Active Chilled Beams

Classroom solution using Active Chilled Beams

INTRODUCTION

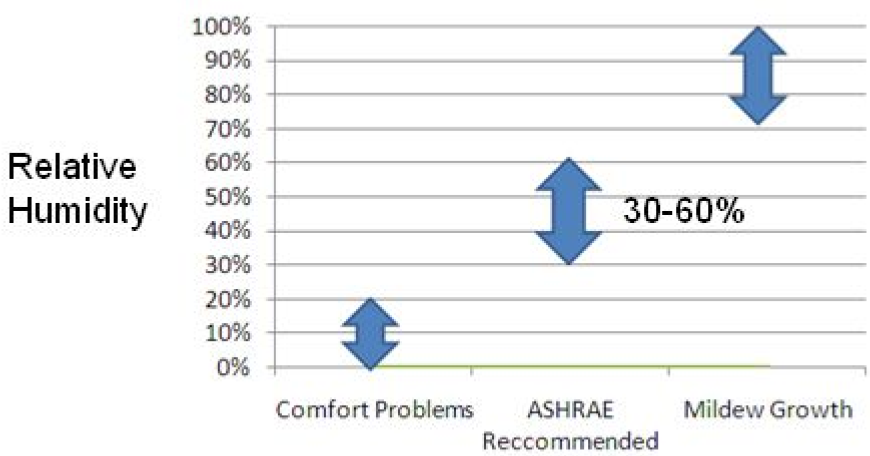
Many people outside of the HVAC industry judge an HVAC system's ability to provide a comfortable environment based solely on its ability to control dry bulb temperature. Most often ignore or place less importance on other aspects affecting comfort such as noise, humidity and ventilation levels.

Noise - Studies have shown that noise levels can significantly affect the learning environment. ANSI standard S12.60 for classroom acoustics requires a maximum background noise level of 35 dBa (about NC-27). As of this writing a number of state-wide organizations have adopted the full standard or modified/derivative versions of the sound standard. With the conventional HVAC systems typically used in schools today (fan-powered VAV, fan coils, unit ventilators) these noise level requirements can be difficult if not impractical to attain.

Humidity - Studies have also shown that mental performance is affected by humidity levels. Most classrooms are fully occupied for much of the day and as such are most often at their full design latent cooling load. Because classrooms are most typically perimeter zones, their sensible cooling demand will vary from the full cooling design load to no cooling load at all such as in the spring and fall. With conventional HVAC systems (VAV, fan coils, unit ventilators) this often leads to a loss of humidity control in the classroom. (For more information, refer to "Failing Grade for Many Schools. Report Card on Humidity Control" by Fischer and Bayer in the May 2003 ASHRAE Journal).

As an example, with a VAV system the supply air will be modulated down as the sensible cooling load decreases. While the design dry bulb temperature of the room will be maintained, the latent cooling capacity being provided will decline with the reduced supply airflows. As a result the relative humidity of the room may rise if insufficient latent cooling capacity is being provided by the supply air. With fan coils and unit ventilators a similar loss of humidity control can occur. As the valves serving the cooling coils cycle or modulate toward closed due to the lower sensible cooling demand, the supply air will not be sufficiently cooled/dehumidified resulting in a similar loss of humidity control.

The ASHRAE 62 Indoor Air Quality standard requires that relative humidity levels be controlled to below 60% for proper air quality. In addition to comfort concerns, higher humidity levels can foster microbial growth (bacteria, fungi, etc). Microbial growth can result in odors, allergens and even toxins. Microbial growth can occur when relative humidity levels are generally over 60-70%.



Ventilation - The issue of ventilation air rates and compliance with the requirements of the ASHRAE 62 can also be an issue with the more conventional systems. Using the previous VAV example, the ventilation air requirement will also likely not be met at part load conditions unless the air-side system is operating on a full economizer mode (much in doubt on temperate, yet humid days).

ACTIVE CHILLED BEAM SOLUTION

Active chilled beam systems address noise, humidity and ventilation issues very effectively and efficiently in a classroom environment, as well as offering other benefits. In a typical classroom active chilled beam system, a central air handling system supplies a constant volume of ventilation air to the active chilled beams. A chiller and boiler supply chilled water and hot water to the active chilled beams. Room air is induced into the active chilled beams where it is cooled or heated to control the classroom's dry bulb temperature.

Benefits of the active chilled beam system are numerous.

- A constant volume of primary air provides the full ventilation air required and humidity control at all times and at all sensible load conditions.

- Very uniform temperatures are achieved throughout the classroom due to the thorough mixing of the primary and temperate induced room air.

- There is less potential for objectionable drafts during cooling or stratification during heating as the discharge air temperatures are much more temperate, as the primary air is blended with the room air before being discharged into the room.

- Low noise levels (when designed at typical unit inlet static pressures of 0.5" w.c.)

- The fan energy consumption is significantly reduced as the primary airflow is much lower than other conventional "all air" HVAC systems (typically about 60-70% less in a classroom application).

- There is an opportunity to reduce the heating plant energy consumption. With the relatively low hot water temperatures typically used by the active chilled beam systems,
  - the efficiencies of condensing boilers are maximized.
  - the use of geothermal or water-water heat pumps is optimized.

- There is an opportunity to reduce the cooling plant energy consumption through several scenarios.

  - The use of water side economizers on the chilled water loop serving the active chilled beams is often very beneficial, as the economizer's effectiveness/usage is maximized at the relatively warmer chilled water temperatures (typically 56-58F) used with the active chilled beams.

  - If the project is large enough for multiple chillers, use a separate chiller serving the active chilled beams as the chiller efficiencies (COP) are maximized (about doubled) at the relatively warmer chilled water temperatures used with the active chilled beams.

  - In a smaller project with a single chiller, decouple the ventilation air requirement/load from the chiller by providing the primary air through a packaged DX dedicated outdoor air system (DOAS), and use the chiller to only serve the active chilled beams at the relatively warmer chilled water temperatures to achieve the higher chiller efficiencies.

- School maintenance is reduced as the active chilled beams have no moving parts requiring regular maintenance. Only very infrequent vacuuming of the coil in the active chilled beam is required, as the system is designed to have a dry coil with all the latent cooling being provided by the primary air from the central air handler. In addition, the active chilled beams use simple, low cost controls (zone valves and wall thermostats).

This technical paper will discuss alternative design scenarios using an active chilled beam system in a classroom environment, as compared to a conventional "all air" VAV system. As the classrooms are most typically perimeter zones, the VAV design assumes the use of fan-powered VAV units.

ASSUMPTIONS

<u>Design conditions</u>

Zone design temperature	75°F db/50% RH (64.9 grains)
Outdoor design temperature	95°F db/78°F wb (118.0 grains) summer & 0°F db winter

<u>Typical perimeter classroom zone description and occupancy</u>

Window area	150 sq. ft. (40% of wall area)
Floor area	1,000 sq. ft. (31.5 feet x 31.5 feet deep)
Wall exposed to perimeter	31.5 feet
Floor-to-suspended ceiling height	10 feet (zone volume = 10,000 cu. ft.)
Maximum occupancy	30 people (50% more than ASHRAE est. occupancy)
Minimum OA ventilation air requirement	450 cfm (15 cfm per person per ASHRAE 62)
OA ventilation rate chosen	600 cfm (30% more to meet LEED IAQ 6.2 credit)
OA rate needed for building pressurization	150 cfm (0.15 CFM/sq. ft.)
Exhaust airflow for bathroom exhaust	100 cfm (based on 1 toilet/75 cfm per 20 people)
Excess airflow available for return	350 cfm

<u>Zone internal sensible and latent cooling loads</u>

Lighting	6,280 Btuh (2 watts/sq. ft. or 6.3 Btuh/sq. ft.)
People	7,500 Btuh (250 Btuh per person)
Equipment	3,410 Btuh (1 watt/sq. ft or 3.41 Btuh/sq. ft.)
Envelope	12, 810 Btuh (12.8 Btuh/sq. ft.)
Total sensible cooling load	30,000 Btuh (30.0 Btuh/sq. ft.)
Room internal latent cooling load	6,000 Btuh (200 Btuh per person)

<u>Zone infiltration sensible and latent cooling loads</u>

Infiltration airflow	40 cfm (see explanation on page 4)
Infiltration sensible cooling load	870 Btuh
Infiltration latent cooling load	1,470 Btuh

<u>Zone total sensible and latent cooling loads</u>

Total sensible cooling load	30,870 Btuh (30.9 Btuh/sq. ft.)
Total latent cooling load	7,470 Btuh (7.5 Btuh/sq. ft.)
Total cooling load	38,340 Btuh (38.3 Btuh/sq. ft.)
Sensible heat ratio	0.81
Total heating load	9,450 Btuh (300 Btuh per lin. foot of perimeter exposure)

<u>HVAC System</u>

Unit inlet static pressures	about 0.5” w.c (for both the ACBs and fan-powered VAV terminals)
Fan and duct heat gain	2ºF db
Pump efficiencies and head pressure	75% at 30 ft

Two design scenarios using the Active Chilled Beams will be examined, as compared to the fan-powered VAV system.

<u>Scenario #1 – Active chilled beams using cold primary air</u>

Fan efficiencies and total static pressure	75% at 3.0” w.c.

<u>Scenario #2 – Active Chilled Beams with sensible heat recovery providing room neutral primary air</u>

Fan efficiencies and total static pressure	75% at 3.5” w.c.
Sensible heat recovery device efficiency	55 — 65%

HUMIDITY CONTROL/CONDENSATION ISSUES

Chilled Water Temperature and Primary Air Latent Capacity - Controlling humidity levels by providing sufficient latent cooling capacity is particularly important in the design of active chilled beam systems due to condensation concerns. Most of the active chilled beam models available currently on the market utilize coils that are oriented horizontally with no condensate drain provisions. (Dadanco does offer some models with drain pans. Even if these are used, its highly preferable to provide all of the latent cooling with the primary air so the coils in the active chilled beam are dry which greatly minimizes maintenance issues).

With a 75ºF db/62.5ºF wb (50% relative humidity) room design condition, the room's dew point temperature is 55ºF. If the chilled water temperature was below 55ºF, condensation could form on the coil. If a 56ºF chilled water temperature or above is chosen in the design of the active chilled beam system, condensation would not occur at these room design conditions.

In addition to the selection of the chilled water temperature, the latent cooling capacity being provided by the primary air must also be sufficient to satisfy the room's latent load. If the latent cooling capacity provided by the primary air is insufficient, the room's design point could not be maintained and the room's relative humidity would increase. If the room's relative humidity increased to 60%, the room's dew point temperature would rise to about 60ºF and condensation could form on the coil.

Infiltration - While some might argue that with proper building construction that an allowance for infiltration is unnecessary, it would be imprudent not to anticipate some amount of infiltration in the system design. Moisture from outside can migrate into the building for a variety of reasons, as well as being generated from sources within the building. As moisture migration is driven by differences in vapor pressure, moisture can enter a building even when it is maintained under a positive pressure. In addition, a building's tightness can decline with age. For these reasons, an allowance for infiltration should be included in the system design. In this paper an infiltration rate of 40 cfm was assumed.

Primary Air Conditions - In an active chilled beam system the primary air provides all of the latent cooling capacity required. The latent cooling capacity of the primary air can be determined by:

$$Q\ latent = 0.68\ x\ CFM\ x\ \Delta\ Humidity\ (Room\ Design - Primary\ Air)$$

The difference in the humidity level between the primary air and room design must be at least 18.3 grains to satisfy the latent load in our classroom using the 600 CFM ventilation air requirement. As such the primary air must have a moisture content of no more than 46.6 grains. To achieve this humidity level, the primary air temperature leaving the cooling coil would be
48.0F db/47.1F wb (46.6 grains).

The primary air condition being delivered to the active chilled beams (accounting for the 2F fan/duct heat gain) would be 50.0F db/48.0F wb (46.6 grains). There should be no concern about drafts using this air temperature as the primary air will be tempered up by the induced room air, and the mixed air will be discharged into the zone at a higher temperature than with a conventional HCAV system.

ENERGY COMPARISON

The cooling and heating plant energy savings possible through the use of water-side economizers, dedicated chillers, high efficiency condensing boilers and water-water heat pumps with an active chilled beam system are not included in this analysis. These can be, however, very significant. This analysis will be confined to comparing the fan and pump energy.

Fan and Pump Energy - The supply airflow required with a conventional “all air” HVAC system (such as VAV) would be 1,422 cfm or 1.4 cfm/sq. ft. (using a conventional 55F primary air temperature) as opposed to the 600 cfm or 0.6 cfm/sq. ft. for the ventilation rate chosen for the active chilled beam system (about a 60% reduction). As fans are typically one of the largest consumers of energy in a typical school, very significant energy savings are realized.

The conventional systems would typically be designed for about a 10°F ΔT on the water side, while the active chilled beam system would be typically designed with a 5-8°F ΔT on the water side. A 5°F ΔT on the water side was used for the comparisons in this technical paper. Hence the pump energy is typically somewhat more with the active chilled beam system due to the higher water flow rates.

With the active chilled beam system the central fan energy consumption per classroom would be 0.282 kWH and the pump energy consumption would be 0.038 kWH at the assumed operating conditions for a total of 0.320 kWH.

Using a conventional “all air” VAV system the central fan energy would be 0.670 kWH and the pump energy would be 0.019 kWH for a total of 0.689 kWH. In addition the fan-powered VAV terminals have a fan motor energy consumption of about
750-900 watts per classroom (varies somewhat by manufacturer). If we use 770 watts for the fan-powered VAV unit, the total pump and fan energy consumption per classroom is 1.459 kWH. The combined active chilled beam fan and pump power is about 80% less than the VAV system.

Scenario #1 - Fan and Pump Power Comparison

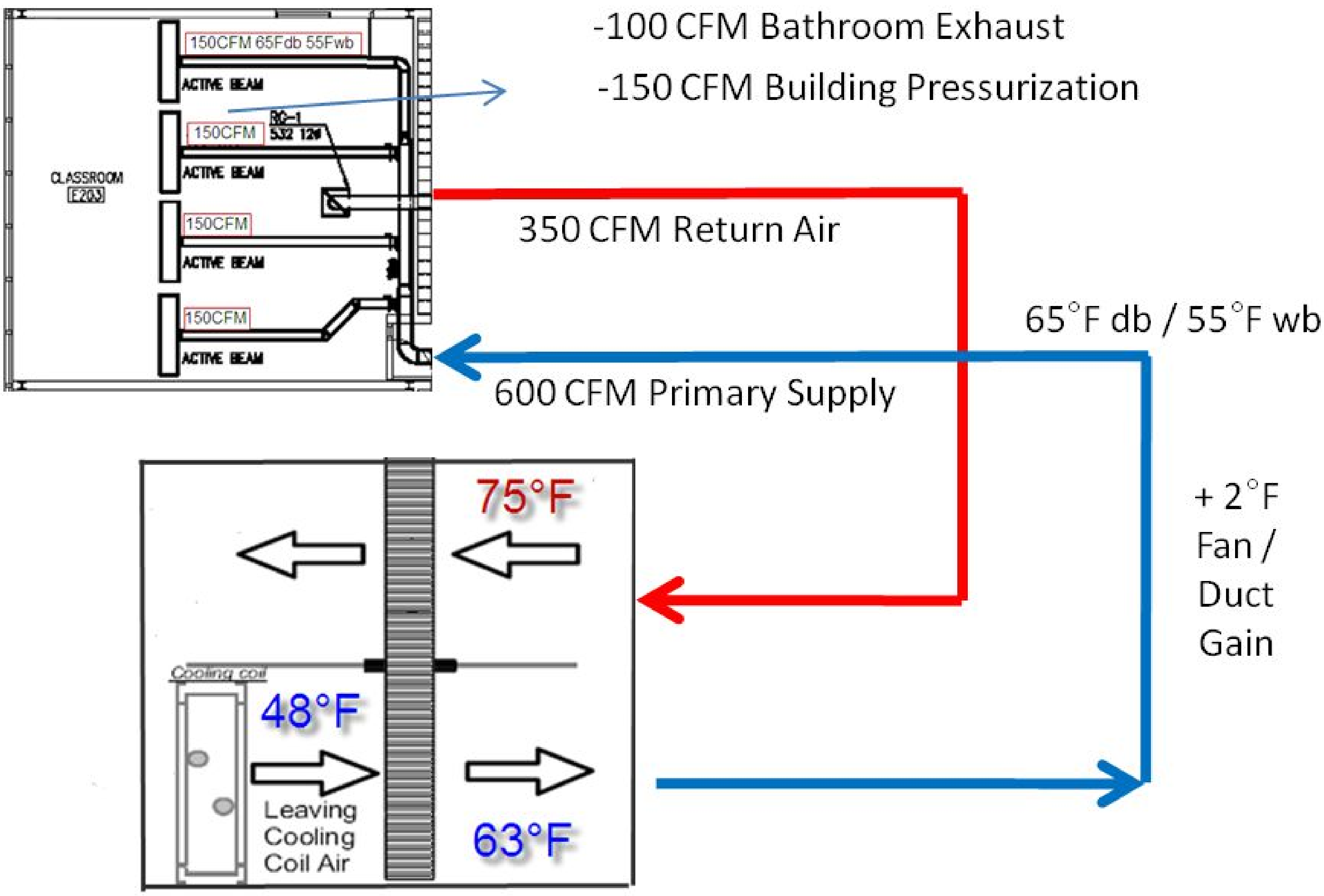
	Fan Power	Pump Power	Total Power
Active Chilled Beams	0.282 kWH	0.038 kWH	0.320 kWH
Fan-powered VAV	1.440 KwH	0.019 kWH	1.459 kWH

Reheat - In a typical active chilled beam system a portion of the sensible cooling load is normally provided by the primary air with the remaining sensible cooling being provided by the water coil in the active chilled beam. It is generally desirable to
minimize the portion of sensible cooling being provided by the primary air to minimize the reheating required to avoid overcooling the classroom when at low sensible loads. With the active chilled beam system the primary airflow of 600 cfm at 50°F db would be providing 16,410 Btuh of sensible cooling or about 50% of the design sensible cooling load. When the sensible load was below that level reheating of the primary air would be required.

With the VAV system, the primary airflow of 600 cfm at 55°F db would be providing 13,020 Btuh of sensible cooling or about 34% of the design sensible cooling load. When the sensible load was below that level reheating of the primary air would be required.

In some cases reheat can be provided through the use of recovered heat from the chiller. A heat exchanger located in the chiller's refrigerant piping between the compressor and air-cooled condenser could be used to heat the hot water circuit serving the active chilled beams.

If condenser heat cannot provide the reheat energy, another approach is to recover the heat in the return/exhaust air tore-heat the primary air. A sensible heat recovery device (i.e. heat wheel, plate recuperator, coil loop) is located in the supply air stream downstream of the cooling coil. The heat recovered from the return/exhaust air stream would be used to tem-per the primary air leaving the cooling coil.



55-65% Efficient Energy Recovery Wheel

With about 150 cfm (0.15 cfm/sq. ft.) per classroom needed for building pressurization and 100 CFM needed for bathroom exhaust make-up, 350 cfm of return/exhaust air would be returned from the classroom as compared to the 600 cfm of outdoor air delivered to the zone. At this ratio a sensible heat recovery device can typically achieve a heat recovery efficiency of about 55-65%. As such in our example where the temperature of the primary air leaving the cooling coil is 48°F and the exhaust air is 75°F, the primary air temperature leaving the sensible heat recovery device could be increased to as much as 63-66°F. Accounting for the 2°F fan/duct heat gain the primary air temperature being delivered to the zones could be around 65°F (nearly thermally room neutral).

The amount of heat recovered used to temper the primary air would be adjusted to only as much as that needed to not over-cool the zone at the lowest part load condition. This would minimize the times when the tempered primary air has to be re-cooled by the active chilled beams. With a plate-type recuperator this is accomplished with a face and bypass damper arrangement to bypass the return/exhaust air around the recuperator. With a heat wheel this is accomplished by slowing the speed of the wheels rotation. With a coil loop this is accomplished with modulating water valves at the coils.

Due to an increase in total fan operating static pressure from the heat recovery device the central fan energy consumption would increase to 0.329 kWH in this second scenario (as compared to 0.282 kWH in the first scenario). Due to the increase in the chilled water flow rate (as the water side is handling more of the load) the pump energy consumption would increase to 0.054 Kwh in this second scenario (as compared 0.038 kWH in the first scenario). The total fan and pump energy in the second scenario would be 0.383 kWH (about 20% more than in the first scenario), still about 75% less than the VAV system.

Scenario #2 - Fan and Pump Power Comparison

	Fan Power	Pump Power	Total Power
Active Chilled Beams	0.329 kWH	0.054 kWH	0.383 kWH
Fan-powered VAV	1.440 KwH	0.019 kWH	1.459 kWH

With the primary air being delivered to the zone at 65°F db, the primary air would be providing 6,570 Btuh of sensible cooling or only about 21% of the design sensible cooling load. As such reheating would only be required when the sensible cooling load was below 21% of the design sensible cooling load, far less than the VAV system.

The extent to which reheating occurs varies by location and weather. In warmer climates reheating may never occur in the second design scenario as the heating loads could be negligible and the internal heat gains significant. As an example if the classroom room was fully occupied, the sensible cooling load strictly due to internal gains (lights, people and equipment) is 17,190 Btuh or 55% of the design sensible cooling load. Even if the room was unoccupied with the lights on, the sensible cooling load would be 6,280 Btuh or about 20% of the design sensible cooling load.

In the northern climates reheating would occur in temperate conditions where some zones need little-to-no cooling, but dehumidification is required. The potential for reheat is far less with the second active chilled beam scenario, than that for the first active chilled beam design scenario and the VAV system.

Reheating Required Comparison

	When cooling load is less than
Active Chilled Beam Scenario #1	50% of design
Active Chilled Beam Scenario #2	21% of design
Fan-powered VAV	34% of design

DEMAND CONTROL VENTILATION

As most classrooms are unoccupied during some hours during the typical day, some would like to employ a "zone demand control ventilation" strategy. With the conventional approach to demand control ventilation, this involves a rather elaborate system using of CO2 sensors which can be expensive, complicated and require maintenance (sensor recalibration).

A simpler and less expensive method can be incorporated into the active chilled beam system design for classroom applications. This is easily accomplished by adding one single duct VAV terminal upstream of the active chilled beams at each classroom. Simple, maintenance-free occupancy sensors would monitor the room's activity. When unoccupied for period of time, the VAV terminal would modulate down the primary air being provided to the active chilled beams serving that classroom. The minimum primary air setting would be determined by the designer and could be easily adjusted later if desired. When the classroom was reoccupied the VAV terminal would modulate the primary air back to its full airflow.

There are few caveats to this approach. As an active chilled beam's performance is affected by the static pressure at the unit's inlet, the VAV terminals would require pressure independent controls (as essentially all now do). All the classrooms being served by the same air handler would need an upstream pressure independent VAV terminal to guard against static pressure fluctuations which would affect the active chilled beams' performance. Secondly, as the capacity of the active chilled beam decreases with the reduction in primary air, it's possible that the room design temperature will not be fully maintained during the unoccupied mode. Most, however, would not object to this issue and liken it to a setback mode. Lastly, the
minimum primary airflow must be sufficient to satisfy the latent load during these unoccupied periods (typically only the infiltration component of the design latent load).

ACTIVE CHILLED BEAM UNIT SELECTIONS

Numerous product models of active chilled beams are available, each with differing performance. While most active chilled beam models currently available do not have drain pans, Dadanco offers models that do. With concerns about unanticipated latent loads exceeding the original design calculations (i.e. mopping floors, wet clothing from students returning from outside during a rainstorm, etc.), some designers choose to use a model equipped with a drain pan to prevent any incidental
condensate formed from falling into the classroom.

As the concern relates to unanticipated conditions, the drain pans are normally capped and not piped to drain. Any incidental condensate formed due to these transient conditions will collect in the drain pan and re-evaporate. (In some areas, code inspectors must be made to understand that the drain pans are safety devices and that the system has been designed to not condense at the beams, thus piping the condensate to drain is not necessary).

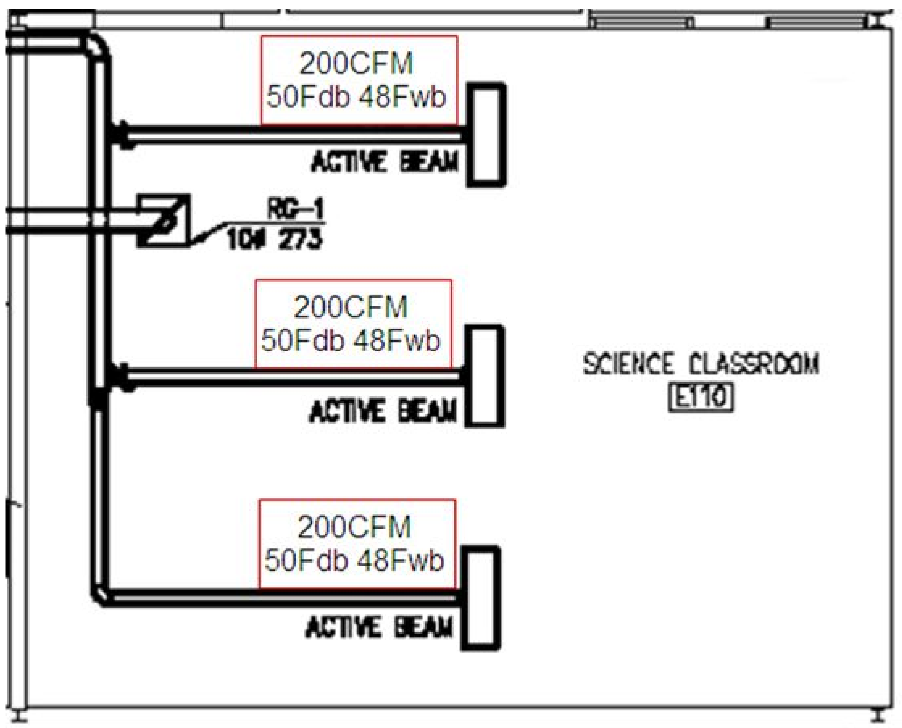
Active chilled beams are also available in both 2 and 4-pipe coil arrangements. While the heating could be provided with 4-pipe units, this would require 4 piping connections per unit (12-16 connections per classroom based on 3-4 units per classroom). An alternative, lower first cost approach is to install a hot water coil in the ductwork upstream of each classroom (or group of classrooms on the same exposure) to provide the heating. This reduces installation costs and maximizes the cooling capacities of the units (as 2-pipe units utilize the full coil surface area for the cooling circuit).

For both active chilled beam design scenarios discussed in this paper, Dadanco Model ACB20 units with integral drain pans and 2-pipe coils have been chosen. The active chilled beam unit selections for the two scenarios are:

Active Chilled Beam Unit Selections for Scenarios #1 & 2

	# Units	Inlet SP (In. w.c.)	Primary Airflow (CFM)	Primary Air Sensible Cooling (Btuh)	Primary Air Latent Cooling (Btuh)	Chilled Water Flow (GPM)	ACB Coil Sensible Cooling (Btuh)	Total Sensible Cooling (Btuh)	Total Cooling (Btuh)
ACB #1	(3) AC-B20	0.5	600	16,410	7,470	3.0	14,740	31,150	38,620 Btuh
ACB#2	(4) AC-B20	0.6	600	6570	7,470	10.0	24,510	31,080	38,550

Scenario #1—Active Chilled Beam with cold primary air

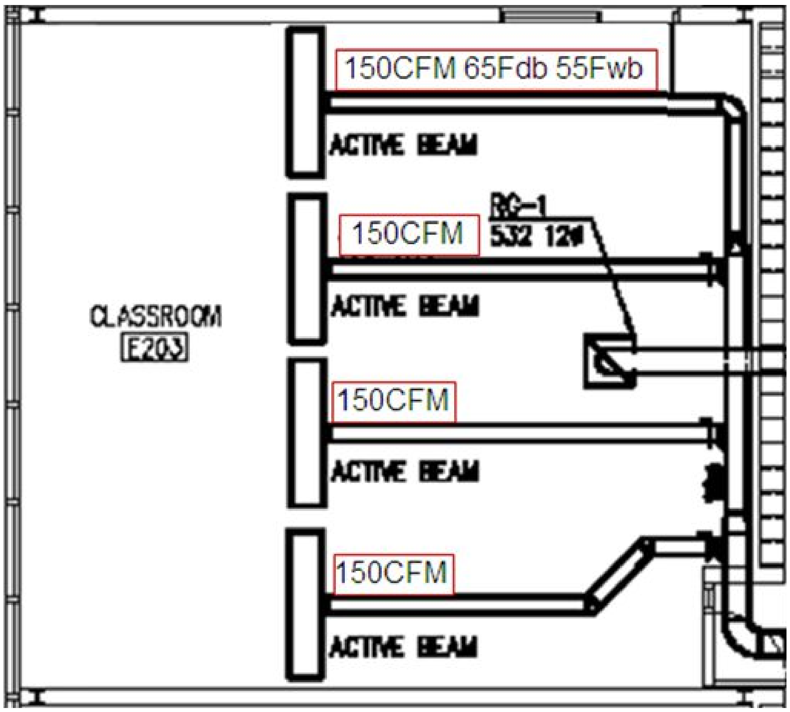


Each unit requires supply and return connections for 3.0gpm, 56F chilled water.

The units are situated in the middle of the classroom for even airflow and temperature distribution.

Each active chilled beam is provided with 200 CFM primary air (0.5 In w.c. Static, 50F db/48F wb)

Scenario #2—Active Chilled Beams with sensible heat recovery providing room neutral primary air



Each unit requires supply and return connections for 10.0 gpm, 56°F chilled water.

The units are situated in the middle of the classroom for even airflow and temperature distribution.

Each active chilled beam is provided with 150 CFM primary air (0.6 In w.c. Static, 65°F db / 55°F wb)

<u>SUMMARY</u>

There is a persuasive argument for the use of active chilled beam systems in classroom applications. The active chilled beam systems can provide superior comfort – temperature, ventilation, humidity and noise levels. In addition, there are significant energy savings due to reductions in the fan energy consumption, as well as the potential for substantial heating and cooling plant energy savings.

While the design scenarios in this paper only discuss classrooms, an active chilled beam system is often an excellent system choice for the office areas in the school as well. They are, however, often less suitable to the common areas such as cafeterias and gymnasiums best served by separate systems dedicated to these areas.

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