FläktGroup
SEMCO

CHILLED BEAMS + PINNACLE

APPLICATION GUIDE

KEY TO SUCCESSFULLY APPLYING CHILLED BEAM TECHNOLOGY IN HUMID CLIMATES

Due to the growing number of LEED certified and “Green” buildings being designed, the significant energy efficiency benefits offered by chilled beam technologies have caught the interest of a growing number of architects and engineers worldwide. Chilled beam systems can improve the quality of the indoor environment; increasing occupant comfort and reducing energy costs, while cutting the size of air handling units and the associated ducted airflow by 66% when compared to traditional HVAC designs.

As a result of these benefits, chilled beam systems (both active and passive) have become the system design of choice in Scandinavia, central Europe and the United Kingdom over the past 20 years. High building efficiency standards, the need to cut carbon emissions and regulations limiting the fan energy that can be used based on building size have accelerated their use in these markets in recent years.

Chilled Beams + Pinnacle: **Increasing occupant comfort and reducing energy costs**

Fläkt Woods has been a dominant chilled beam supplier and technology leader in the European market for more than 30 years. SEMCO, a Fläkt Group Brand, is now producing this chilled beam technology in the US. Ambient humidity levels in the US are often much higher than those in Europe. High space humidity presents a challenge to applying chilled beams. SEMCO has invested in design tools and test facilities to understand and address this challenge.

Chilled beams systems demand that internal latent loads be accurately estimated and that indoor humidity levels be controlled effectively to avoid the possibility of condensation on the "cooled" coil (active beam) mounted at the ceiling surfaces.

In humid climates, this demands the primary airflow to be delivered at low dew points if the energy efficiency benefits offered by chilled beams are to be recognized. This document explains why low supply air dew points are required and how the SEMCO Pinnacle® system is uniquely suited to achieve these conditions in the most cost effective, energy efficient manner.

1. INTRODUCTION TO ACTIVE CHILLED BEAMS

Active chilled beam technology provides an energy efficient secondary, sensible only cooling system. The beam incorporates a cooling coil, which is served with moderate temperature chilled water typically ranging between 56°F to 59°F. Active beams achieve a much greater cooling capacity than provided by passive chilled beams by introducing primary air to the device and using strategically positioned slots to induce room air through the coil.

By adjusting the width of the slots, the amount of induction air can be varied from a ratio of about 1:1 (induction air to primary air) to about 4:1. By doing so, the amount of cooling capacity achieved, the amount of outdoor air provided and the amount of supply airflow delivered to the space can all be adjusted to meet the needs of the individual space. For example, a typical selection for a 6 foot chilled beam served by 58°F chilled water and 40 cfm of primary, outdoor air will deliver approximately 3,600 BTU/Hr of sensible cooling at a very low sound level (25 decibels).

The Fläkt Group IQIC, IQID and IQCA active chilled beams introduced to the US have some particularly important design enhancements that help the installation and field adjustment of these devices to match the needs of the occupied spaces. One example is the **Comfort Control** (see **Figure 2**) feature, which allows for the amount of induction air and thereby cooling capacity to be easily adjusted after installation.

Equally important is the **Flow Pattern Control** (see **Figure 2**) feature, which allows the installer or building occupant to direct the supply air from the beam, as needed to fit the space configuration, compensate for heat gain through windows and accommodate specific comfort needs of individuals.

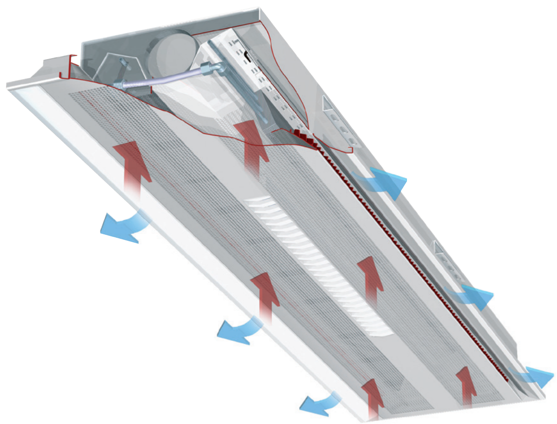


FIGURE 1: Graphic of SEMCO's Flexicool® Chilled Beam

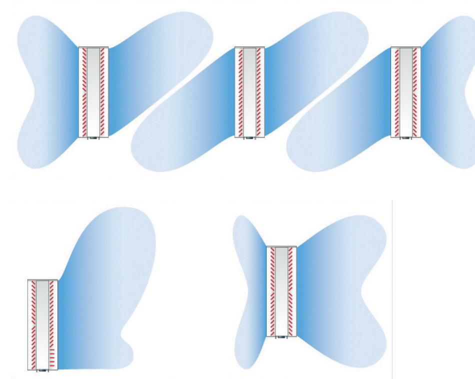


FIGURE 2: Flow Pattern Control and Comfort Control technologies

Benefits of Active Chilled Beams

Active chilled beams offer numerous advantages over more common HVAC systems including variable air volume (VAV), constant volume and fan coil approaches. The following list discusses some of the more important advantages:

Greatly reduced airflow through ductwork: The primary airflow introduced to the active chilled beams will typically be one half to one third of that required by a VAV or constant volume system at peak cooling conditions. This allows for a smaller air handling unit(s), smaller ductwork, reduction in the floor to ceiling height, reduced air shaft area requirements and lower filter cost.

Most importantly, the reduction in airflow results in significant fan energy savings, which lowers the installed fan horsepower and cuts the cost of the electrical service to the air handling systems.

Optimum occupant comfort and indoor air quality (IAQ): An active chilled beam system controls both temperature and humidity within the occupied space. With a constant supply of primary air (often all outdoor air) the minimum outdoor air ventilation requirements are met at all conditions and in all spaces, a major problem for VAV systems. As a result, ASHRAE Standard 62 requires more outdoor air to be processed by a VAV system than the chilled beam system to achieve similar ventilation effectiveness.

The air delivery from the beams is evenly distributed throughout the space so the risk of drafts and "dumping" of cold air is reduced while actually supplying more airflow (due to the induced room air) to the room than would be supplied by a VAV system at peak conditions.

Very quiet operation: It is common for a properly designed active chilled beam system to contribute almost no detectable noise to the occupied space with sound power levels at or below 25 to 30 decibels.

Simple controls: Chilled beams are most often controlled by simply opening and closing a chilled water valve, based on a call for cooling from the wall thermostat. These valves are configured to fail closed and are commonly fitted with a condensation sensor to protect against beam condensation in case of a chiller problem or widow opening on humid days.

Low maintenance: Unlike a fan coil unit, the chilled beam does not include a fan, require electrical wiring or need a filter. The low velocity associated with the induced air across the cooling coil limits the collection of airborne dust such that operating history has shown that simply vacuuming once every two to three years is sufficient to maintain optimum performance.

2. PROPERLY CALCULATING THE PRIMARY AIRFLOW TO THE CHILLED BEAM

Early adopters of active chilled beam technology in the US have been predominantly engineering firms who specialize in the design of laboratory facilities. Several excellent articles have been written on this design approach by McLay[1] and Barnet[2]. Both authors highlight the advantage provided by active chilled beams in laboratories where high peak sensible loads (typical of these facilities) and high ventilation rates (generally about 6 air changes per hour) benefit from the performance offered by the chilled beam approach.

According to one of the authors, a typical VAV system would need to provide a supply air volume that would equal 15 air changes per hour to accommodate the sensible load. As a result, energy consumption was reported to be far less with chilled beams, as much as 50% less. Other important advantages are summarized within these articles.

Laboratory facilities are unusual in that the outdoor air quantities are so high and the internal sensible loads so significant, that it simplifies the process of determining the primary airflow volume and dew point required to avoid condensation at the chilled beams. The primary airflow is simply set by the laboratory outdoor ventilation air needs which are far greater than the minimum ventilation rate required by other building types based on occupancy. The high sensible load allows the primary air to be delivered to the chilled beams quite and cool without the risk of over-cooling spaces. Finally, the relatively low occupancy density and high primary airflow ensures that the internal latent loads are effectively satisfied thus condensation on the beams can be avoided even with a primary air handling unit delivering conventional leaving coil temperatures (say 54°F). This is not the case, however, for almost all other applications.

Calculating the primary airflow and dew point for non-laboratory applications

It is most desirable that the primary airflow delivered to the active chilled beams is similar to that required for building ventilation in order to optimize energy efficiency and first cost. Whether or not this can be accomplished depends on three important design parameters:

The internal latent load: which must be handled by the primary airflow.

The ventilation rate required or desired: ASHRAE 62 requirements, for example.

The dehumidification driving force: differential between the dew point desired within the space and that delivered by the primary air system to the beams/space.

Upon careful analysis it becomes clear that the internal latent load (and thereby primary airflow quantity) is dominated by ambient humidity levels (i.e. climate). For example, it is very common that chilled beam systems located in Europe (and other low ambient humidity regions), can be effectively operated in the most desired manner; using only the outdoor ventilation air as primary air serving the beams, cooled to moderate (54°F) dew point conditions. This is possible since the ventilation rates used in many parts of Europe exceed the ASHRAE minimum recommendations and, more importantly, the ambient humidity levels and therefore the internal latent loads are significantly less than encountered here in the US (and most other countries around the world).

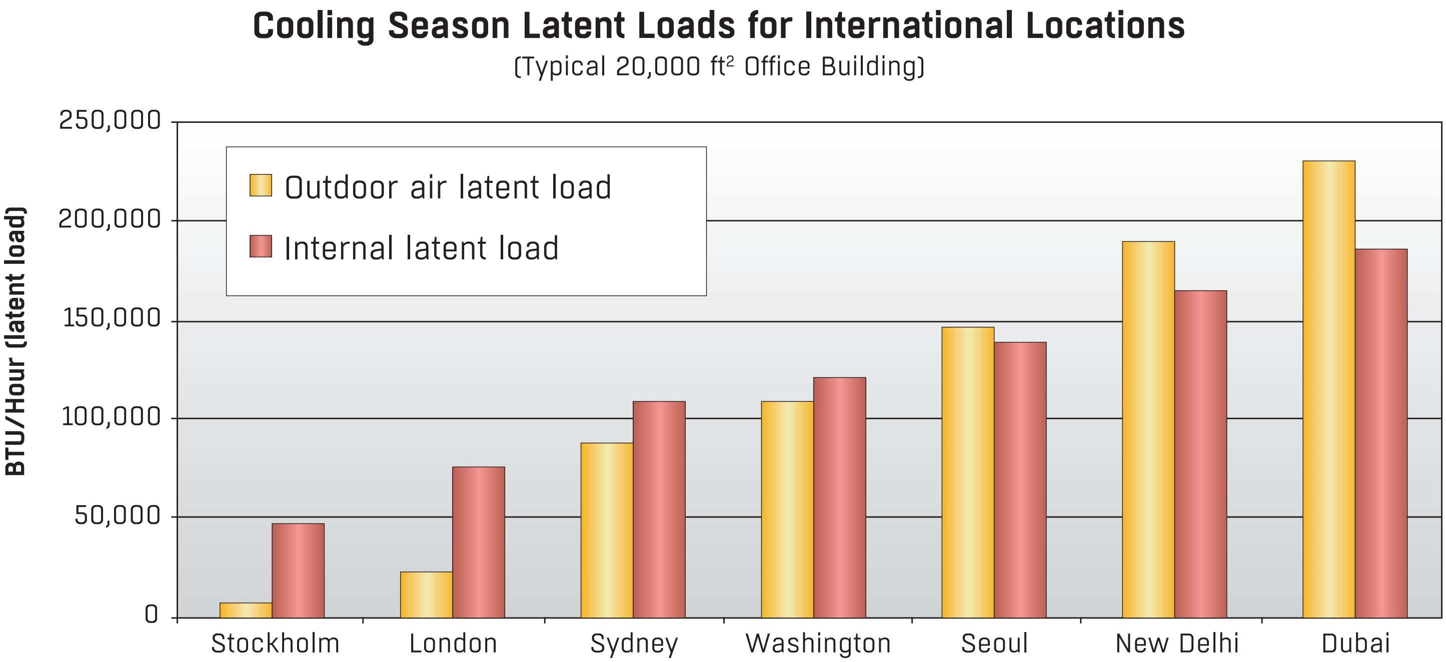


FIGURE 3: Indoor and outdoor latent loads for a 20,000 ft² office in various cities globally

Figure 3 compares modeled internal and outdoor air latent loads for a 20,000 ft² office building for various cities around the globe. This graphic highlights the challenge imposed on buildings designed to incorporate chilled beam systems located in humid climates. Higher outdoor humidity levels result in greater indoor latent loads due to infiltration, door openings and permeance[3]. When considering chilled beam technologies, it is critical that the internal latent loads be calculated correctly and that they accurately reflect the properties of the building envelope. Commercially available building load modeling programs must often be carefully configured to provide accurate latent load values. An excellent resource for determining internal latent loads is the ASHRAE Humidity Control Design Guide[3]. SEMCO has combined the methodology recommended by this design guide along with other ASHRAE recommended default values to create an internal latent design tool specifically for chilled beam systems[4] that greatly simplifies this process.

Calculating the primary airflow in non-humid climates (Stockholm example)

Using this 20,000 ft² office building example, assuming 200 occupants and tight construction (infiltration at .1 cfm/ ft² of facade), the primary airflow needed to avoid condensation on the beams served by 60°F chilled water for a project in Stockholm can be easily calculated. We will assume that the space humidity will be maintained at or below a 59°F dew point or 76 grains of moisture (1°F below the chilled water temperature serving the beams) and that a traditional air handling unit delivering 54°F air at 61 grains of moisture provides the primary air. The internal latent load is estimated to be:

> 49,600 BTU/hr for Stockholm location. 49,600 BTU/hr = Primary airflow * .68 * (76 grains – 61 grains)
> **Primary airflow required in Stockholm = 4,863 cfm**

Since an office building in Stockholm will be provided with at least 5,000 cfm of outdoor air for ventilation purposes (25 cfm/ person) then the ventilation air requirement determines the primary airflow to the chilled beams and not the need to control space humidity.

Calculating the primary airflow in more humid climates (Washington, DC example)

In contrast, when this same 20,000 ft^2 office building is moved from Stockholm Sweden, where the cooling season peak humidity level is 80 grains, to Washington DC, having a peak humidity level of 125 grains of moisture, determining the proper primary airflow and supply moisture condition gets more complicated.

When the primary airflow required to avoid beam condensation is determined using the increased latent load (124,800 BTU/Hr) that results from the Washington DC elevated ambient humidity, a different outcome is observed. Using the same 54°F (61 grains) supply air condition from the primary air handling unit we now get:

124,800 BTU/hr = Primary airflow * .68 * (76 grains – 61 grains)
Primary airflow required in Washington DC = 12,235 cfm

In this case the primary airflow is clearly set by the space dehumidification requirement and not by the ventilation airflow needed. Assuming the same 25 cfm/person ventilation rate, the primary airflow required for dehumidification is almost 2.5 times that required for ventilation purposes (12,235/5,000). This increased primary airflow erodes much of the benefit provided by the chilled beam technology. Much of the system energy efficiency advantage is lost as the primary airflow approaches that typically used for a conventional VAV system. Cost savings associated with smaller air handling units, ductwork, fan horsepower and electrical service are reduced or lost. A less obvious problem is that this may also result in a degradation of the comfort level in the occupied space along with an increase in the noise level generated by the chilled beam system.

By increasing the amount of 54°F primary air to accommodate the higher internal latent loads, a very large portion of the space sensible cooling load is now handled by the primary airflow and not the chilled beams. At low load conditions, this may result in little if any of the cooling load being handled by the cooling coil within the chilled beam. A serious problem occurs when the primary air over-cools the space. In VAV systems these conditions are typically addressed by running parasitic reheat in the VAV boxes but chilled beams are not designed nor intended to reheat during the cooling season.

Under such conditions a significant rise in the primary cooling coil temperature might be considered to place sensible cooling load back on the chilled beams, but humidity control is lost and condensation may occur. Reducing the primary airflow without reducing the supply dew point would have the same negative result.

Optimizing chilled beam primary airflow by reducing the supply air dew point

Fortunately the many benefits offered by active chilled beams can be recognized even for buildings located in humid environments provided that the primary airstream can be efficiently conditioned to a low enough dew point. For example, if the same Washington DC office building is designed to utilize a primary air system capable of supplying air at a condition of say 64°F but at a 45°F dew point (44 grains), with all other aspects of the design remaining the same, the airflow required is cut in half.

124,800 BTU/hr = Primary airflow * .68 * (76 grains – 44 grains)
Primary airflow required = 5,735 cfm

This one important change to the overall system design now allows the chilled beam system approach to be effectively employed in markets where the ambient humidity is high. Rather than requiring a primary airflow that is 250% of the 5,000 cfm ventilation air requirement, the flow is cut to only 15% (5,735/5,000) more than that required for ventilation purposes.

Most facilities considering a chilled beam approach are putting a high value on energy efficiency, indoor air quality (IAQ) and the comfort of the building occupants. Many are also seeking "green building" certifications like the Leadership in Energy and Environmental Design[5] (LEED) program. As a result, these facilities are often designed to include more outdoor air than the minimum recommended by ASHRAE Standard 62[6]. LEED program credits are also provided for exceeding the minimum ventilation requirements. Therefore, designers considering the use of chilled beam systems (whether passive or active) in locations with peak cooling season design conditions above about a 64°F dew point (90 grains), should incorporate energy efficient primary air systems with the capability of operating at or near 100% outdoor air while simultaneously delivering air at dew points well below that associated with traditional chilled water systems (i.e. 54°F). The system should ideally have a dehumidification mode for unoccupied hours and provide a very high level of total energy recovery during both the cooling and heating seasons.

3. PINNACLE SYSTEM: APPLYING CHILLED BEAMS IN HUMID CLIMATES

The SEMCO Pinnacle System (PVS) incorporates all of these capabilities and is therefore an ideal chilled beam primary air system for applications in humid climates. The Pinnacle system allows outdoor air streams to be dehumidified to low dew points; levels unattainable with conventional cooling approaches. This enhanced dehumidification capacity is achieved without the heated regeneration source required by "active" desiccant based dehumidification systems. The PVS approach incorporates effective total energy recovery which, when combined with the added dew point depression provide by the passive dehumidification wheel, minimizes cooling requirements and energy consumption while simultaneously delivering primary air to the beams at the temperature and humidity level needed for optimum system performance. **Figure 4** shows a typical cooling mode condition demonstrating how the PVS system functions. The supply air stream is cooled and dehumidified by passing it through a dry and cool zone of the total energy recovery wheel which has been rotated through and reached near equilibrium with the relatively cool, dry exhaust air stream leaving the "passive" dehumidification wheel. The air stream is then further cooled and dehumidified by passing it through the cooling coil. Before it is supplied to the space, the supply air is further dehumidified and moderately reheated by passing it through a warm and dry zone of the passive dehumidification wheel, which has been rotated through and reached near equilibrium with the warm, dry exhaust air stream leaving the conditioned space.

The PVS system can also be operated with minimal or no outdoor air during unoccupied periods to provide the humidity control necessary to avoid condensation on the chilled beams. The passive dehumidification wheel within the system provides most of the dehumidification capacity needed, minimizing chilled water requirements, and is cycled to operate only when dehumidification is needed. As a result, controlling the humidity within unoccupied facilities is both practical and energy efficient. This "unoccupied mode" of the PVS system is shown by the schematic labeled **Figure 5**.

The PVS system is also a very effective heating season primary air system. It can be controlled to optimize both temperature and humidity recovery (humidification), to the extent necessary, during the heating season by increasing the passive dehumidification wheel speed from a low of .25 RPM (dehumidification mode) to about 5 rpm. A sample flow schematic showing the operation of the PVS system during the heating mode is presented in **Figure 6**. More specific details concerning the benefits and operation of this Pinnacle system along with a performance comparison with other dedicated outdoor air systems can be found in the SEMCO technical paper entitled "Pinnacle Ventilation System Integrates Total Energy Recovery, Conventional Cooling and a Novel 'Passive' Dehumidification Wheel to Mitigate the Energy, Humidity Control and First Cost Concerns Often Raised when Designing for ASHRAE Standard 62-1999 Compliance".

Advantage of Pinnacle over other potential chilled beam primary air systems

To quantify the advantage of using Pinnacle systems in conjunction with chilled beams systems located in humid climates; this paper analyzes two common building types (office and school facilities). The analyses contrast a traditional VAV system employing total energy recovery with three chilled beam designs; one using a single wheel energy recovery system (**Figure 7**), one using a dual wheel (sometimes called a twin wheel) energy recovery system (**Figure 8**) and the third using a Pinnacle system.

FIGURE 4: Typical Pinnacle cooling season performance

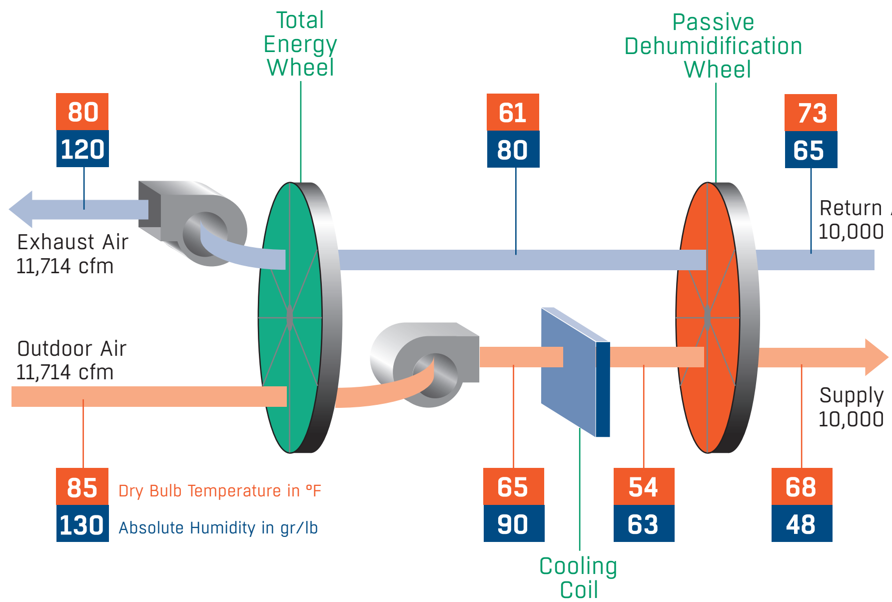
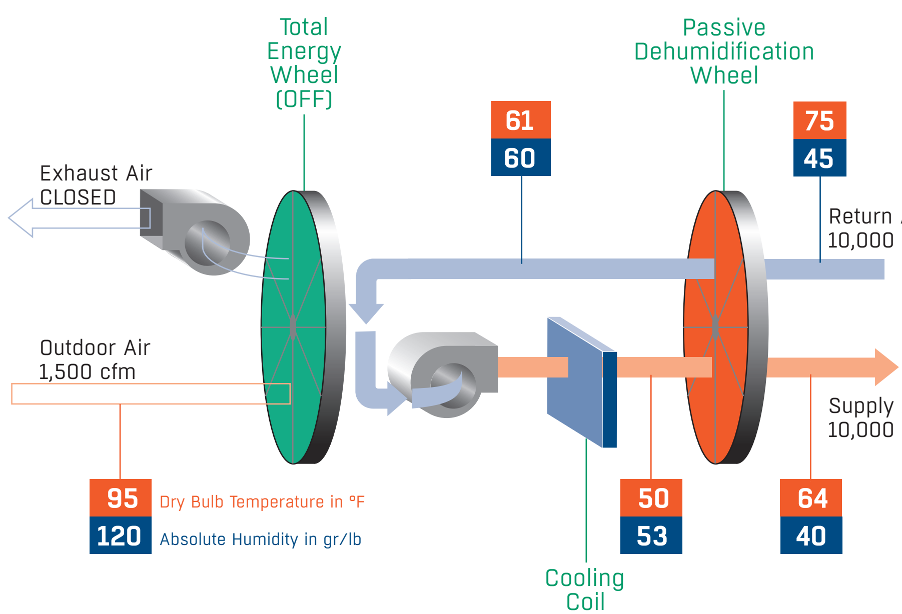


FIGURE 5: Pinnacle system in the unoccupied mode



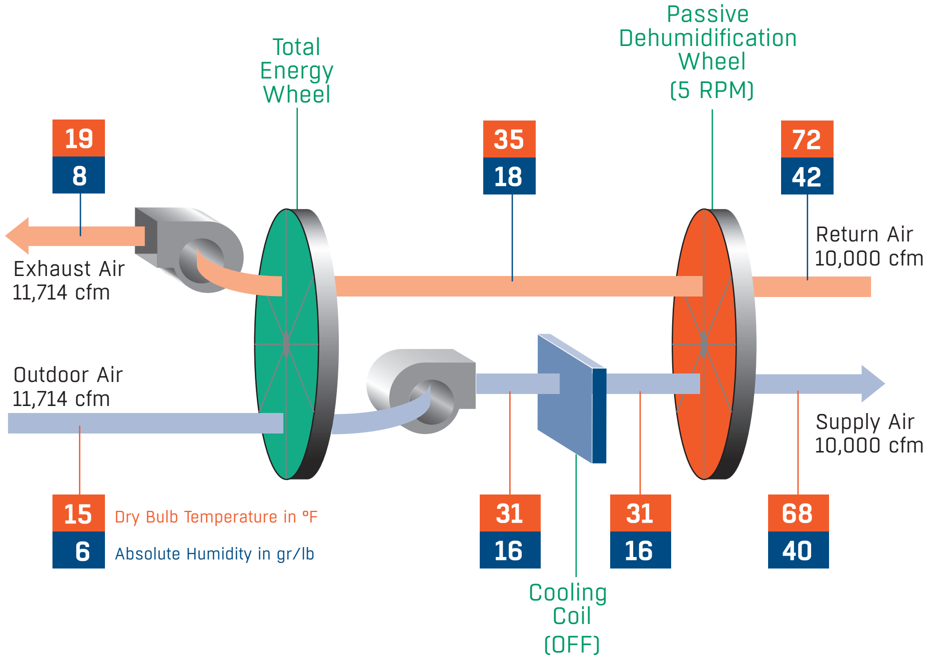


FIGURE 6: Pinnacle system in the heating mode

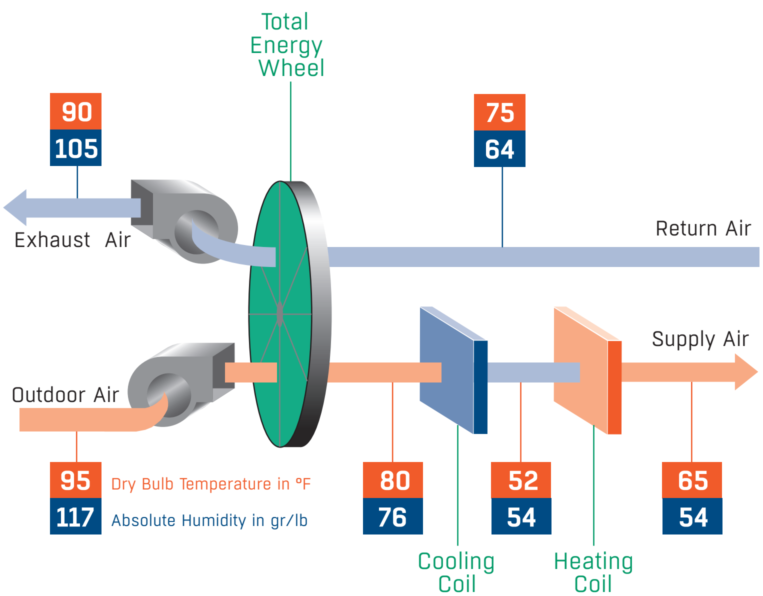


FIGURE 7: Single total energy wheel with reheat

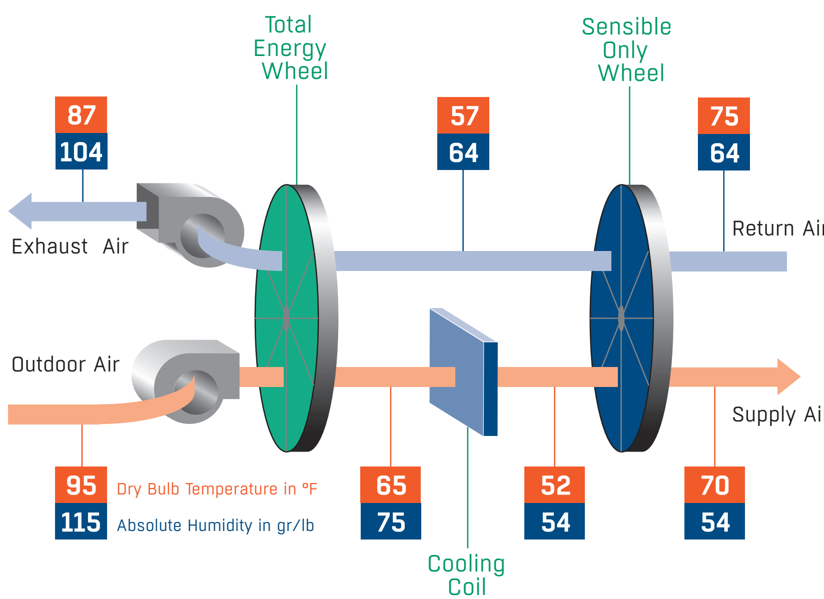


FIGURE 8: SEMCO EPD dual total energy wheel with free reheat

Although the Pinnacle system has numerous important advantages over the single wheel and dual wheel approach, including higher energy efficiency, an effective unoccupied mode and reduced chiller capacity, the main benefit over these systems is the ability to provide much lower supply air dew points at a given chilled water temperature.

Put simply, the supply air dew point capability of the single wheel and dual wheel approaches shown as **Figures 7 and 8** is limited by the leaving coil temperature. This is not the case for a Pinnacle system. Contrasting the performance shown in **Figures 4, 7 and 8** it is clear to see that for the same leaving coil temperature, the supply air humidity content changes significantly. Assuming the desire to maintain the space at 75°F and 50% relative humidity (65 grains), the 45 grains provided by the Pinnacle would require less than half the primary airflow needed by the other two approaches delivering air at say 56 grains. Other benefits are highlighted by the office and school facilities analyses included in section 4.

Psychrometric comparison of primary air systems

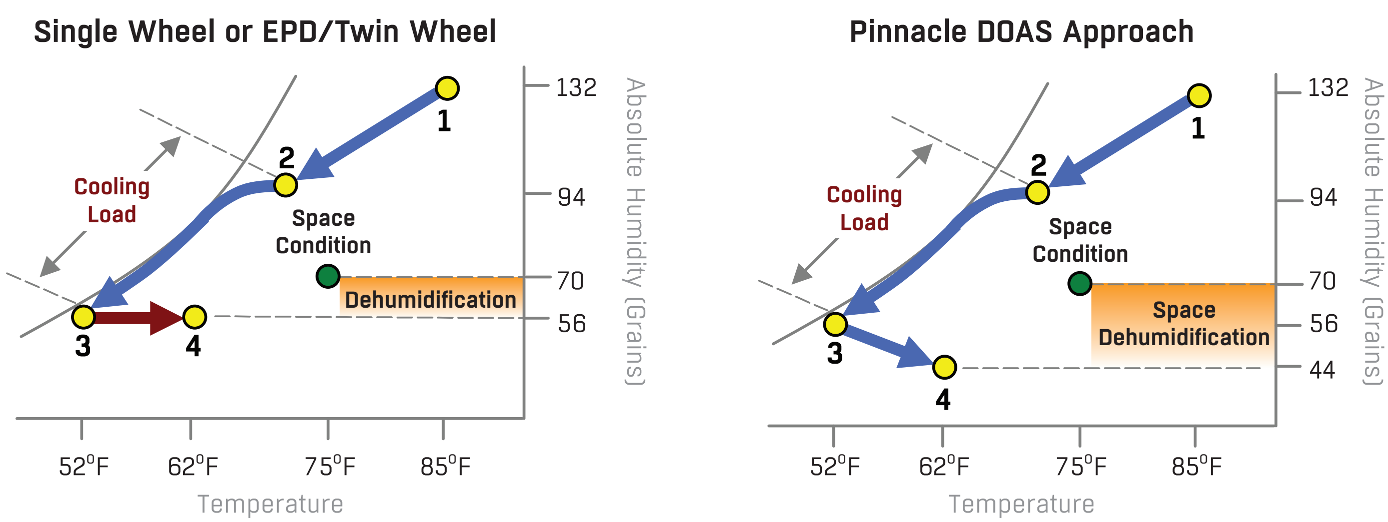
Figure 9 shows the performance advantage of the Pinnacle system in a psychrometric format. As shown, both systems have effective outdoor air preconditioning accomplished by the total energy wheel. Point 1 is the outdoor air condition being cooled and dehumidified by the recovery wheel to condition 2, which then enters the cooling coil. Both systems then cool and dehumidify the primary air further by the cooling coil, to condition 3. For the benefit of simplicity, the example assumes that approximately the same cooling input is used and the leaving coil temperatures are the same. That is where the similarity ends.

As shown for the single wheel/dual wheel approach, the supply air humidity is limited by the dew point leaving the cooling coil. In sharp contrast, the Pinnacle further dehumidifies the air leaving the coil using air exhausted from the space, to deliver a much lower dew point to the space. As represented by the orange region, the dehumidification capacity is significantly increased allowing approximately half the airflow required by the single/dual wheel approach to be used by the Pinnacle system.

Highlighting the impact of low dew point air on primary airflow

Considering the same 20,000 square foot office building discussed in section 2 and reflected in **Figures 3 and 10** was prepared to show the difference in primary airflow required by a conventional system delivering air at a typical 55°F dew point and a Pinnacle system operated to deliver air at a dew point of 45°F.

FIGURE 9: Psychrometric comparison of single wheel/dual wheel with a Pinnacle system



The yellow line represents the outdoor air volume required to satisfy the ventilation air requirement for the office building in accordance with ASHRAE 62. As previously mentioned, it is highly desirable to operate the chilled beam systems with the primary airflow at or only slightly higher than this ventilation airflow requirement. As shown, the Pinnacle system allows this to be achieved, even in humid climates.

In contrast, the conventional approach can only accommodate the internal latent load using the ventilation air quantity in very dry climates (Stockholm). In all other markets the primary airflow needs to be much greater than the ventilation air requirement, nearly three times as much in the more humid environments.

As discussed later in sections, reducing the primary airflow with the Pinnacle approach not only reduces energy consumption, but allows the chilled beam/ Pinnacle system to have a competitive installed cost with even conventional VAV and fan coil systems.

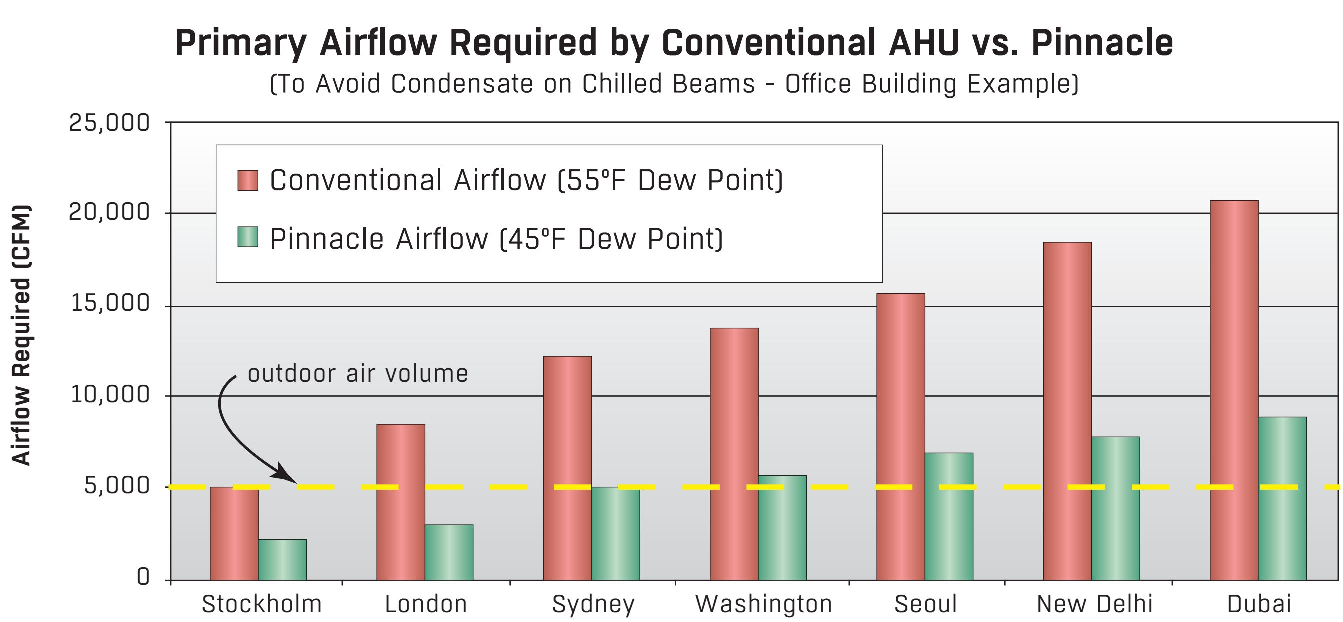
4. COMPARING SYSTEM OPERATION EFFICIENCIES

To contrast the energy efficiency (and other metrics) of a traditional VAV system and an active chilled beam approach, two sample facilities were evaluated. The first facility investigated was a single story office building. An office was chosen for analysis since it is a facility that has a relatively low occupant density and thereby less internal latent load and less ventilation air required per floor area than most other facilities. Offices tend to be sensibly driven, having a high sensible heat ratio (SHR) and are generally thought to be good applications for conventional variable air volume (VAV) systems.

The second facility investigated is a wing of school classrooms. This application was chosen since, compared to the office, the higher occupant density requires far more ventilation per floor area. It also, as a result, has a higher internal latent load, lower SHR and therefore makes controlling indoor humidity conditions more difficult. Many schools designed with VAV systems require parasitic reheat at the VAV box during low load conditions to avoid over-cooling if the appropriate quantity of ventilation air is provided.

Although many VAV systems are designed without the benefit of total energy recovery preconditioning, this analysis assumes that total energy recovery is included

FIGURE 10: Comparing primary airflow required by conventional and Pinnacle systems

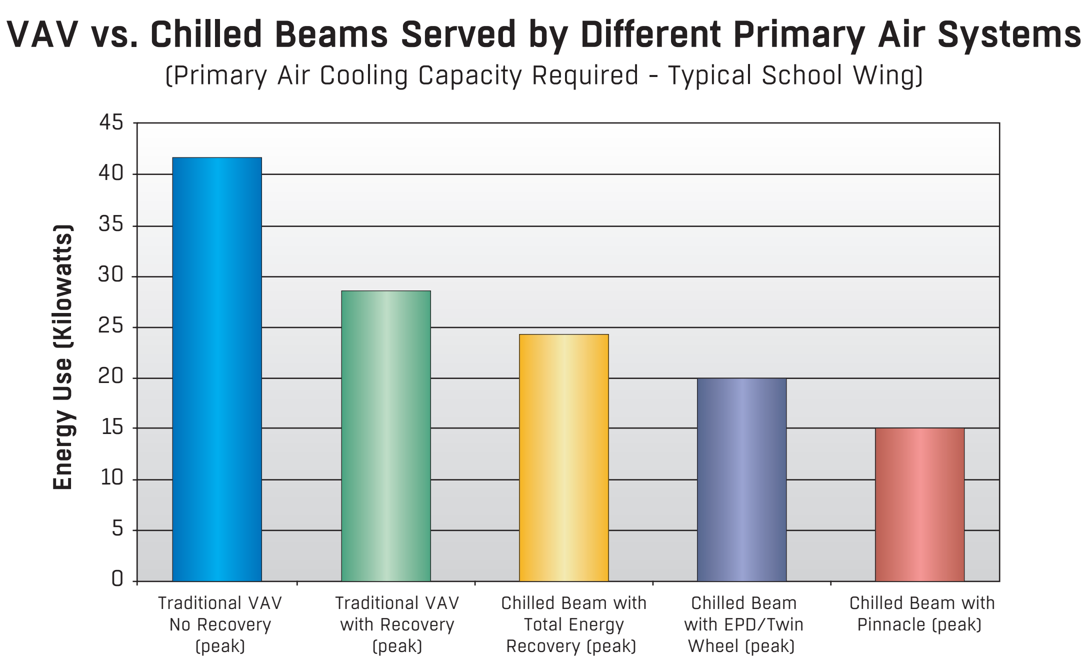


in all system approaches evaluated. As shown by **Figure 11**, it would have been inappropriate to omit the use of total energy recovery for the traditional VAV approach since the impact on energy consumption and the chiller/boiler capacity required is significant.

The analyses assume that the buildings are well constructed, tight buildings, use high efficiency lighting and design practices as per ASHRAE Standard 90.1. ASHRAE Standard 62 default values are used for occupancy levels. It is assumed that the facilities are located in Atlanta, Georgia. All other assumptions are outlined within the application summary documentation included in the appendix section.

Sensible loads for these buildings were estimated using the conventional methodology like that incorporated into most commercially available building simulation software programs. The latent loads were analyzed in detail, using ASHRAE recommendations outlined in the Humidity Control Design Guide.

FIGURE 11: Primary air-cooling capacity required with VAV shown with and without total recovery



EXAMPLE 1 - OFFICE BUILDING: RESULTS & CONCLUSIONS

The Atlanta based single story office building investigated was assumed to be 8,500 square feet, with 85 occupants. Space conditions during the cooling season were maintained at 75°F and 52% relative humidity (70 grains). The 2% peak humidity design condition of 85°F and 74% RH (132 grains) was obtained from the ASHRAE Fundamentals[6] and the part load condition of 77°F and 80% RH (109 grains) was selected from the Atlanta ASHRAE weather database.

The internal sensible loads at peak and part load conditions were estimated to be 165,520 BTU/hr. and 107,500 BTU/hr. respectively. The corresponding internal latent loads were determined to be 49,060 BTU/hr. and 41,060 BTU/hr. using ASHRAE Humidity Design Guide recommendations.

One important advantage of the chilled beam approach is that the space humidity can be maintained at the desired level throughout the cooling season. In contrast, the modeling confirmed that the VAV approach could not consistently maintain the space humidity set point, without increasing the supply airflow and employing substantial parasitic reheat. This energy intensive approach was not considered viable for this analysis.

In practice, discomfort associated with the elevated humidity levels that can exist with VAV systems is addressed by lowering the space temperature set point, as documented by the work of Berglund (Fischer, 2003[7]) which provided the basis for the ASHRAE 55 comfort standard.

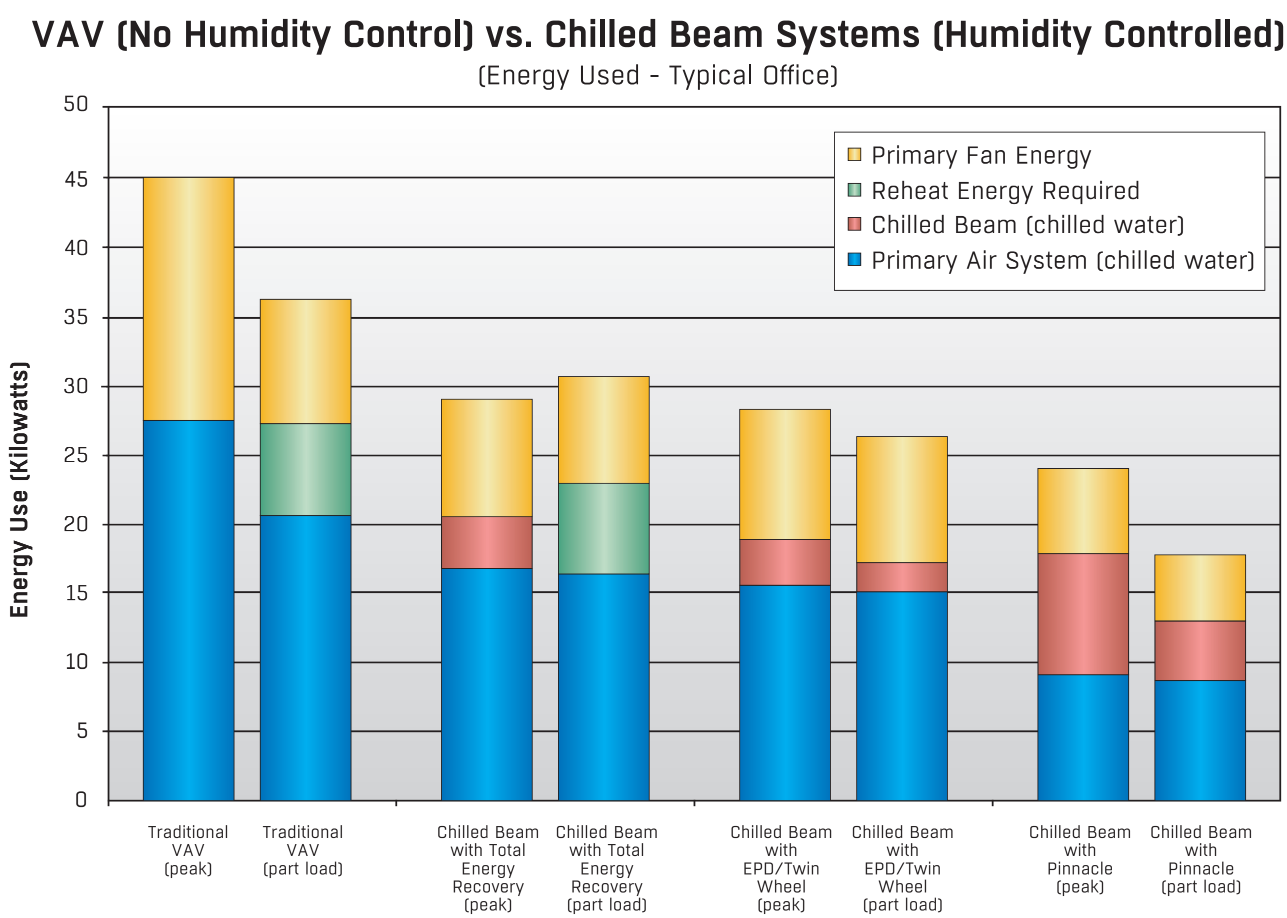
Therefore, in an attempt to compare all systems at a comparable comfort level, the space temperature set point for the VAV approach was modeled two degrees lower than used for the chilled beam systems (73°F vs. 75°F). Data is also provided in the appendix section (**Figure A1**) for a comparison between all systems using the same set point so that the impact of this "comfort correction" can be observed.

Figure 12 provides a graphical summary of the total energy consumed, shown in kilowatts (KW) for the HVAC systems analyzed, as well as sub-categories for fan energy, air handing unit chilled water consumption, chilled beam chilled water consumption and reheat energy, if required. Details for this analysis are shown in tabular form by **Figure 13** and all other assumptions are included within the appendix section of this document.

Energy consumption comparison at peak and part load cooling conditions

The combination of chilled beams and the Pinnacle primary system showed a substantial reduction in energy consumed at all conditions investigated. As shown in **Figures 12 and 13**, the chilled beam/Pinnacle system operates with 48% less energy than the VAV alternative at the peak cooling condition and 52% less at the more frequent part load condition. If a set point of 75°F is used for the VAV system as well (no

FIGURE 12: Modeled results comparing energy consumed by the VAV and chilled beam approaches



compensation for elevated space humidity) the energy reductions remain substantial with the chilled beam/ Pinnacle approach; estimated at 38% and 44% for peak and part load respectively.

When comparing the three chilled beam approaches, using different primary air systems, considerable energy consumption differences exist as well. During part load conditions, the Pinnacle system approach is 41% and 32% more efficient than the single total energy wheel and dual wheel, EPD system respectively. At peak load conditions the savings are 19% and 17%.

The energy advantages associated with the Pinnacle system result from the ability to use significantly less primary airflow. In this analysis, the Pinnacle system needed only 3,000 cfm of primary airflow compared to 5,500 for the single wheel and dual wheel systems. The ability to deliver a much lower supply air dew point (48.5°F) compared to that delivered by the other chilled beam primary air systems (51°F), using the same chilled water temperature, allowed for the reduction in airflow and the corresponding savings in fan energy. The VAV system required 9,000 cfm at peak cooling and approximately 7,000 cfm at the part load condition. In addition to savings in fan energy (yellow bars in **Figure 12**), significantly less chilled water was required by the Pinnacle primary air system as a result of the lower airflow and the effectiveness of the passive dehumidification wheel (blue bars in **Figure 12**). Additionally, all of the necessary reheat energy required to avoid over-cooling low load spaces during the part load condition is provided by the second, passive dehumidification wheel. Unlike the single wheel and the VAV alternative, no parasitic energy is required for this purpose (green bars in **Figure 12**).

Indoor air quality (IAQ), occupant comfort and control effectiveness

A subtle yet very important advantage offered by the chilled beam/Pinnacle approach is that most of the space sensible cooling is being provided by the chilled beams (large red bars with Pinnacle in **Figure 12**), not the primary airflow. Placing most of the cooling load on the beams should be an important design objective. It allows for ideal space temperature control, even at very low load conditions and limits the risk of over-cooling spaces.

With the chilled beam/single wheel approach, the high airflow to the space, which has been cooled (in this case to 51°F) in order to satisfy the internal latent load can easily exceed the cooling capacity required by some spaces during low load conditions. Since space humidity must be controlled, the supply air temperature leaving the cooling coil must remain low, leaving parasitic reheat as the only option available to avoid over-cooling some spaces (small red bars in **Figure 12**). The EPD, dual wheel approach provides free reheat when needed, at the part load conditions analyzed, but at very low load conditions this system too may become problematic due to the relatively high primary airflow needed.

All chilled beam approaches have the advantage of delivering a constant supply of outdoor air to the individual zones, thereby optimizing IAQ while the VAV system will often under-ventilate spaces during low load conditions, if reheat is not utilized. By controlling space humidity under all conditions and distributing the air along the ceilings and walls, the occupant comfort level attained with chilled beams is very high.

Maintaining a high supply air exchange rate within the occupied space

A common misconception is that the low primary airflow associated with chilled beam systems may result in a “stagnant”, uncomfortable space due to low airflow. However, as a result of the chilled beam induction airflow, the supply airflow to the occupied spaces is actually higher than that provided by a typical VAV system, even at peak condition, despite the fact that the flow through the ductwork is much lower. For example, **Figure 13** shows that for our example, the peak VAV airflow is 9,000 cfm while the primary airflow delivered to the chilled beams by the Pinnacle unit is only 3,000 cfm. However, the airflow delivered to the spaces by the beams is 10,200 cfm, when the primary air and induced airflows are combined. As space loads are satisfied with the VAV system, the airflow to the space is reduced considerable, while the supply air to the space with chilled beams remains high.

FIGURE 13: Tabular summary of Office Example: Space humidity controlled with chilled beam approaches, not VAV approach

	Traditional VAV (with Total Energy Recovery)	Chilled Beams		
		Total Energy Recovery	EPD/Twin Wheel	Pinnacle
Airflows:	**Cubic Feet per Minute (CFM)**			
Primary supply airflow at peak cooling (includes outdoor air)	9,015	5,550	5,550	3,000
Primary supply airflow at part load cooling (includes outdoor air)	7,026	5,550	5,550	3,000
Airflow within space at peak cooling (approximate with chilled beams induced air)	9,015	10,947	10,947	10,217
Outdoor airflow (note 5)	2,320	2,000	2,000	2,000
Supply Conditions (peak)	**Design Parameters**			
Primary air temperature	56	54	62	62
Primary air humidity (grains)	61	57	57	46
Primary air duct humidity (RH%)	88%	92%	70%	58%
Primary air dew point	53	51	51	48.5
Space supply air temperature	57	61	61	60
Space supply air humidity (grains)	61	63	63	63
Estimated Energy Use (peak cooling)	**Kilowatts per Hour (KWH)**			
Primary air system	27.7	17.6	15.8	9.0
Chilled beam	n/a	3.0	3.0	9.3
Reheat energy required	0.0	0.0	0.0	0.0
Primary fan energy	17.3	8.3	9.6	5.2
Total	**45.0**	**28.8**	**28.3**	**23.4**
Estimated Energy Use (part load)	**Kilowatts per Hour (KWH)**			
Primary air system	20.7	17.0	14.8	8.2
Chilled beam	n/a	0.0	2.2	4.9
Reheat energy required	6.3	5.4	0.0	0.0
Primary fan energy	10.6	8.3	9.6	5.2
Total	**37.6**	**30.6**	**26.6**	**18.2**
Key System Design Issues	**Additional Design Benefits**			
Primary airflow duct diameter	41 inches	32 inches	32 inches	23 inches
Final filter area required (MERV 13)	23 sq ft	14 sq ft	14 sq ft	8 sq ft
Typical space sound levels	40-45 dB	38 dB	38 dB	35 dB
Desired ventilation to each space	Not necessarily	Yes	Yes	Yes
Humidity maintained	Not controlled	Yes with reheat	Yes	Yes
Estimated potential LEED 2.2 points	2 to 5	4 to 9	6 to 12	8 to 15

EXAMPLE 2 - SCHOOL CLASSROOM WING: RESULTS & CONCLUSIONS

The Atlanta based school facility investigated involved a wing with 10 classrooms housed in an 8,500 square foot traditional block, single story building occupied by 210 students and teachers. As with the office example, space conditions during the cooling season were maintained at 75°F and 52% relative humidity (70 grains). The 2% peak humidity design condition of 85°F and 74% RH (132 grains) was obtained from the ASHRAE Fundamentals[6] and the part load condition of 77°F and 80% RH (109 grains) was selected from the Atlanta ASHRAE weather database.

The internal sensible loads at peak and part load conditions were estimated to be 154,507 BTU/hr. and 93,690 BTU/hr. respectively. The corresponding internal latent loads were determined to be 64,100 BTU/hr. and 52,590 BTU/hr. using ASHRAE Humidity Design Guide recommendations.

Once again, in an attempt to assess all systems at a comparable comfort level, the space temperature set point for the VAV approach was modeled two degrees lower than used for the chilled beam systems (73°F vs. 75°F) (**Figure 14**). For comparison, summary data is also provided for the same school with the VAV operated to maintain the same humidity levels as maintained by the chilled beam systems (**Figure 16**). This requires substantial additional reheat energy and, for this reason, is seldom used in practice. The appendix section includes a comparison between all systems operated at the same 75°F space set point, allowing elevated space humidity levels for the VAV approach, so that the impact of this "comfort correction" can be observed (**Figure A2**).

Figure 14 provides a graphical summary of the energy consumption for this school example. Details for this analysis are shown in tabular form by **Figure 15**, with all other assumptions included within the appendix section of this document.

Energy consumption comparison at peak and part load cooling conditions

As for the previous office example, the combination of chilled beams and the Pinnacle primary system showed a substantial reduction in energy consumed at all conditions investigated when compared to the VAV alternative. As shown in **Figures 14 and 15**, the chilled beam/Pinnacle system operates with 37% less energy at the peak cooling condition and 33% less at the more frequent part load condition. If a set point of 75°F is used for the VAV system as well (no compensation for elevated space humidity) the energy reductions remain substantial with the chilled beam/Pinnacle approach; estimated at 31% and 24% for peak and part load respectively.

If the VAV system was operated with reheat to control humidity to the desired set point as done by the chilled beam approaches, and operated to maintain the same 75°F temperature set point, then the energy reduction associated with the chilled beam/Pinnacle approach was estimated at 41% and 65% for peak and part load respectively. This is shown graphically as **Figure 16**. The tabular data to support this graphic is included in the appendix section of this document (**Figure A3**).

When comparing the three chilled beam approaches, using different primary air systems, considerable energy consumption differences exist as well. During part load conditions, the Pinnacle system approach is 53% and 29% more efficient than the single total energy wheel and dual wheel, EPD system respectively. At peak load conditions the savings are 17% and 11%.

Once again, the energy advantages associated with the Pinnacle system result from the ability to use far less primary airflow. In this analysis, the Pinnacle system needed only 3,650 cfm of primary airflow compared to 6,284 for the single wheel and dual wheel systems. The ability to deliver a much lower supply air dew point (44°F) compared to that delivered by the other chilled beam primary air systems (51°F), allowed for the reduction in airflow and the corresponding savings in fan energy. The VAV system required 8,415 cfm at peak cooling and approximately 5,828 cfm at the part load condition.

In addition to savings in fan energy (yellow bars in **Figure 14**), far less chilled water was required by the Pinnacle primary air system as a result of the lower airflow and the effectiveness of the passive dehumidification wheel (blue bars in **Figure 14**). All of the necessary reheat energy required to avoid over-cooling low load spaces during the part load condition is provided by the second, passive dehumidification wheel so, unlike the single wheel and the VAV alternative, no parasitic energy is required for this purpose (green bars in **Figure 14**).

FIGURE 14: Modeled results comparing energy consumed by the VAV and chilled beam approaches

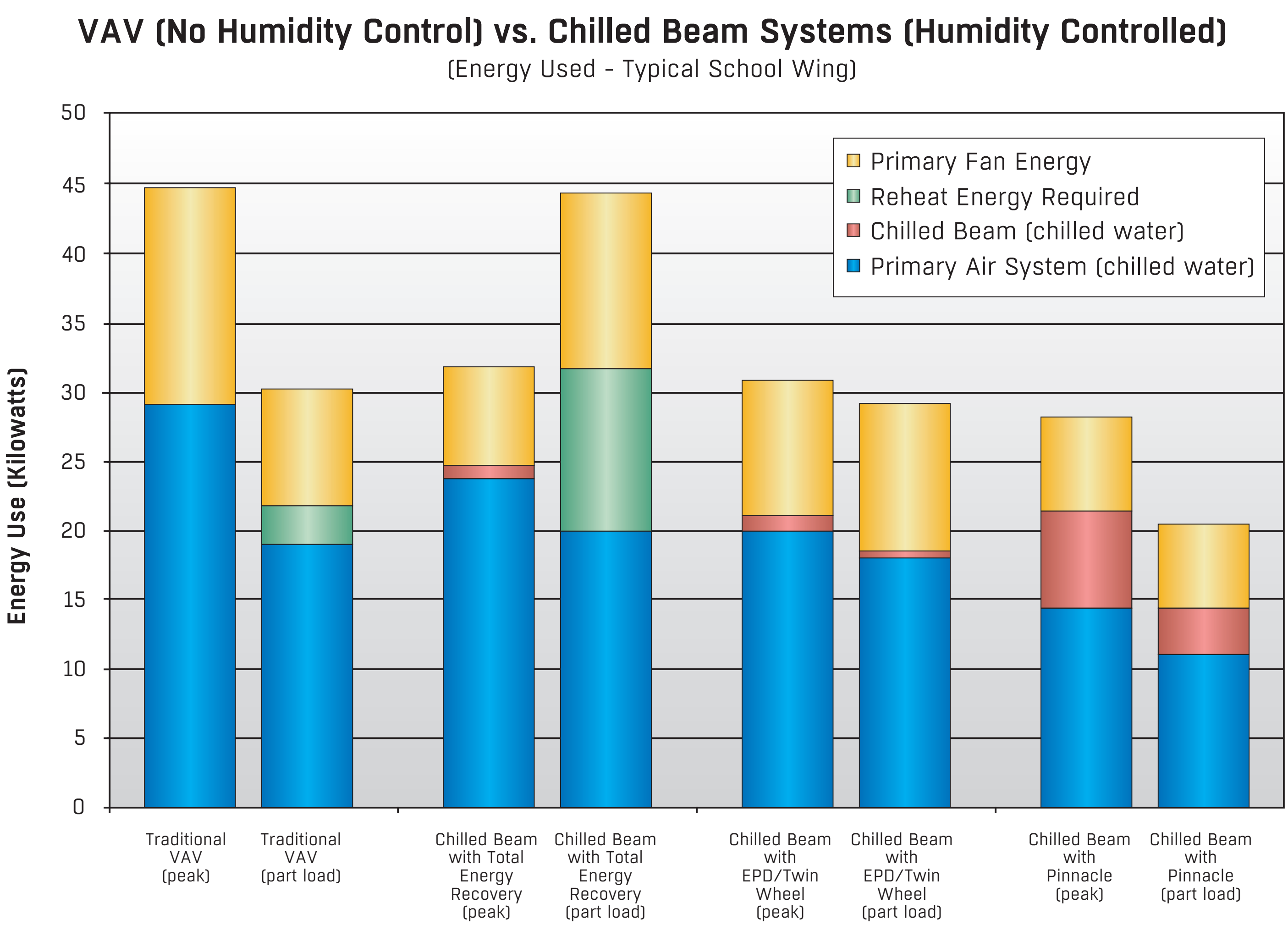


FIGURE 15: Tabular summary of School Example: Space humidity controlled with chilled beam approaches, not VAV approach

	Traditional VAV (with Total Energy Recovery)	Chilled Beams		
		Total Energy Recovery	EPD/Twin Wheel	Pinnacle
Airflows:	**Cubic Feet per Minute (CFM)**			
Primary supply airflow at peak cooling (includes outdoor air)	8,415	6,284	6,284	3,650
Primary supply airflow at part load cooling (includes outdoor air)	5,828	6,284	6,284	3,650
Airflow within space at peak cooling (approximate with chilled beams induced air)	8,415	11,005	11,005	10,219
Outdoor airflow (note 5)	4,290	3,650	3,650	3,650
Supply Conditions (peak)	**Design Parameters**			
Primary air temperature	56	53	53 - 62	62
Primary air humidity (grains)	61	55	55	44
Primary air duct humidity (RH%)	88%	89%	89% - 60%	51%
Primary air dew point	53	51	51	44
Space supply air temperature	56	62	62	61
Space supply air humidity (grains)	61	61.4	61.4	60.8
Estimated Energy Use (peak cooling)	**Kilowatts per Hour (KWH)**			
Primary air system	28.7	23.8	20.0	14.1
Chilled beam	n/a	0.9	0.9	7.7
Reheat energy required	0.0	0.0	0.0	0.0
Primary fan energy	16.1	9.4	10.8	6.3
Total	**44.8**	**34.1**	**31.7**	**28.1**
Estimated Energy Use (part load)	**Kilowatts per Hour (KWH)**			
Primary air system	18.5	20.3	17.5	11.0
Chilled beam	n/a	0.0	0.4	3.2
Reheat energy required	3.9	14.3	0.0	0.0
Primary fan energy	8.1	9.4	10.8	6.3
Total	**30.5**	**44.0**	**28.8**	**20.5**
Key System Design Issues	**Additional Design Benefits**			
Primary airflow duct diameter	39 inches	34 inches	34 inches	26 inches
Final filter area required (MERV 13)	21 sq ft	16 sq ft	14 sq ft	9 sq ft
Typical space sound levels	40-45 dB	38 dB	38 dB	35 dB
Desired ventilation to each space	Not necessarily	Yes	Yes	Yes
Humidity maintained	Not controlled	Yes with reheat	Yes	Yes
Estimated potential LEED 2.2 points	2 to 5	4 to 9	6 to 12	8 to 15

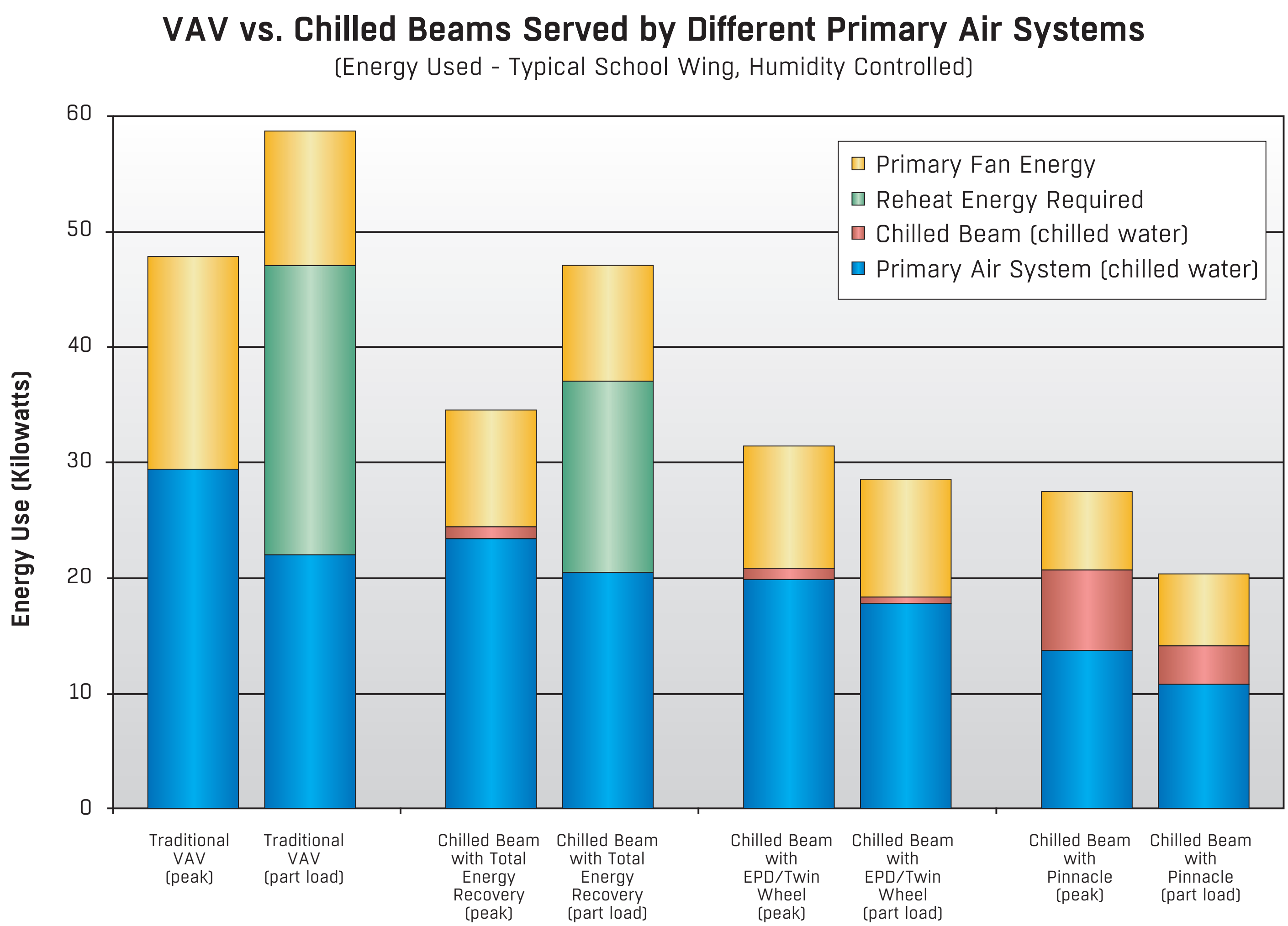


FIGURE 16: Results comparing energy consumed with VAV and chilled beams, all with humidity control

Indoor air quality (IAQ), occupant comfort and control effectiveness

Once again, the chilled beam/Pinnacle approach has the important advantage of satisfying most of the space sensible cooling load with the chilled beams (large red bars with Pinnacle in **Figures 14 and 16**), and not the primary airflow. The high ventilation load and internal latent load per square foot of a school facility results in high primary airflows at low supply air temperatures with the single wheel and EPD systems. As a result, over-cooling of spaces at low load conditions will be a common problem with both VAV and chilled beams using these two primary air system approaches if reheat is not utilized. This is shown clearly by the large green bars in **Figures 12 and 14**.

The EPD approach eliminates the need for reheat energy in most cases, but due to the high primary airflow, places very little load on the chilled beams. This may present comfort control problems at very light load conditions, where a teacher is in an otherwise unoccupied room, grading papers, for example.

In school facilities the advantage of delivering a constant supply of outdoor air to the individual zones offered by the chilled beam approaches is particularly important. This ensures that the relatively high outdoor air ventilation rate is delivered to each classroom, thereby optimizing IAQ. VAV systems without reheat will often under-ventilate school classrooms during low load conditions.

By distributing the supply air along the ceilings and walls, drafts and dumping of cold air is avoided; both common problems with VAV systems in school classroom.

Background Classroom Noise (ANSI/ASA Standard S12.60-2002)

A particularly important design criterion for schools is to maintain a desirable teaching environment by controlling background noise in classroom areas. Much industry controversy has surrounded the ANSI standard recommending background noise levels to be maintained below 35 decibels.

With chilled beams, a higher airflow rate than typically delivered by a VAV system can be introduced into the classroom space while maintaining the associated noise generation below 35 decibels. In the example summarized in **Figure 15**, the airflow delivered by the chilled beam/Pinnacle approach is 21% higher than the VAV airflow at peak cooling conditions and 75% higher at part load. Delivering this high air change rate without drafts provides a high degree of comfort.

5. COMPARISON OF VARIOUS SYSTEMS' OPERATING COST

A detailed building simulation to compare the relative cost of HVAC system operation was completed for all systems considered for both the office and classroom wing examples. Fan coils were also added to this analysis for comparative purposes. Indoor space conditions during occupied hours were assumed to be 75°F and 50% relative humidity during the cooling season and 70°F at 30% relative humidity during the heating season. The modeling considered unoccupied periods with thermostat setback conditions of 78°F and 65°F for the cooling and heating seasons respectively. It also incorporated enthalpy based economizer operation. Atlanta weather data was utilized for this comparison. The cost of electricity used was $.08/KWH with a gas cost of $10/million BTUs.

The VAV and fan coil systems were not modeled to maintain the cooling season space humidity set point at all conditions. This would have required parasitic reheat to accomplish and would have resulted in a significant energy penalty for these systems. All of the chilled beam approaches were modeled to consistently maintain the desired cooling season space humidity levels. As previously mentioned, all systems (VAV, fan coil and chilled beams) benefited from the incorporation of total energy recovery.

Figure 17 provides the estimated total annual energy cost associated with the HVAC systems considered for both building types. This energy cost is also shown on a per square foot basis along with the percent increase over that required by the best, most effective system (chilled beams with Pinnacle system).

The results of this analysis show that a significant reduction in annual energy costs can be recognized by employing chilled beam systems over the baseline VAV and fan coil options. This was especially true when combining the Pinnacle system with the chilled beams.

As shown by **Figure 17**, the VAV and fan coil systems were projected to cost 94% and 71% more to operate than the chilled beam/ Pinnacle approach for the small, single story office building modeled. For the school classroom wing, the VAV and fan coil systems were projected to cost 58% and 39% more. Once again, had the VAV and fan coil systems been modeled to control cooling season space humidity under all conditions or if the space thermostat settings had been lowered in an attempt to model comparable comfort, the cost of operation would have been much higher.

When considering the total energy consumed within these facilities, the cost of lighting, computers, outlets etc. must be considered. These additional loads for the office and school facilities were estimated to be $.51 and $.44/square foot respectively. As a result, designing the office building to include chilled beams and the Pinnacle system was projected to reduce the overall energy consumed by the building (HVAC, lighting, computers, etc.) by 33% compared to the VAV approach incorporating total energy recovery. The same comparison made for the school facility projected a 25% reduction in total energy use over the baseline VAV system. If the baseline VAV system had not included total energy recovery, the energy reductions would have been far greater. ASHRAE 90.1 does not require recovery for most VAV systems so LEED points would be determined by a comparison between the Pinnacle/Chilled beam system and the baseline VAV without recovery.

FIGURE 17: Results of energy cost modeling completed for various HVAC approaches serving an office and school facility

Office Example					
HVAC Operating Costs	**VAV Total Recovery**	**Fan Coils Total Recovery**	**CB Single Wheel**	**CB Dual Wheel**	**CB Pinnacle**
Total Annual HVAC Cost	$9,469	$8,351	$6,025	$5,619	$4,883
Square Footage	8,500	8,500	8,500	8,500	8,500
HVAC Energy Cost / Sq Ft	$1.11	$0.98	$0.71	$0.66	$0.57
Percent Increase vs. Best	94%	71%	23%	15%	Best
School Example					
HVAC Operating Costs	**VAV Total Recovery**	**Fan Coils Total Recovery**	**CB Single Wheel**	**CB Dual Wheel**	**CB Pinnacle**
Total Annual HVAC Cost	$8,112	$7,126	$6,214	$5,935	$5,137
Square Footage	8,500	8,500	8,500	8,500	8,500
HVAC Energy Cost / Sq Ft	$0.95	$0.84	$0.73	$0.70	$0.60
Percent Increase vs. Best	58%	39%	21%	16%	Best

NOTES:
(1) VAV and fan coil units do not include cooling season humidity (costs would be higher), chilled beam do
(2) Atlanta weather data used, electricity at $.08/KWHH and gas at $10/MMBtu., conventional building simulation modeling used
(3) Office operated in occupied mode 12 hrs/day, 7 days/wk, school operated in occupied mode 12 hrs/day, 6 days/wk
(4) Cooling season space conditions of 75°F and 50% RH with 78°F setback, heating season at 70°F and 30% RH with a 65°F setback

FIGURE 18: Results of construction cost analyses completed for various HVAC approaches serving an office and school facility

Office Example					
Equipment	VAV Total Recovery	Fan Coils Total Recovery	CB Single Wheel	CB Dual Wheel	CB Pinnacle
AHUs, Fans and Installation	$41,186	$40,150	$36,350	$47,350	$40,500
HVAC Electrical	$18,200	$15,500	$7,500	$8,000	$5,250
DDC Controls and Valves	$27,000	$14,400	$12,240	$12,240	$12,240
Chiller Tower and Boiler	$30,500	$23,500	$23,500	$21,200	$18,200
Ductwork and Installation	$21,250	$14,875	$12,644	$12,644	$9,031
Piping and Installation	$9,800	$50,585	$36,050	$35,350	$32,900
Fan Coil Units	-	$10,800	-	-	-
VAV Boxes	$12,360	-	-	-	-
Chilled Beams	-	-	$19,585	$19,585	$19,585
Dampers and Diffusers	$10,560	$10,560	$3,520	$3,520	$3,520
Other Installation/markup	$16,586	$16,091	$16,094	$16,349	$15,789
Total Cost	**$187,441**	**$196,461**	**$167,483**	**$176,238**	**$157,016**
Square Footage	**8,500**	**8,500**	**8,500**	**8,500**	**8,500**
Cost / Sq Ft	**$22.10**	**$23.10**	**$19.70**	**$20.70**	**$18.50**
School Example					
Equipment	VAV Total Recovery	Fan Coils Total Recovery	CB Single Wheel	CB Dual Wheel	CB Pinnacle
AHUs, Fans and Installation	$45,071	$44,000	$40,200	$52,850	$43,675
HVAC Electrical	$19,250	$17,500	$7,750	$8,250	$4,950
DDC Controls and Valves	$33,750	$18,000	$14,240	$14,240	$14,240
Chiller Tower and Boiler	$31,500	$27,500	$27,500	$23,200	$21,200
Ductwork and Installation	$19,763	$16,363	$13,908	$13,908	$7,907
Piping and Installation	$10,150	$52,685	$38,150	$36,750	$34,650
Fan Coil Units	-	$13,500	-	-	-
VAV Boxes	$15,450	-	-	-	-
Chilled Beams	-	-	$21,762	$21,762	$21,762
Dampers and Diffusers	$11,400	$11,400	$3,800	$3,800	$3,800
Other Installation/markup	$19,015	$18,478	$17,800	$18,024	$17,347
Total Cost	**$205,348**	**$219,426**	**$185,110**	**$192,784**	**$169,530**
Square Footage	**8,500**	**8,500**	**8,500**	**8,500**	**8,500**
Cost / Sq Ft	**$24.20**	**$25.80**	**$21.80**	**$22.70**	**$19.90**

6. INSTALLATION COST COMPARISON OF SYSTEMS INVESTIGATED

The improved energy efficiency and comfort provided by the chilled beam approach may not be considered by the end user if the first cost premium to install the technology is excessive. Therefore, in an attempt to investigate the approximate installed cost of each system considered, numerous contractor supplied project estimates were evaluated along with actual equipment costs to create the data presented as **Figure 18**. This figure accounts for all major expenses associated with installing each system. A list of the assumptions used to create this figure is included within the appendix section.

Unexpected findings resulted from this analysis. For both building types, the estimated cost to install the chilled beam/Pinnacle combination was found to be less than that associated with the VAV, fan coil or other chilled beam approaches. As expected, a careful review of the data showed that the cost of both the chilled beam components and the Pinnacle system, on a airflow (CFM) basis was considerable more expensive than the other primary air handling units and VAV boxes or fan coil units. However, significant savings in other areas more than compensated for these cost premiums.

The main contributor to the chilled beam/Pinnacle system cost advantage is the impact of the significantly reduced primary airflow serving the chilled beams made possible by the low supply air dew point capability of the Pinnacle system. First, the reduced airflow allows for the higher performing Pinnacle system to be cost competitive with the much larger VAV air handling unit and other conventional systems serving the fan coils and chilled beam systems. Secondly, the size of the ductwork required is greatly reduced. Finally the greater system efficiency requires less cooling capacity which reduces the size of the chiller, cooling tower, boiler and electrical service required compared to all other options investigated.

Significant cost savings are also contributed by the chilled beams since much of the cost associated with diffusers and dampers is eliminated when compared to the conventional approaches. Likewise the cost and complexity of the controls are substantially less with the chilled beam/Pinnacle approach than required by an effective VAV system.

Not factored into this analysis, but significant, is the cost benefit associated with a smaller mechanical room and shaft size made possible by the reduced primary airflow associated with the Pinnacle system. In addition, the ceiling space required by a chilled beam approach is typically 8 inches while the ductwork associated with the more conventional VAV system is typically 20 inches. This reduction in required ceiling mechanical space can be used to provide a higher ceiling height, a shorter building or, in some cases, allow for an additional floor to be included in a multistory building where height restrictions are in place.

7. ADDITIONAL IMPORTANT DESIGN ADVANTAGES OBSERVED

Substantial LEED Point Potential: The growing desire for energy efficient buildings and compliance with sustainable certification programs like LEED is providing increasing opportunities for chilled beam systems globally. In addition to the substantial economic benefits provided by the chilled beam/Pinnacle approach, the combined technologies can qualify for a significant number (up to 18) of LEED points as shown below. When integrated with a geothermal heat pump system, for example, approximately half of all the point required to reach LEED 2.2 Silver certification can be provided by the HVAC system alone. Approximately 16 can be reached using a chiller in lieu of the geothermal heat pump based on performance modeling completed as part of this investigation.

Potential LEED 2.2 Points	Credit Category
9	Potential 38.5% energy reduction with chilled beams/Pinnacle and geothermal heat pump system
2	30% water savings associated with reduced chiller capacity at the cooling tower
2	Increased ventilation air provided and measured by the Pinnacle system
3	Designing for, controlling and verifying thermal comfort with chilled beams and Pinnacle
2	Innovation and design points - chilled beam technology and Pinnacle dedicated outdoor air system

Reduced Maintenance: Filter maintenance and replacement cost both benefit from the reduced primary airflow associated with the chilled beam/Pinnacle approach. As shown by **Figure 15**, the MERV 13 filter area is reduced from 21 square feet with the VAV primary air system to only 9 square feet with Pinnacle.

Chilled beams do not require integral filtration and are simply vacuumed once every 2 to 3 years. Fan coils do require integral filtration at each unit and major manufacturers recommend changing these filters every one to two months. The cost of the filters and the labor involved to change them is a significant ongoing cost that is avoided with the chilled beam approach.

Very Low Noise Level: A properly designed active chilled beam system contributes essentially no detectable noise to the occupied space, producing sound power levels at or below 25 to 30 dB.

Aside from the improved satisfaction with the indoor environment, the low sound power levels provided by chilled beams also allow compliance with ANSI standard S12.60-2002 which requires background noise levels to be maintained below 35 decibels.

8. CONCLUSIONS AND OBSERVATIONS

The analyses presented by this document confirm the many advantages offered by the chilled beam technology over more traditional systems like variable air volume (VAV) and fan coil units, two of the most prevalent design methods used in the US HVAC market. Due to these advantages, chilled beams (both active and passive) are the prevalent HVAC approach now used in the most energy conscious "green building" markets globally, especially Scandinavia, the UK and central Europe.

The extended cooling season and greater cooling loads associated with the US market allows the chilled beam technology to offer even greater benefits than experienced in Europe, provided that the indoor humidity levels are carefully controlled. This is challenging in most parts of the US due to the high ambient humidity levels. High ambient humidity levels result in elevated internal latent loads which, if not handled effectively, can result in condensation on the chilled beams and/or over-cooling of the occupied space.

The findings of this investigation show that by combining the Pinnacle technology with chilled beams, energy efficiency is optimized and the same benefits experienced in the dry European climates can be recognized in far more humid climates like the US, without concern for condensation on the beams or over-cooling spaces.

The low dew point capability of the Pinnacle system allows for the use of a primary airflow that is approximately 33% of that required by the VAV system, at peak, and approximately 50% of that required by fan coil units or other more conventional systems serving chilled beams. The combination of this low dew point capability and increased energy efficiency resulted in the chilled beam/Pinnacle system having both the lowest cost of operation and a competitive cost of installation.

The chilled beam and Pinnacle system combination operated at substantially lower operating costs for both building types investigated, requiring 49% and 37% less energy than the VAV system incorporating efficient total energy recovery for the office and school examples respectively.

Unexpectedly, the reduced chiller, cooling tower and boiler capacity required coupled with the smaller ductwork (amongst other benefits) resulted in the estimated cost for the chilled beam/Pinnacle system being the lowest of all options investigated. First cost estimates completed suggested that the chilled beam/ Pinnacle system would cost 18% and 20% less to install than the VAV and fan coil systems respectively.

Numerous additional advantages were offered by the chilled beam/Pinnacle approach when compared with traditional HVAC systems. Some of the more important advantages included improved space humidity control, improved air distribution and IAQ, lower noise, reduced maintenance and simplified control complexity.

The predicted energy efficiency offered by this system approach provides an attractive economic life cycle investment for the end user but also qualifies for a large number of credits towards LEED and other "green building" certifications.

9. APPENDIX

List of assumptions used for office & school energy modeling

1) Load calculations are based on typical construction practices; lighting and other products are per ASHRAE 90.1 recommendations.
2) Latent loads are based upon recommendations made by ASHRAE in the Humidity Control Design Guide.
3) Chilled beam data based on the SEMCO IQIC beam, 57 degree chilled water, space dew point controlled at 2 degrees below this water temperature.
4) Design data and psychrometrics are based on Atlanta, GA using the ASHRAE humidity design data.
5) Analysis assumes that humidity is controlled at or below the 70 grain set point for all chilled beam approaches. Humidity is not controlled with VAV approach so space humidity will be higher than the stated/desired 70 grain set point at times and therefore occupant comfort may be compromised with the VAV approach.
6) Fan heat is included in all performance calculations.
7) Primary air system energy is based on chiller, cooling tower and pumps producing 45 degree water. Chilled beam energy is based on 57 degree water.
8) Reheat energy is required when the cooling load delivered to the space exceeds the space sensible load to avoid individual zone overcooling at minimum airflow.
9) Primary fan energy includes duct losses, filters, coils, energy recovery wheels and all other system components. VAV assumes the use of an inverter.
10) Duct pressure losses assume the same velocity for all system approaches - no credit taken for potential duct pressure reduction by the chilled beam approach.
11) MERV 13 filters used as per ASHRAE recommendations.
12) Range of potential LEED points is provided for comparison only; actual points will have to be determined by a LEED professional on a job-by-job basis.
13) Outdoor air volumes set by ASHRAE ventilation requirements and rest room exhaust.
14) Outdoor air volume is higher for the VAV approach as required by ASHRAE to reflect the VRP method or Z factor required by VAV systems to ensure adequate ventilation to multiple spaces.
15) In some cases, where listed, the space temperature set point for the VAV approach is assumed to be 73 degrees and 60% RH, in attempt to enhance occupant comfort since the desired humidity control could not be achieved with this approach. Decreasing the space set point temperature is the common response to the HVAC systems inability to maintain space humidity.

FIGURE A1: Tabular summary Office Example - Humidity controlled with chilled beam approaches, not VAV approach, same 75°F set point

	Traditional VAV (with Total Energy Recovery)	Chilled Beams		
		Total Energy Recovery	EPD/Twin Wheel	Pinnacle
Airflows:	**Cubic Feet per Minute (CFM)**			
Primary supply airflow at peak cooling (includes outdoor air)	8,066	5,550	5,550	3,000
Primary supply airflow at part load cooling (includes outdoor air)	6,287	5,550	5,550	3,000
Airflow within space at peak cooling (approximate with chilled beams induced air)	8,066	10,947	10,947	10,217
Outdoor airflow (note 5)	2,320	2,000	2,000	2,000
Supply Conditions (peak)	**Design Parameters**			
Primary air temperature	56	54	62	62
Primary air humidity (grains)	61	57	57	46
Primary air duct humidity (RH%)	88%	92%	70%	58%
Primary air dew point	53	51	51	48.5
Space supply air temperature	57	61	61	60
Space supply air humidity (grains)	61	63	63	63
Estimated Energy Use (peak cooling)	**Kilowatts per Hour (KWH)**			
Primary air system	22.5	17.6	15.8	9.0
Chilled beam	n/a	3.0	3.0	9.3
Reheat energy required	0.0	0.0	0.0	0.0
Primary fan energy	15.4	8.3	9.6	5.2
Total	**37.9**	**28.8**	**28.3**	**23.4**
Estimated Energy Use (part load)	**Kilowatts per Hour (KWH)**			
Primary air system	16.4	17.0	14.8	8.2
Chilled beam	n/a	0.0	2.2	4.9
Reheat energy required	6.3	5.4	0.0	0.0
Primary fan energy	9.5	8.3	9.6	5.2
Total	**32.2**	**30.6**	**26.6**	**18.2**
Key System Design Issues	**Additional Design Benefits**			
Primary airflow duct diameter	38 inches	32 inches	32 inches	23 inches
Final filter area required (MERV 13)	20 sq ft	14 sq ft	14 sq ft	8 sq ft
Typical space sound levels	40-45 dB	38 dB	38 dB	35 dB
Desired ventilation to each space	Not necessarily	Yes	Yes	Yes
Humidity maintained	Not controlled	Yes with reheat	Yes	Yes
Estimated potential LEED 2.2 points	2 to 5	4 to 9	6 to 12	8 to 15

FIGURE A2: Tabular summary School Example - Humidity controlled with chilled beam approaches, not VAV approach, same 75°F set point

	Traditional VAV (with Total Energy Recovery)	Chilled Beams		
		Total Energy Recovery	EPD/Twin Wheel	Pinnacle
Airflows:	**Cubic Feet per Minute (CFM)**			
Primary supply airflow at peak cooling (includes outdoor air)	7,530	6,284	6,284	3,650
Primary supply airflow at part load cooling (includes outdoor air)	5,115	6,284	6,284	3,650
Airflow within space at peak cooling (approximate with chilled beams induced air)	7,530	11,005	11,005	10,219
Outdoor airflow (note 5)	4,290	3,650	3,650	3,650
Supply Conditions (peak)	**Design Parameters**			
Primary air temperature	56	53	53 - 62	62
Primary air humidity (grains)	61	55	55	44
Primary air duct humidity (RH%)	88%	89%	89% - 60%	51%
Primary air dew point	53	51	51	44
Space supply air temperature	56	62	62	61
Space supply air humidity (grains)	61	61.4	61.4	60.8
Estimated Energy Use (peak cooling)	**Kilowatts per Hour (KWH)**			
Primary air system	26.4	23.8	20.0	14.1
Chilled beam	n/a	0.9	0.9	7.7
Reheat energy required	0.0	0.0	0.0	0.0
Primary fan energy	14.4	9.4	10.8	6.3
Total	**40.8**	**34.1**	**31.7**	**28.1**
Estimated Energy Use (part load)	**Kilowatts per Hour (KWH)**			
Primary air system	16.5	20.3	17.5	11.0
Chilled beam	n/a	0.0	0.4	3.2
Reheat energy required	3.3	14.3	0.0	0.0
Primary fan energy	7.0	9.4	10.8	6.3
Total	**26.8**	**44.0**	**28.8**	**20.5**
Key System Design Issues	**Additional Design Benefits**			
Primary airflow duct diameter	37 inches	34 inches	34 inches	26 inches
Final filter area required (MERV 13)	19 sq ft	16 sq ft	16 sq ft	9 sq ft
Typical space sound levels	40-45 dB	38 dB	38 dB	35 dB
Desired ventilation to each space	Not necessarily	Yes	Yes	Yes
Humidity maintained	Not controlled	Yes with reheat	Yes	Yes
Estimated potential LEED 2.2 points	2 to 5	4 to 9	6 to 12	8 to 15

FIGURE A3: Tabular summary School Example - Humidity controlled with chilled beam

	Traditional VAV (with Total Energy Recovery)	Chilled Beams		
		Total Energy Recovery	EPD/Twin Wheel	Pinnacle
Airflows:	**Cubic Feet per Minute (CFM)**			
Primary supply airflow at peak cooling (includes outdoor air)	9,426	6,284	6,284	3,650
Primary supply airflow at part load cooling (includes outdoor air)	7,734	6,284	6,284	3,650
Airflow within space at peak cooling (approximate with chilled beams induced air)	9,426	11,005	11,005	10,219
Outdoor airflow (note 5)	4,290	3,650	3,650	3,650
Supply Conditions (peak)	**Design Parameters**			
Primary air temperature	55	53	53 - 62	62
Primary air humidity (grains)	60	55	55	44
Primary air duct humidity (RH%)	89%	89%	89% - 60%	51%
Primary air dew point	52	51	51	44
Space supply air temperature	55	62	62	61
Space supply air humidity (grains)	60	61.4	61.4	60.8
Estimated Energy Use (peak cooling)	**Kilowatts per Hour (KWH)**			
Primary air system	29.6	23.8	20.0	14.1
Chilled beam	n/a	0.9	0.9	7.7
Reheat energy required	0.0	0.0	0.0	0.0
Primary fan energy	18.0	9.4	10.8	6.3
Total	**47.7**	**34.1**	**31.7**	**28.1**
Estimated Energy Use (part load)	**Kilowatts per Hour (KWH)**			
Primary air system	22.5	20.6	17.5	11.0
Chilled beam	n/a	0.0	0.4	3.2
Reheat energy required	23.9	16.3	0.0	0.0
Primary fan energy	12.2	9.4	10.8	6.3
Total	**58.6**	**46.3**	**28.8**	**20.5**
Key System Design Issues	**Additional Design Benefits**			
Primary airflow duct diameter	42 inches	34 inches	34 inches	26 inches
Final filter area required (MERV 13)	24 sq ft	16 sq ft	16 sq ft	9 sq ft
Typical space sound levels	40-45 dB	38 dB	38 dB	35 dB
Desired ventilation to each space	Not necessarily	Yes	Yes	Yes
Humidity maintained	Not controlled	Yes with reheat	Yes	Yes
Estimated potential LEED 2.2 points	2 to 5	4 to 9	6 to 12	8 to 15

List of assumptions used for installation cost analyses

1) School chilled beams- 40 eight foot beams for 8,500 square foot building, two pipe heating - $29,016.
2) Office chilled beams - 27 eight foot beams for 8,500 square foot building, two pipe heating - $19,585.
3) Piping estimate - Primary AHU tons times $350 plus $5/sq ft for fan coils and $3.50/sq ft for beams, fan coils at $.21/sq ft for condensate piping.
4) AHU for VAV conventional modular air handling unit with total energy recovery ($2.25/cfm unit plus $2/cfm energy recovery) installation at 50% of system cost.
5) AHU providing outdoor air to fan coil SEMCO EPCH normal market pricing installation at 50% of system cost.
6) AHU for beams normal market pricing for EPCH, EPD and Pinnacle installation at 50% of system cost.
7) Controls for AHUs are included in the controls pricing.
8) Controls for VAV at $1,500 per VAV box plus installation at 150% of material cost.
9) Controls for Fan coils at $800 per fan coil plus installation at 150% of material cost.
10) Controls for chilled beam at 85% of fan coil cost.
11) Chiller and tower cost at $1,000/ton installed, boiler at $600/boiler HP installed.
12) Office VAV ductwork at 2.5 cfm/sq ft (highest flow), fan coil at $1.75/sq ft and chilled beam ductwork with single and dual wheel systems at $1.50/sq ft and chilled beams with Pinnacle at $1.06/sq ft (schools ductwork slightly less due to fewer beams).
13) Fan coils, VAV boxes and chilled beams from market price data.
14) VAV and Fan coils reflect supply air grills, smoke dampers and return grills, Chilled beams only smoke dampers and return grills.
15) Other installation is .5 time the cost of the VAV, fan coil or beams plus all dampers and diffusers.
16) Markup at .03 times total cost of project.
17) Buildings are assumed to have very tight envelopes and vapor barriers, properly located as per ASHRAE Standard 90.1 recommendations to achieve the low infiltration rate and permeance used for the analyses presented in this document. Poor envelop or vapor barrier characteristics would greatly increase the internal latent loads estimated and require greater quantities and/or drier primary airflow values.

References Cited

1. McLay, B., "The Chilled Beam Alternative", 2006 Laboratory Design Handbook, Lab Design News, 2006.
2. Barnet B., "Chilled Beams for Labs: Using Dual Energy Recovery", ASHRAE Journal December, 2008.
3. Harriman L., Brundrett G. and Kittler R., "Humidity Control Design Guide" , ASHRAE 2001, Atlanta GA.
4. SEMCO Inc, Columbia MO, 2008 Internal Latent Load Design Tool for Chilled Beam Systems, software tool.
5. United States Green Building Council, "Leadership in Energy and Environmental Design" LEED Certification program.
6. ASHRAE 2008, "2008 Fundamentals", ASHRAE, Atlanta.
7. Fischer J., Bayer C.W., 2003, "Report Card on Humidity Control", ASHRAE Journal May, 2003.
8. Fischer J., 2000, "Pinnacle Ventilation System Integrates Total Energy Recovery, Conventional Cooling and a Novel "Passive" Dehumidification Wheel to Mitigate the Energy, Humidity Control and First Cost Concerns Often Raised when Designing for ASHRAE Standard 62-1999 Compliance", SEMCO Inc, Columbia MO.

FläktGroup
SEMCO

Pinnacle 6 Chilled Beam Application Guide - FläktGroup Semco 2020-01

EXCELLENCE IN SOLUTIONS

WWW.SEMCOHVAC.COM / CHILLED BEAMS + PINNACLE / 20203101

FläktGroup SEMCO delivers smart and energy efficient Indoor Air and Critical Air solutions to support every application area. We offer our customers innovative technologies, high quality and outstanding performance supported by more than fifty years of accumulated industry experience. The widest product range in the market, and strong market presence in 65 countries worldwide, guarantee that we are always by your side, ready to deliver Excellence in Solutions.

FläktGroup SEMCO
Corporate Headquarters
1800 East Pointe Drive
Columbia, Missouri 65201 USA
573.443.1481
sales.semco@flaktgroup.com

To learn more about FläktGroup SEMCO offerings and to contact your nearest representative please visit
www.semcohvac.com