flow rate (gpm) and/or more coil surface area is needed to provide the required cooling capacity.

2) The higher the static pressure entering the nozzles of an ACB, the more room air is induced through the coils inside the chilled beam. However, a higher inlet pressure requires more fan power. To keep fan energy use low—and avoid the fan energy penalty that plagued the high-pressure induction systems that were popular in the 1960s and 1970s—active chilled beams are typically selected with an inlet static pressure between 0.3 and 0.5 in. H_2O, which is similar to the inlet pressure required by a VAV terminal unit. To induce sufficient room airflow with this low inlet static pressure, the pressure drop for induced airflow through the coil must be very small, which again results in the need for a lot of coil surface area.

Figure 11. Larger area of ceiling space required for chilled beams



Example: 1000 ft^2 office space

Note: This example is based on two-way, four-pipe active chilled beams. Primary airflow of 360 cfm (0.36 cfm/ft^2) is delivered to the beams at 55°F dry bulb and 0.50 in. H_2O inlet static pressure. The entering water temperature is 58°F and the waterside ΔT is 5°F. The design space sensible cooling load is 19,500 Btu/h (19.5 Btu/h/ft^2) and the zone cooling setpoint is 75°F.

Figure 12. Examples of active chilled beam installations

Returning to the same example, four 6-ft long active chilled beams are needed to offset the space sensible cooling load for this 1000 ft^2 office space (Figure 11). Notice how much ceiling area is covered with the beams.

Figure 12 includes photos of some active chilled beam installations. Notice that 6- or 8-ft long chilled beams are installed almost end-to-end down the entire length of the space, and another row of beams is installed four to six feet over.

To reduce the installed cost associated with the beams, system design variables can be changed to reduce the number of active chilled beams required (Table 3). However, these changes are not without consequences. Changing a system design variable often impacts the installed cost, or energy use, of another component in the system.

For example, increasing the water flow rate through the chilled beam increases its cooling capacity, but also increases pump energy use and requires the installation of larger pipes and pumps.

Need to prevent condensation. As mentioned earlier, a chilled beam does not typically contain a condensate drainage system. Because air from the occupied space passes over the cold surface of the coils, the indoor dew point must be maintained below the surface temperature of the chilled beam coil to prevent moisture from condensing on the coil and dripping into the space. Bottom line: chilled beams are for sensible cooling only, not dehumidification.

In ACB systems, the primary air system is used to offset the space latent load and typically maintain the indoor dew point at or below 55°F to prevent condensation. In addition, the water temperature delivered to the beams is typically maintained between 58°F and 60°F, sufficiently above the dew point of the space.

For most applications, maintaining the indoor dew point at 55°F means that the primary air must be dehumidified to somewhere between 45°F and 53°F dew point (44 to 60 grains/lb). Of course, how dry the primary air must be depends on the quantity of primary air delivered, as well as the latent load in the space. If the primary air is delivered drier (at a lower dew point), less airflow (cfm) is needed to dehumidify the space (Table 1). Similar to the entering water temperature, if the dry-bulb temperature of the primary air is too cold, there might be increased risk of condensation forming on the housing of the ACB.

Even though the primary air system is designed to control the indoor dew point, it is also important to a) limit the infiltration of humid outdoor air by designing and constructing a *tight building envelope* and b) control the HVAC system to maintain *positive building pressurization* during humid weather.

In some climates, indoor humidity can increase overnight or over the weekend. To avoid condensation at startup, it may be necessary to lower the indoor dew point prior to activating the chilled beams. This operating mode, often called "humidity pull-down," requires the primary air system to start prior to occupancy, and operate long enough for the humidity inside the building to reach the desired dew point (55°F, for example) before chilled water is supplied to the beams.

Risk of water leaks. Since chilled beams are water-based systems, piping (both supply and return) and control valves must be field-installed to distribute chilled water to multiple beams in every space of the building. This impacts installed cost, but also increases the risk of water leaks due to the increased piping and pipe connections.

Table 3. Impact of various system design variables on installed cost of active chilled beams

system design variable	impact on installed cost of the chilled beams	impact on performance of the overall system
2-pipe versus 4-pipe chilled beams	A 2-pipe beam provides more cooling capacity than a 4-pipe beam because more coil surface is available.	Using 2-pipe beams requires a separate heating system, otherwise it can result in poorer comfort control because either cold water or warm water is delivered to all zones.
Primary airflow rate (cfm)	Increasing the primary airflow rate through the nozzles results in more air being induced from the space, which increases the capacity of the chilled beam coils.	Increasing the primary airflow rate increases primary AHU fan energy use, increases noise in the space, and requires a larger primary AHU and larger ductwork.
Inlet static pressure of the primary air.	Increasing the static pressure at the inlet to the nozzles results in more air being induced from the space, which increases the capacity of the chilled beam coils.	Increasing the inlet pressure increases primary AHU fan energy use, and increases noise in the space.
Dry-bulb temperature of the primary air (see sidebar, p. 7)	Delivering the primary air at a colder temperature means that less of the space sensible cooling load needs to be offset by the chilled beams.	Using a colder primary-air temperature may cause the space to overcool at low sensible cooling loads, thus requiring the chilled beam (or separate heating system) to add heat to prevent overcooling the space.
Entering water temperature	Supplying colder water to the chilled beam increases the cooling capacity of the beam.	Using a colder water temperature requires the space dewpoint to be lower to avoid condensation, which means the primary air needs to be dehumidified to a lower dew point.
Water flow rate (gpm)	Increasing the water flow rate increases the cooling capacity of the beam.	Increasing the water flow rate increases pump energy use and requires larger pipes and pumps.

As mentioned earlier, active chilled beams are available in either two-pipe or four-pipe configurations. Four-pipe systems can provide better zone-by-zone comfort control because some zones can receive chilled water for space cooling while others simultaneously receive hot water for space heating. But four-pipe systems also require twice as much piping and twice as many pipe connections, which further increases installed cost and increases the risk of water leaks.

Of course, if a VAV system uses hot-water heating coils in the VAV terminals, piping and control valves must be field-installed to distribute hot water to each terminal. But there will be fewer pipe connections since a zone served by one VAV terminal would likely contain several chilled beams. And the chilled-water piping need only distribute water to the centralized mechanical rooms, not to each zone.

Chilled beam and natural ventilation

It is not uncommon for design teams to consider using chilled beams in buildings with operable windows or other natural ventilation openings. The concept is to use natural ventilation when outdoor conditions permit, and switch to mechanical ventilation when it's too hot or too cold outside. But it's not easy to control indoor dew point with the windows open. If natural ventilation is to be used, pay special attention to preventing condensation on the chilled beams.

No filtration of locally-recirculated air. Chilled beams are typically not equipped with particulate air filters. Since the coils are intended to operate dry (no condensation), there may be less concern about preventing wet coil surfaces from getting dirty. But, there is still the concern about removing particles that are generated within the space or brought into the space—on shoes or clothing, for example.

Limited heating capability. Active chilled beams can provide some heating capacity by flowing hot water through the coils. But capacity is somewhat limited and it can be a challenge to deliver the warm air at a high enough velocity to force it down into the occupied space. Historically, many buildings with chilled beams have used a separate heating system, such as baseboard radiators or convectors or in-floor radiant heating.

One effective approach for dehumidifying the primary air in a chilled beam system is to use a series, Type III desiccant dehumidification wheel (Figure 13).[4]

This configuration places a Type III desiccant wheel in series with the cooling coil. The regeneration side of the wheel is upstream of the coil and the process side is downstream of the coil. The desiccant adsorbs water vapor from the process air downstream of the coil, enabling the system to deliver drier (lower dew point) primary air (PA) without lowering the leaving-coil temperature (CA).

When the wheel rotates, it releases the adsorbed water vapor into the air upstream of the coil (MA'), and the cooling coil gets a second chance to remove the transferred water vapor via condensation. Unlike a traditional desiccant dehumidification system, moisture transfer occurs without the need to add regeneration heat, at most operating conditions.

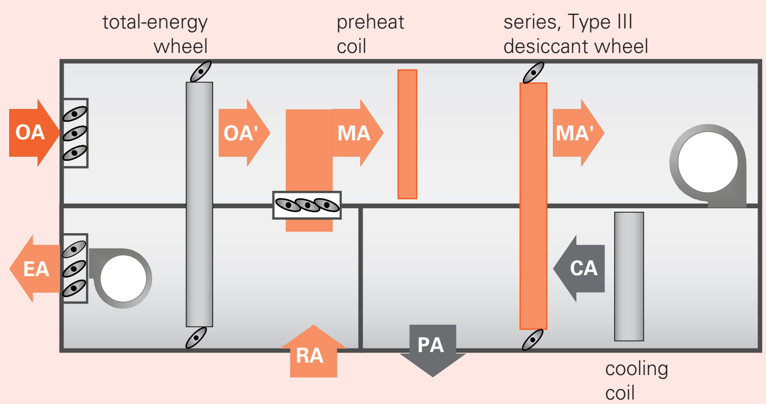
This particular primary AHU configuration also includes a total-energy (or enthalpy) wheel to precondition the outdoor air before it enters the building.

Figure 14 shows the performance of this dual-wheel primary AHU for an active chilled beam system. The Type III desiccant wheel adsorbs water vapor from the saturated air (very high RH) leaving the cooling coil (CA), dehumidifying the primary air (PA) to a dew point of 43°F. Sensible heat added by this adsorption process raises the primary-air temperature to 55°F dry bulb.

Mixed air (MA) entering the regeneration side of the wheel is less humid, about 50 percent RH in this example. At this RH, the wheel can no longer hold the water vapor it adsorbed downstream of the coil, so it is released from the desiccant into the mixed air (MA'), without needing to add any heat for regeneration.

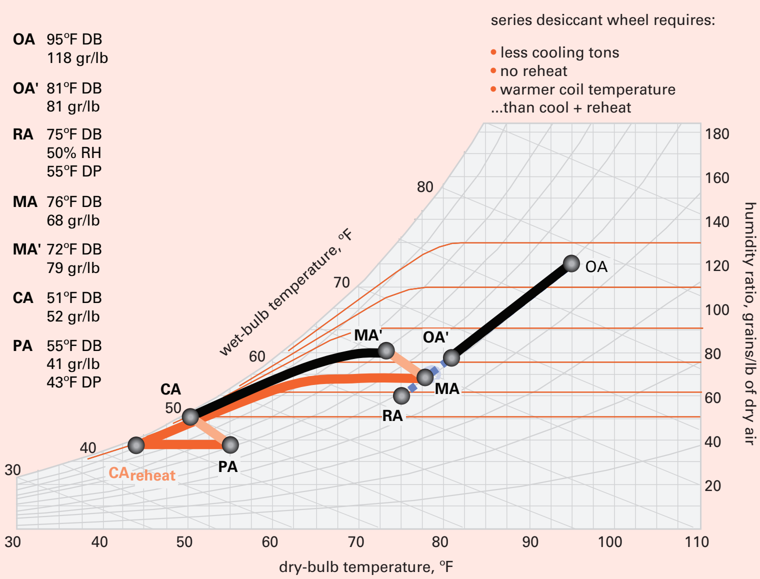
To deliver the same primary air (PA) condition using a traditional "cool+reheat" system, the cooling coil would need to cool the air to nearly 43°F dry bulb (CA_{reheat}) to achieve the same 43°F dew point. Then a reheat coil must raise the dry-bulb temperature to 55°F. By contrast, the dual-wheel AHU configuration can deliver the same dew point using fewer tons, no reheat, and with a warmer leaving-coil temperature—51°F vs. 43°F (Figure 14). This warmer coil temperature allows for more efficient mechanical cooling.

Figure 13. Example primary air-handling unit for an active chilled beam system



Note: Trane refers to the series Types III desiccant wheel as CDQ™ (Cool, Dry, Quiet).

Figure 14. Performance of dual-wheel primary AHU



Summary

Although active chilled beam systems have some advantages over all-air VAV systems, some of the claimed advantages are likely being overstated. And, chilled beams present some unique challenges that must be properly addressed in the design and operation of the system.

The actual energy use of a specific building depends on climate, building usage, and system design, so it warrants analysis using whole-building energy simulation software.

It is the responsibility of the design team to understand the benefits, as well as the challenges, to determine if chilled beams are the right system choice for a given project.

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You can find this and previous issues of the Engineers Newsletter at www.trane.com/engineersnewsletter. To comment, e-mail us at comfort@trane.com

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