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 article is to investigate the common claims about chilled beam systems. This is part two.

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.

Table 1

<table>
<thead>
<tr>
<th>Function</th>
<th>Impact on Outdoor Airflow</th>
</tr>
</thead>
<tbody>
<tr>
<td>Primary Airflow</td>
<td>0.085 cfm/ft²</td>
</tr>
<tr>
<td>Dehumidification</td>
<td>47°F dew point</td>
</tr>
<tr>
<td>Heating</td>
<td>55°F indoor dew point</td>
</tr>
</tbody>
</table>

Based on typical occupant latent loads in an office space, if primary airflow is only 0.085 cfm/ft², 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/ft² of primary air is required to provide the required sensible cooling capacity.

For this same example office space, the design airflow delivered by a conventional VAV system would be 0.90 cfm/ft². 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 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.

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

<table>
<thead>
<tr>
<th>Description</th>
<th>Formula</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum outdoor airflow required (per ASHRAE 62.1-2007)</td>
<td>0.085 cfm/ft² (for LEED® EQ credit, 1.3 x 0.085 = 0.11 cfm/ft²)</td>
</tr>
<tr>
<td>Active chilled-beam system</td>
<td></td>
</tr>
<tr>
<td>Primary airflow required to offset space latent load</td>
<td>0.085 cfm/ft² (DPT&lt;sub&gt;PA&lt;/sub&gt;=47°F or W&lt;sub&gt;PA&lt;/sub&gt;=47 gr/lb)</td>
</tr>
<tr>
<td></td>
<td>0.11 cfm/ft² (DPT&lt;sub&gt;PA&lt;/sub&gt;=49°F or W&lt;sub&gt;PA&lt;/sub&gt;=51 gr/lb)</td>
</tr>
<tr>
<td></td>
<td>0.36 cfm/ft² (DPT&lt;sub&gt;PA&lt;/sub&gt;=53°F or W&lt;sub&gt;PA&lt;/sub&gt;=60 gr/lb)</td>
</tr>
<tr>
<td>Primary airflow needed to induce sufficient room airflow to provide sensible cooling</td>
<td>mfr A: 0.36 cfm/ft² (DBT&lt;sub&gt;PA&lt;/sub&gt; = 55°F)</td>
</tr>
<tr>
<td></td>
<td>mfr B: 0.38 cfm/ft² (DBT&lt;sub&gt;PA&lt;/sub&gt; = 55°F)</td>
</tr>
<tr>
<td></td>
<td>mfr C: 0.35 cfm/ft² (DBT&lt;sub&gt;PA&lt;/sub&gt; = 55°F)</td>
</tr>
<tr>
<td>Conventional VAV system</td>
<td></td>
</tr>
<tr>
<td>Primary airflow needed to provide sensible cooling</td>
<td>0.90 cfm/ft² (DBT&lt;sub&gt;PA&lt;/sub&gt; = 55°F)</td>
</tr>
</tbody>
</table>

1 For an office space, Table 6-1 of ASHRAE Standard 62.1-2007, Ventilation for Acceptable Indoor Air Quality, requires 5 cfm/pt (R<sub>j</sub>) plus 0.16 cfm/ft² (R<sub>a</sub>), and suggests a default occupant density of 5 people/1000 ft²: V<sub>oz</sub> = (6 cfm/person x 5 people/1000 ft²) + 0.06 cfm/ft² = 0.085 cfm/ft².

2 For moderately active office work, the 2009 ASHRAE Handbook-Fundamentals (Table 1, Chapter 18) suggests a latent load of 200 Btu/pt/person. Using the same default occupant density (5 people/1000 ft²): Q<sub>latent</sub> = (200 Btu/h/pt/person x 5 people/1000 ft²) x P<sub>t</sub> = 1.0 Btu/h/ft² = 0.69 x (64 - 47 gr/lb): V<sub>PA</sub> = 1.0 Btu/ft² / 0.69 x (64 - 47 gr/lb) = 0.085 cfm/ft², or V<sub>PA</sub> = 0.11 cfm/ft² ( DBT<sub>PA</sub> = 55°F).

3 For fan coil units, the space sensible cooling load typically ranges from 1.7 to 24 Btu/ft²/person (which equates to about 0.8 to 1.1 cfm/ft² if a conventional VAV system is used). Assuming a space sensible cooling load of 19.5 Btu/ft², 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/ft² 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/ft² of primary air, respectively.

4 Assuming the same space sensible cooling load of 19.5 Btu/ft², a zone cooling setpoint of 75°F, and a primary-air dry-bulb temperature of 55°F: V<sub>PA</sub> = 19.5 Btu/ft² / (0.69 x (64 - 55°F)) = 0.90 cfm/ft².
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 multiplezone 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.

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/ft² for this example.

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/ft² at design cooling conditions. For the ACB system, primary airflow (0.36 cfm/ft²), and therefore fan power, remains constant at all load conditions.

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

<table>
<thead>
<tr>
<th>supply fan airflow, % of design</th>
<th>supply fan airflow, cfm/ft²</th>
<th>supply fan power², bhp/1000 ft²</th>
</tr>
</thead>
<tbody>
<tr>
<td>100%</td>
<td>0.72¹</td>
<td>0.76²</td>
</tr>
<tr>
<td>90%</td>
<td>0.65</td>
<td>0.63</td>
</tr>
<tr>
<td>80%</td>
<td>0.58</td>
<td>0.51</td>
</tr>
<tr>
<td>70%</td>
<td>0.50</td>
<td>0.40</td>
</tr>
<tr>
<td>60%</td>
<td>0.43</td>
<td>0.31</td>
</tr>
<tr>
<td>50%</td>
<td>0.36</td>
<td>0.22</td>
</tr>
<tr>
<td>40%</td>
<td>0.29</td>
<td>0.15</td>
</tr>
<tr>
<td>30%</td>
<td>0.22</td>
<td>0.09</td>
</tr>
<tr>
<td>20%</td>
<td>0.14</td>
<td>0.05</td>
</tr>
<tr>
<td>10%</td>
<td>0.07</td>
<td>0.01</td>
</tr>
</tbody>
</table>

¹Assuming 80% system load diversity, design airflow for the supply fan is 0.72 cfm/ft² (0.80 x 0.90 cfm/ft²).
²Assuming total static pressure of 4 in. H₂O and 60% fan efficiency: bhp = (0.72 cfm/ft² x 4 in. H₂O) / 8356 x 0.601 = 0.00076 bhp/ft² or 0.76 bhp/1000 ft².
³Part-load fan power determined using Table G3.1.15 (depicted in Figure 5f, Appendix G, ASHRAE/ESNA Standard 90.1-2007, Energy Standard for Buildings Except Low-Rise Residential Buildings).

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

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.

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CONGRATULATIONS!
NEW NEBB TECHNICIAN

Robert Cortez
ACCO
ASHRAE Hawaii Chapter’s November meeting was held at the Plaza Club Honolulu. The meeting topic was Testing, Adjusting, Balancing Roundtable, and the 3 speakers did a great job reinforcing the values of NEBB and sharing interesting points about our profession. Steve Smith, President of the Northern California/Hawaii NEBB Chapter was one of the speakers. Ryan Chang, a NEBB Certified Professional with TAB Engineers, LLC was the moderator for the event.
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- Duct Leakage Testing per SMACNA or as specified
- Cleanroom Performance Testing (CPT) (NEBB certified professional)
- Biological Safety Cabinet (BSC) (NSF accredited field certifier)
- Laboratory Fume Hood Testing and Certification (ICB/TABB certified professional)
- Sound and Vibration Testing (NEBB and ICB/TABB certified professional)
- As-built mechanical drawing services
- HVAC system installation, operational analysis, and troubleshooting
- Fire/Smoke Control System Testing (ICB/TABB certified professional)
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