PENNSTATE

Phone: (814) 355-1504

Stanley A. Mumma, Ph.D., P.E., *Prof Emeritus*
College of Engineering
Department of Architectural Engineering

104 Engineering Unit A
The Pennsylvania State University
University Park, PA 16802-1400
e-mail: sam11@psu.edu

July 6, 2012

Comments regarding the April 2012 ASHRAE Journal Chilled Beam article: by Dr. Livchak

I have been hesitant to advocate the use of Chilled Beams with DOAS because many US applications use considerable recirculated air, mixed centrally with OA, to achieve the desired beam sensible cooling capacity. DOAS, by definition and design, employs 100% OA, hence overcoming the challenges of meeting ASHRAE Std. 62.1 with multi-space all-air designs employing recirculated air. Unfortunately, many in the industry have come to consider DOAS primarily as a way to decouple the space sensible and latent loads, which it excels at, and have lost sight of its primary goal: which is to energy efficiently meet ASHRAE Std. 62.1 with assurance, a most difficult task for multi space all air systems.

The excellence I see in Dr. Livchak's paper is his recognition of the problem caused by using centrally recirculated air with DOAS to obtain the desired chilled beam sensible cooling capacity. This point is discussed in his attached paper on pages 54 to 59 starting with the section HOW TO INCREASE BEAM EFFECTIVENESS. His discussion is supplemented with two excellent graphics, Figures 4 and 5 on page 56. The booster terminals shown in the figures are more clearly illustrated at these sites: http://www.krueger-hvac.com/lit/press/klpsdrelease.asp, http://www.krueger-hvac.com/ecatalog/model.aspx?refid=1569.

Conclusion, ***do not use any recirculated air*** with DOAS, if it's use compromises the system's ability to meet Std. 62.1 with assurance.

Stanley A. Mumma, Ph.D., P.E.
Professor Emeritus of Architectural Engineering

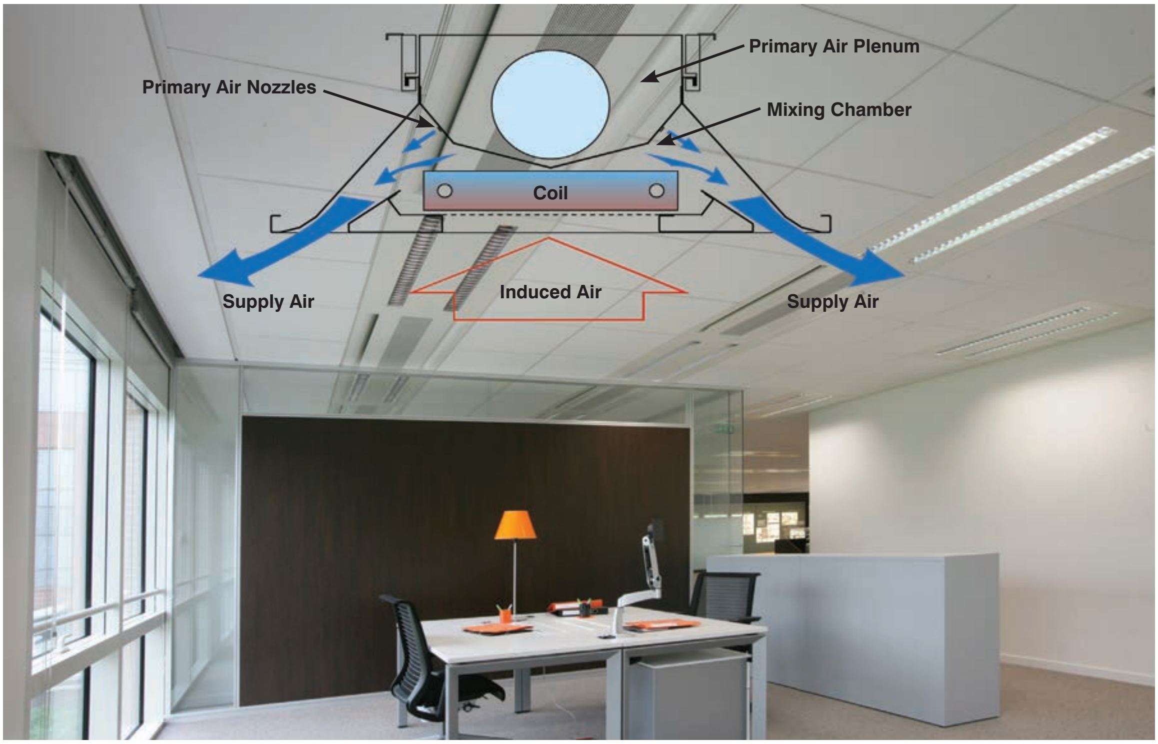

Photo 1: Typical active beam functional diagram.

Don't Turn Active Beams Into Expensive Diffusers

By **Andrey Livchak, Ph.D.,** Member ASHRAE; and **Chris Lowell,** Member ASHRAE

Active chilled beams have been used for more than 20 years. The term "active chilled beam" became an oxymoron, with active beams being used for cooling and heating. Now, they are called "active beams" or simply "beams." Beams are gaining popularity in North America and are being designed with higher airflows to match increasing space loads.

Beam designs with primary airflows significantly exceeding space latent load and minimum ventilation requirements are also driven by engineers' attempting to reduce system first costs and total number of beams. Unfortunately, this approach compromises the system's energy performance and diminishes advantages of active beam systems over all-air systems. This often leads to active beams being used as expensive diffusers.

This article will help engineers gain fundamental knowledge about what parameters affect active beam performance and introduce new criteria for beam selection.

This article is for the HVAC engineers who are familiar with chilled beams and have used them in design practice. For readers who want to know more about chilled beams, please refer to the prior publications in this journal and references at the end of this article.

Primary air in active beams (*Photo 1*) is supplied into a mixing chamber through rows of nozzles. Negative pressure that is created in the mixing chamber facilitates induction of room air

About the Authors

Andrey Livchak, Ph.D., is vice president of engineering for Halton Group Americas. **Chris Lowell** is managing director of Halton Indoors Americas.

through the cooling coil. Induced air, cooled by the cooling coil, mixes with the primary air. This mixture of recirculated cooled air and primary air is supplied to the space. In an optimum design, primary airflow is intended to satisfy space outside air requirements and dehumidification to avoid any condensation on beams' surfaces. The cooling coil is used to compensate for space sensible load only. Primary air is always cooled and dehumidified before it enters a beam.

Designing Chilled Beam Systems

When first introduced in Northern Europe, the design objective for active beam systems was to separate ventilation load from space sensible load and handle space cooling and dehumidification with minimum airflow. Water is a more effective media than air to transport energy due to its higher density and specific heat. One unit volume of water can carry about 3,500 times more energy compared to the same volume of air.

Already high space loads in the U.S. are often further overestimated by design programs not accounting for transient heat transfer effect, as well as the tendency of engineers to put a "safety margin" on top of the estimates, resulting in HVAC systems designed with oversized cooling capacity. In active chilled beam applications, this leads to beams designed to operate with excessive airflows. *As a consequence, the active chilled beam often works as an expensive diffuser, with the water valve shut and all cooling provided by primary air.* Indeed, beam cooling output is controlled by either a mixing valve, regulating water temperature in the coil, or by an on-off valve modulating water flow through the coil. This valve closes when space thermostat setting is satisfied. When the system is oversized and primary air provides sufficient space cooling, the water valve stays closed. We did see installations where all of the control valves on active beams were closed throughout the entire summer.

Active beam total cooling capacity is the sum of cooling capacity provided by the primary air and the beam coil.

$$P = P_a + P_w \tag{1}$$

Cooling capacity provided by the primary air is calculated using the following equation:

$$P_a = m_p \times c_{pa}\,(t_p - t_r) \tag{2}$$

Assuming primary air is supplied at 55°F (12.8°C) and space temperature is maintained at 75°F (23.9°C), the primary air provides about 22 Btu/h (6.45 W) of cooling per cfm of primary air (10.4 W per 1 L/s). *Figure 1* demonstrates contribution of air (P_a) and water (P_w) to the total cooling capacity of an active beam (P) as a function of primary airflow. As the primary airflow increases, the water contribution to the total beam cooling capacity drops and the air contribution in total beam cooling capacity increases. This chart is representative of a beam designed to operate at fairly low primary airflow. There are chilled beam systems operating at 20 cfm per linear ft of beam (31 L/s·m) and higher with primary air contributing 60% or more to the total beam cooling output.

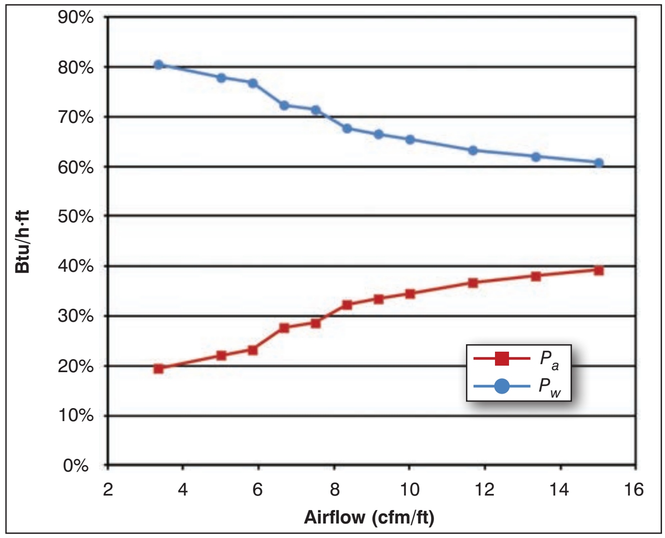

Figure 1: Contribution of air and water to total cooling capacity of an active beam.

C. Wilkins and M. Hosni[1] demonstrated that plug loads are overestimated for office buildings. This, along with added safety design factor for HVAC equipment, often results in the air-conditioning systems operating only at 80% capacity on a design day. As we mentioned previously, most active beams are designed as constant air volume systems with water in the coil providing space temperature control. Let's see what happens to an office space with an active beam sized with primary airflow to cover 60% of total cooling load. Assuming 20% safety margin for extra cooling capacity, this leaves only 28% (100% – 1.2 × 60%) for cooling output adjustment via cooling coil. This is certainly not enough to adequately respond to a variable load in the space in the intermediate season. As a result, the building will be overcooled in summer, thermal comfort compromised and overall HVAC system energy consumption increased.

With that being said, we don't want to underestimate benefits of active beams. When properly applied, it is an energy-efficient, low maintenance and comfortable system. Our recommendation is to *design active beams to operate at minimum primary airflows. If that is not possible, use a variable air volume* (*VAV*) *beam system,* which is described later in this article.

Designing Beams for Minimum Primary Airflow

As concluded earlier, the most efficient chilled beam system is the one that operates at minimum primary airflow and satisfies space sensible load primarily by using the cooling coil. The most efficient, by cooling performance, active beam is the one that provides the highest cooling output at minimum primary airflow per unit length of beam. Let's define a parameter that represents this important performance of an active beam and call it *coil output to primary airflow ratio* (*COPA*).

COPA represents the amount of cooling (or heating, when active beams are used for heating) energy produced by the active beam coil per volume of primary air used. COPA is calculated at typical space temperature, inlet water temperature and water flow through the coil.

$$\text{COPA} = \frac{P_w}{q_p}$$

The higher the COPA ratio, the more efficient chilled beam design, the more effectively primary air is used. COPA is an important parameter to consider when selecting active beams with primary airflows exceeding minimum outside air requirements. In spaces with high latent load or high outside air requirements, where primary air provides most of the cooling along with dehumidification, application of active beams operating as a constant air volume system becomes less desirable and the COPA ratio loses its importance.

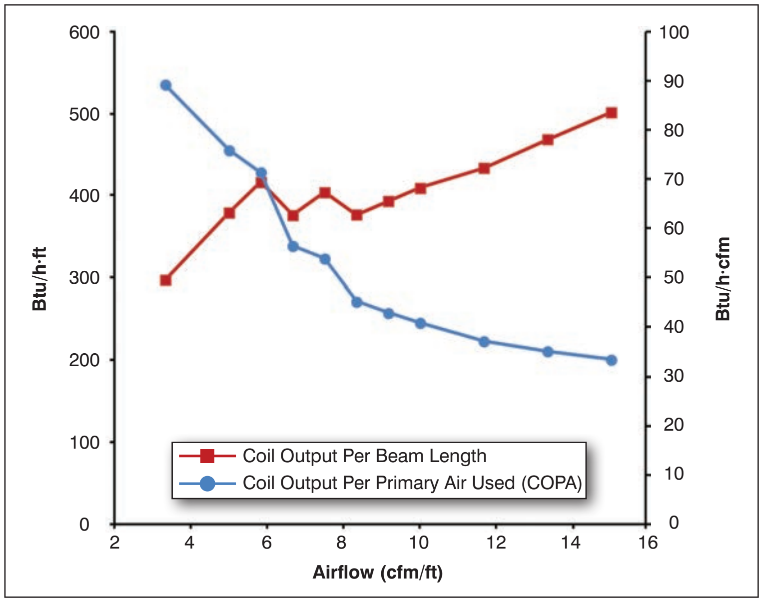

Figure 2: Coil cooling output as a function of primary airflow.

As an example, the chart in *Figure 2* demonstrates the relationship between coil cooling capacity and primary airflow for an active beam. As primary airflow increases from 3.3 to 15 cfm per linear foot of beam (5.1 to 23.2 L/s·m), coil cooling output increases by 70% as well, however coil cooling output per primary airflow (COPA) becomes three times smaller.

Figure 2 represents beams with four-nozzle configurations. As can be seen from the chart, the correlation between the COPA and the primary airflow is similar for a single-nozzle design, as well as for multiple-nozzle configurations. Even though a given beam design with a fixed-nozzle configuration may have the same induction coefficient, correlation between coil heat transfer coefficient and primary airflow is not linear (Equation 8). As the primary airflow increases, the coil heat transfer coefficient grows slower than the cooling capacity of primary air. In the previous example, water cooling output increases by 70% while cooling by the primary air (at constant supply air temperature) increases by (15/3.3 – 1) × 100% = 355%.

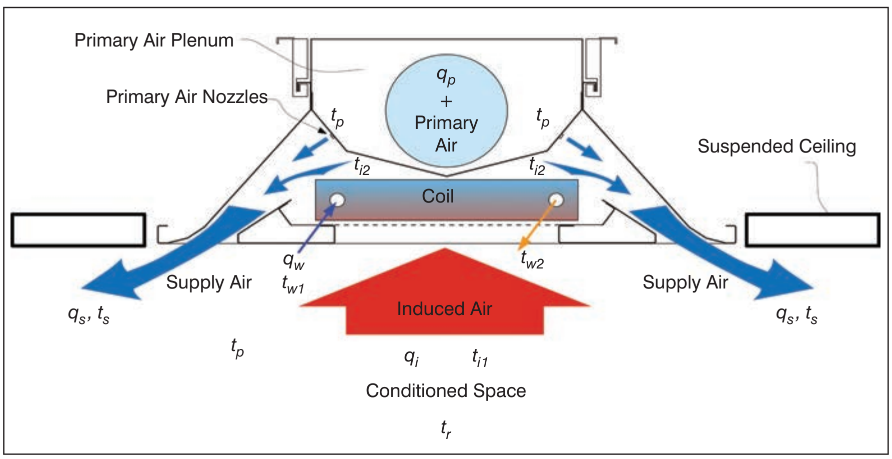

Figure 3: Cross-sectional view for a typical active beam.

How to Increase Beam Effectiveness

Increase cooling coil output while maintaining minimum primary airflow. In this section we present equations governing active beam cooling capacity to better understand their performance. *Figure 3* shows a cross-section of a typical active beam.

Coil Cooling Capacity

The following system of equations describes coil heat transfer under steady-state conditions assuming no condensation on the coil surface.

$$P_w = m_w \times c_{pw} (t_{w2} - t_{w1}) \tag{3}$$

$$P_w = K \times A \times \Delta t \tag{4}$$

$$P_w = m_i \times c_{pa} (t_{i1} - t_{i2}) \tag{5}$$

It is not uncommon in design practice to see the chilled beam water side cooling capacity estimated using single Equation 3. Often, the water temperature difference is assumed to be 4°F to 6°F (2.2°C to 3.3°C) and the other two equations affecting coil cooling capacity are neglected. It is important to understand that the temperature of water leaving the coil t_{w2} is a function of several parameters including the temperature and velocity of induced air travelling across the coil, as well as the temperature and velocity of the water passing through it. For a given coil, heat transfer coefficient K is a function of all the previously mentioned parameters, and it should be calculated but never assumed. Effectiveness of the active beam design is defined by

Nomenclature

A Coil heat transfer area, ft^2

A_f Coil free cross-sectional area perpendicular to the direction of induced airflow, ft^2

$a, a', b, c, n, n1, n2$ Empirical coefficients

c_{pa} Specific heat of air, Btu/(lb·°F)

c_{pw} Specific heat of water (liquid media), Btu/(lb·°F)

K Coil heat transfer coefficient, Btu/h/(ft^2·°F)

K' Coil heat transfer coefficient times coil heat transfer area, Btu/h·°F

k_{alt} Correction factor for coil heat transfer at different elevations above sea level

K_{in} Induction coefficient $K_{in} = q_i / q_p$

m_i Mass flow rate of induced air, lb/h

m_p Mass flow rate of primary air, lb/h

m_w Water mass (liquid media) flow rate, lb/h

P Chilled beam total cooling capacity, Btu/h

P_a Cooling capacity, provided by primary air, Btu/h

P_w Coil cooling capacity, Btu/h

P'_w Coil cooling capacity per beam length, Btu/h·ft

q_i Induced airflow, cfm

q_p Primary airflow, cfm

t_{i1} Induced air temperature entering the coil, °F

t_{i2} Induced air temperature leaving the coil, °F

t_p Primary air temperature, °F

t_r Average room air temperature, °F

t_{w1} Temperature of water (liquid media) entering the coil, °F

t_{w2} Temperature of water (liquid media) leaving the coil, °F

Δt Average temperature difference between cooling media in the coil and induced air temperature before and after the coil $\Delta t = \frac{t_{i1} + t_{i2}}{2} - \frac{t_{w1} + t_{w2}}{2}$, °F

ρ_i Induced air density, lb/ft^3

ρ_s Supply air density, lb/ft^3

ω Velocity of water (liquid media), measured in the cross-section of the coil pipe, fpm

its heat transfer coefficient and coil heat transfer surface area. The higher the KA value, the higher the coil cooling output, the higher the COPA.

Coil Heat Transfer Coefficient

Coil heat transfer coefficient K for a given chilled beam design depends on:

- Mass velocity (velocity times density or mass airflow divided by free cross-sectional area of the coil) of induced air travelling across the coil $v\rho_i$; and
- Velocity of water (liquid media) in the coil ω.

It is governed by equations of forced convection for air passing through the coil with water (or other cooling media circulating inside the coil) and can be described by the following empirical equation.

$$K = a' (v\rho_i)^{n1} \omega^{n2} \qquad (6)$$

Convective heat transfer coefficient from the water to the pipe is significantly higher than that from the coil fins to induced air passing through the coil. That is why $v\rho_i$ has dominant effect in equation 6.* Our own mea-

* This statement assumes turbulent flow conditions in the pipes, which is always the case as long as water flow is above 0.5 gpm (0.03 L/s) for a ½ in. pipe.

surements show that power factor $n1$ is three to four times higher than $n2$.

Since coil heat transfer area is constant for a given active beam, a similar equation can be used to calculate heat transfer coefficient times the coil surface area or coil cooling output per degree of temperature difference Δt.

$$K' = KA = a\,(v\rho_i)^{n1}\omega^{n2} \tag{6a}$$

The velocity of induced air v, which is defined by induced airflow per unit length of coil and coil cross-sectional free area, depends on primary airflow q_p, beam induction coefficient K_{in} and temperature difference Δt. The first two parameters take into account active beam induction force and the second: buoyancy force acting on non-isothermal air moving in a vertical direction across the coil. For example, if warm induced air moves up across the coil, it cools down and buoyancy force slows its motion. On the contrary, if active beam design deploys downward movement of induced air, this buoyancy force will be accelerating the air movement across the coil when in cooling mode.

$$v = \frac{K_{in} \times q_p + b \times \Delta t}{A_f} \tag{7}$$

Combining Equations 6a and 7 and taking into consideration that A_f is constant for a given beam design, we can derive the equation defining coil heat transfer coefficient as function of temperature difference Δt, velocity of water in the pipes ω and primary airflow q_p.

$$K' = a\,[(c \times \Delta t^n + K_{in} \times q_p)\rho_i]^{n1}\omega^{n2} \tag{8}$$

In active beams with induced air moving horizontally across the coil, coefficient b in Equation 7 becomes 0 because velocity across the coil is not affected by buoyancy force and Equation 8 is reduced to:

$$K' = a(K_{in} \times q_p \times \rho_i)^{n1}\omega^{n2} \tag{8a}$$

Equation 8a can also be used to define the heat transfer coefficient for passive beams, where airflow through the coil is determined by convection forces only as represented by the following equation.

$$K' = a(\Delta t^n \times \rho_i)^{n1}\omega^{n2} \tag{8b}$$

Photo 2: Active beam with a booster fan for hotel applications.

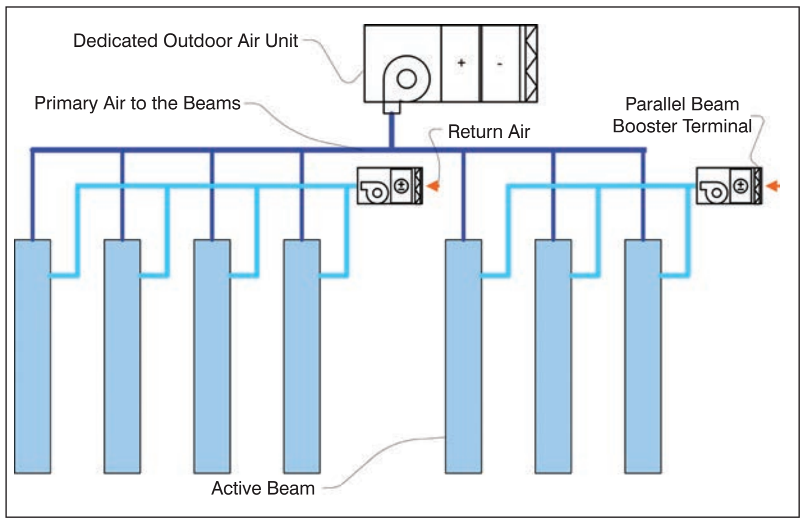

Figure 4: Variable air volume beams with the parallel boosting terminal.

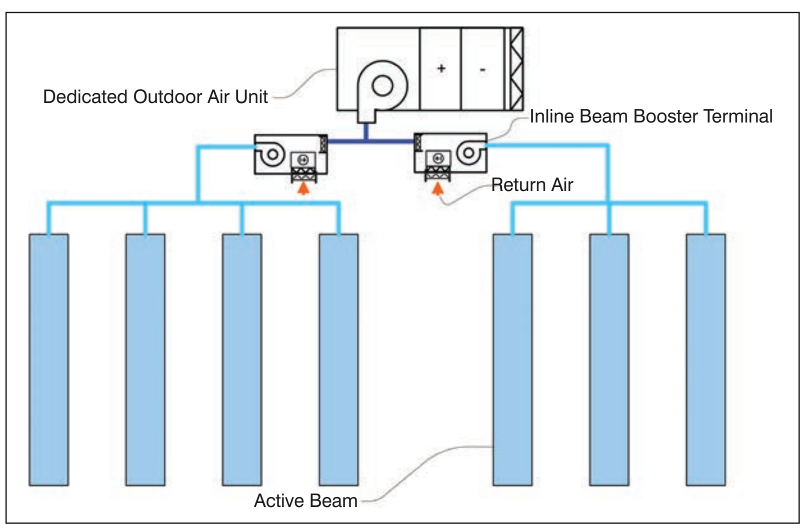

Figure 5: Variable air volume beams with the inline boosting terminal.

Equation 8 and its derivatives are important for understanding what parameters affect coil cooling or heating output. They provide sufficient information to simulate any active beam in energy simulation software. All empirical coefficients a, c, n, $n1$, $n2$ and K_{in} are constant for a given beam design and can be derived from the manufacturer's cooling and heating (when testing active beams for heating) capacity tests. The test sequence along with the calculation

Advertisement formerly in this space.

procedure to determine these coefficients can be part of the method of tests for active beams currently being developed by ASHRAE. This would help *integration of active beam systems in the energy simulation software.*

Power factor *n1* in these equations is three to four times higher than power factor *n2*. That leads to the conclusion that *increasing* K_{in} *has a major effect on the COPA. The higher induced airflow through the coil, the higher the coil output, hence the higher the COPA.*

Equation 8 also contains air density, which allows for calculating a correction factor for the coil heat transfer coefficient when designing active beam systems for high elevations above sea level. Assuming the manufacturer's coil heat transfer data is measured and presented at sea level, the correction factor to account for a higher elevation above sea level is:

$$k_{alt} = (1 - 6.8754 \times 10^{-6}\ \mathrm{ft})^{5.2599\ n1}$$

Where *ft* is elevation above sea level in feet.

Engineers involved in energy simulations and product development will find useful the detailed system of equations presented in this section. Those, who work on active beams projects, look for manufacturers' design tools that transform the complexity of these equations in an easy-to-use electronic selection tool.

Evolution of Active Beams: VAV Beams

As discussed earlier, active beams operating as constant air volume systems have significant limitations in adjusting cooling output to manage variable space load. It is a matter of time until we see active beams evolving and being introduced as variable air volume systems. Widespread use of high efficiency electronically commutated motors make designing fan-assisted beams feasible without sacrificing energy efficiency.

A fan-assisted or VAV beam uses a built-in fan to increase the circulation of room air through the cooling coil during peak loads. It operates as an active beam, with the fan off, most of the time. Such a system is designed for hotel applications where the fan is used to boost the cooling or heating output of a beam up to 30% during peak load hours and to accelerate room conditioning in transition from unoccupied to occupied mode. An example of such a beam is shown in *Photo 2*.

Another variation of the VAV active beam system, shown in *Figures 4* and *5*, uses beam booster terminals (BBT) to increase the cooling/heating output for a group of beams. A BBT, fitted with a cooling coil and a condensate drain, can also be used for dehumidification in case of excessive latent load in a zone served by this unit.

The parallel booster terminal is similar in design to a fan-coil unit with a variable speed fan. As shown in *Figure 4*, this design uses a special beam design where air from the BBT connects to a separate plenum with additional induction nozzles. This arrangement increases return airflow through the coil and boosts its cooling/heating output. It is more efficient, but requires additional ductwork.

The design of an inline BBT is similar to a fan-powered terminal with the only difference that induced airflow can be controlled independently of primary airflow, supplied into the BBT. As shown in *Figure 5*, the inline BBT does not require additional ductwork and can work with regular active beams. It relies primarily on the boosting capability of the BBT because its ability to increase circulation through the beam coil is limited. A design with an inline BBT has an advantage when used with active beams equipped with VAV dampers. These VAV dampers are designed to bleed part of the beam's primary air directly into the room allowing for a significant airflow increase.

Both designs with the inline and parallel BBTs require an integrated control system enabling boosting terminals only when the cooling or heating capacity of the coil in an active beam has reached its limit, but a space thermostat still calls for cooling or heating.

Conclusions and Recommendations

When designing active beam systems, don't limit your effort to sizing the beams under peak load conditions only, verify beams' performance under partial load. Design objective should be to minimize primary airflow and maximize use of water coil for cooling and heating. Minimum airflow shall satisfy space latent load and minimum ventilation requirements. If such design is not feasible, use variable air volume active beams to maximize use of water cooling/heating under partial load conditions.

Active beam coil output to primary airflow ratio is an important parameter in active beam selection. The higher the COPA value, the more efficient the active beam design.

Presented system of equations describing active beam cooling/heating output, along with empirical formulae for coil heat transfer coefficient, can be used to represent active beams in energy simulation programs.

Acknowledgment

The authors would like to thank Vladimir Yarmarkovich, senior vice president of Cosentini Associates, for his valuable comments and advice in preparation of this article.

References

1. Wilkins C., M. Hosni. 2011 "Plug load design factors. " *ASHRAE Journal* 53(5):30 – 34.

2. Virta M. et al. 2004. "Chilled Beam Application Guidebook." Brussels: REHVA Guidebook.

3. *2011 ASHRAE Handbook—HVAC Applications*, chap. 57.

4. European Standard EN 15116 Ventilation in buildings—Chilled beams—Testing and rating of chilled beams. CEN 2008. ■