Introduction to Chilled Beams

A chilled beam is an air distribution device with an integral coil that may be installed within a space in order to provide sensible cooling and heating.

There are two main types of chilled beams: active and passive.

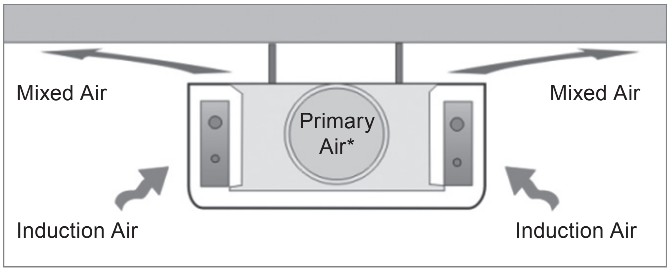
Active chilled beams are those that have duct work supplied to them providing a specific amount of primary air to the pressurized plenum within the device to be discharged through induction nozzles, mix with entrained air, and ventilate the room.

Active beams should be utilized when sensible cooling, heating, and ventilation air are required. If the design calls for supplementary cooling only, or the complete ventilation requirements of the building's design are being met by some other means, passive beams may be used. Classrooms, private and public office buildings, meeting facilities, health care facilities, other environments that may have moderate to high sensible heat ratios, and building retrofits where space for new mechanical equipment may be limited are all good applications for active chilled beams.

A passive chilled beam is one that is not ducted, does not supply primary air, and does not utilize fan powered equipment for any portion of the air that crosses the coil; they rely on induction air being drawn across the coil by the natural gravitation and buoyancy of air. To cool the space, warm air rising to the ceiling enters the beam from above as the chilled air that has passed through the coil drops down; the motion of the cool air dropping creates a pressure drop behind it that draws more warm air through the coil. Passive chilled beams are a good solution to provide sensible cooling in labs or other spaces where processes and people generate high heat loads, especially those that are sensitive to changes in pressure or ventilation and require no additional airflow. Some applications may have a ventilation rate requirement or high enough latent load that a traditional HVAC system would be more appropriate for use than active chilled beams, but could benefit from the use of passive beams for supplementary cooling. Building retrofits where additional cooling is required but the original ventilation system will remain in place are also good passive beam applications.

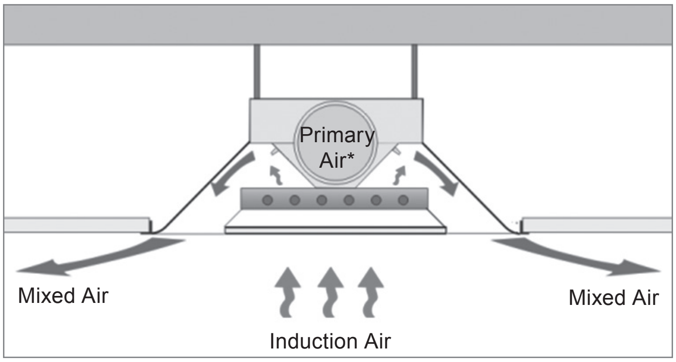
Due to the tendency of warm air to rise to the ceiling, stratification is possible with all air distribution devices if the discharge air temperature is too high or discharge velocity is too low. With chilled beams, stratification is likely if the temperature of the water entering the coil is too high. Due to air velocity and the mixing required to distribute treated air to the occupied zone, passive beams should not be used for heating applications.

ACTIVE CHILLED BEAM, EXPOSED INSTALLATION



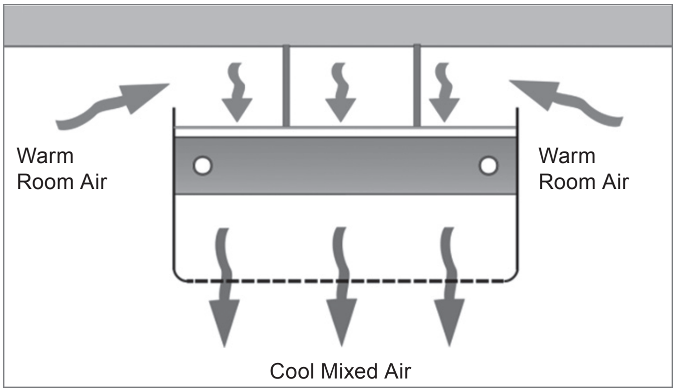
NOTE: Ducted primary air supply.

ACTIVE CHILLED BEAM, SUSPENDED INSTALLATION



NOTE: Ducted primary air supply.

PASSIVE CHILLED BEAM



USE ACTIVE BEAMS
When both cooling and ventilation are required.
When normal cooling capacities are needed (20-40 BTU/h/ft^2) and required heating capacity is not extremely high.
When required airflow is less than 1.5 cfm/ft^2, as low as .15 to .25 cfm/ft^2.
When ceiling heights do not require that the beam is to be installed more than approximately 13 ft. Above Finished Floor (A.F.F.)

USE PASSIVE BEAMS
When supplementary cooling is needed but additional ventilation is not required.
When fresh air is being supplied with a separate system such as displacement ventilation.
When the airflow rate within the space is high.
When installation is in a space with higher ceilings.

Introduction to Chilled Beams (Continued)

Multiservice beams are those that incorporate multiple ceiling components such as lighting, sound devices, fire sprinklers, motion detectors, and other various options. They are well suited for certain applications that may require off-site assembly and integration of those components. In order to properly select and implement multiservice beams, extensive integrated building design is necessary. Multiservice beams are also well suited for applications such as retrofits, where ceiling space is limited. They offer an opportunity to provide aesthetically pleasing solutions to conceal service items in exposed ceiling spaces and consolidate services to save space where ceiling height is low.

There are some applications where chilled beams may not be an appropriate design. Places such as natatoriums, saunas, bathrooms, locker rooms, and kitchens may expose beams to conditions with latent gains too high for the beams to overcome. Spaces with operable windows that may have no method of control and where beams may be immediately exposed to outdoor temperature and humidity conditions are poor applications for any type of chilled beam.

History of Chilled Beams

Although designing with chilled beams is a relatively new technique in North America, chilled beam design as we understand it today has been in use in Europe for decades. The concepts behind induction technology have been understood in America since the early 1900s with Willis Carrier's introduction of HVAC induction units. Carrier's induction units utilized the similar principle of high velocity air jets inducing space air across a coil, but were primarily provided as "under-window" units similar in construction to today's fan coil units. Utilizing mechanical equipment to heat and cool via ceiling radiation can be traced back to the 1940's, when Norwegian engineer Gunnar Frenger developed and patented a device configured of a pipe attached to an aluminum profile to provide radiant temperature control. The first radiant ceiling was installed in Gothenburg, Sweden in the late 1960s. Combining the concepts of utilizing high velocity jet air to induce airflow and Gunnar Frenger's applications of radiant panel cooling within the ceiling, designs began to more closely model what we see in chilled beam technology today. The first radiant cooling device that incorporated supply air into its design was installed, also in Gothenburg, in 1972 as the first step toward today's active chilled beams. What could be considered the predecessor to today's passive chilled beams were first installed in Stockholm, Sweden in 1986. Today, chilled beams are one of the most common HVAC systems installed in Europe. It has only been within the last decade that the technology started to catch on in designs in North America, but the comfort, quiet, and efficiencies that a chilled beam system provides are making it a popular and well established technology.

Benefits of Chilled Beams

Various methods of design exist, each possessing their own set of risk and reward. Chilled beam systems may provide several benefits, some of which are:

- Potential reductions in initial costs of equipment and construction material
- Increase in occupant comfort beyond that which is achievable through traditional systems
- Space adaptability and energy efficiency
- Simple operation and maintenance

FIRST COST

"First cost" discussions of chilled beam systems frequently involve assumptions that the first costs of providing chilled beams are higher than those associated with providing traditional air distribution products. That is correct, of course, when considering only the air distribution device itself. However, that assumption does not take into account the reduction in sizes and capacities of other equipment and construction material. Traditional systems involve mixing return air with outdoor air, thus handling the total supply air volume and sensible capacity at the air handling unit. Chilled beam systems handle return air and sensible load within the space, reducing the total volume of supply air and shifting partial loads from the unit to the space which results in a large decrease in the amount of duct work (and associated handling and labor costs) required. Reducing space required for the duct work can yield significant savings in floor-to-floor height space requirements. This leaves more room for occupants, processes, and in some cases, even reduces costs of structural components by decreasing overall building height. Handling the sensible load in the space and re-circulating entrained air also allows for significant reduction in size and capacity of air handler components such as heating/cooling coils, filters, supply fans, etc., which frequently reduces the total overall size of the air handler.

COMFORT

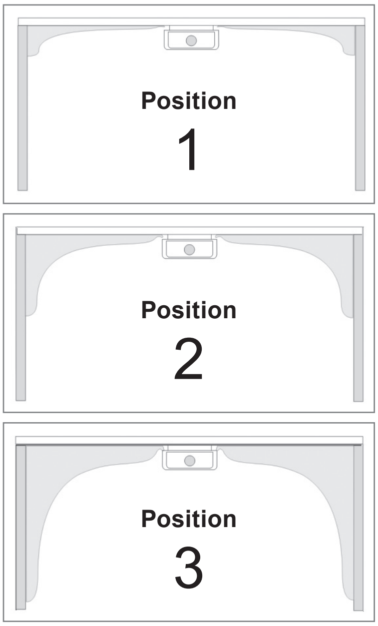
An increase in overall comfort in a space conditioned with chilled beams is mostly a result of the decrease in noise, draft conditions, and temperature inconsistency. Where traditional overhead air distribution systems produce sound levels in the range of NC 35-40, chilled beam systems typically operate with sound levels under 20 NC. Chilled beams are designed to deliver air at lower velocity than standard overhead systems, thus reducing the possibility of unpleasant draft conditions. Additionally, the design of chilled beams results in highly effective mixing of room air and primary air supply which creates comfortable and consistent room temperatures.

ADAPTABLE

Designing a chilled beam system according to an 'adaptable' strategy can allow flexibility in design that may produce excellent indoor climate conditions for the life of the building, regardless of changes of use or layout within the space. This adaptability is

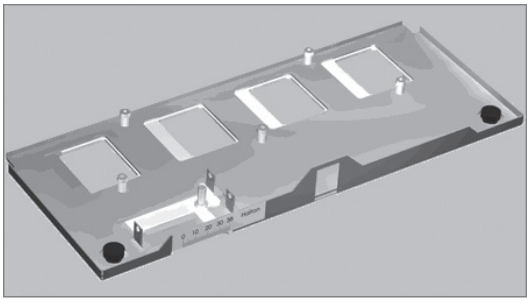
Benefits of Chilled Beams

accomplished by utilizing chilled beams with options such as velocity controllers, air quality devices, and controls that allow for water flow and temperature regulation. The manually adjustable *Krueger by Halton* Velocity Control (HVC) device has three positions which change the width of the discharge slot, thereby increasing or decreasing the throw. The HVC has a very slight impact on capacity, pressure, and sound.



HVC Adjustment Positions

The *Krueger by Halton* Air Quality (HAQ) device is part of an adaptable design that gives the end user the ability to increase or decrease fresh-air CFM to change the chamber pressure to match the available duct pressure at the branch; the adjustable device can be manually or electronically moved to open or close the end of the plenum where primary air can be discharged into the space without crossing the coil or impacting the efficiency of the induction nozzles in the active portion of the beam.



Manual HAQ

Whether air is being supplied at the minimum ventilation rate or if mixed air is being supplied, a means to exhaust air from the space is required in order to maintain appropriate building pressure. Krueger offers an exhaust valve option on several different chilled beam models as an additional way of relieving air from the room. The exhaust valve is integrated into the chilled beam on either the left or right side of the beam. In chilled beam applications, there are typically two lengths referenced: total length and active length. The active length indicates the length of the coil within the beam; total length is for the beam itself including the pressurized plenum and architectural casing. For example, the “active” portion of *Krueger by Halton* ABC active chilled beam is 13” less than the “total” length. If the exhaust valve option is selected, the active length of the beam is reduced by 16” of the total length of the beam as it requires 3” of additional space.

Exhaust Valve

ENERGY EFFICIENCY

Energy efficiency in design is a major reason for recent increases in popularity of the use of chilled beam systems. The handling of “return air” and the sensible load within the space that occurs with the utilization of chilled beam technology yields several energy efficiency benefits. Removing all or part of the sensible load from the air handler decreases the energy consumption at the unit. In addition, since energy transfer by water is more efficient than by air, there is a reduction in potential energy losses experienced while the air is delivered through the duct work to the space. By separating sensible cooling loads from ventilation loads, airflow supply rates may be decreased just enough to satisfy minimum ventilation requirements. Utilizing chilled beams with a constant volume Dedicated Outdoor Air Supply (DOAS) unit that delivers required minimum ventilation rate airflow may decrease the overall amount of outdoor air being supplied and therefore decrease the energy output required to treat that air. Due to the fact that chilled beams utilize induced air from the space and air does not need to be returned to the unit to be re-conditioned, fan energy consumption at the air handler is reduced. In addition to energy savings as a result of changing air movement and capacity considerations, chilled beams offer energy savings related to chiller efficiencies as well. Chilled beam systems require that higher temperature chilled water be supplied to the beams. In traditional systems, the water utilized by traditional ancillary air handling devices such as terminal units and fan coil units may be supplied at anywhere from 40°F to 45°F. With chilled beam systems, chilled water is rarely supplied at temperatures below 58°F. This difference in water temperature supply and increase in return water temperature may produce significant increases in chiller energy efficiency. The need to re-heat cooled air is decreased by the provision of higher chilled water supply temperatures and by the mixing of the treated primary air supply with the large volume of entrained air from the room.

OPERATION & MAINTENANCE

Operation and maintenance of chilled beam systems is very simple, reliable and far less likely to require maintenance or replacement parts. Because air is supplied by fans upstream of the device, there are no moving parts in the chilled beam itself that may be prone to wear. In most designs, filtration occurs upstream, so replacement filters are not required for individual beams. Service costs are minimal with infrequent and/or “as necessary” vacuuming of the coil, which can be accessed via a hinged front panel in most devices, is typically all the maintenance required for the beam itself.

Selection & Design Guidelines

Whether active or passive beams are selected, any primary air supply with an associated chilled beam system must be treated; that is, the air must be sufficiently conditioned (heated/cooled, humidified/dehumidified) to handle both the outdoor load and the latent space load. Chilled beam systems should be selected based upon the calculated heat gain for the space, less the cooling effect of the primary ventilation air. The outdoor load and latent load should be considered in the design of the air handler; chilled beams within the space are designed to operate dry (without condensate) and handle sensible temperature load only, or in the case of active chilled beams – sensible temperature load and minimum (or greater) ventilation air supply. In order to provide an appropriate selection and layout of chilled beam systems, designers must take into consideration several factors:

- Airflow and air distribution requirements.
- Structural and architectural design.
- Type of system(s) available for design.
- Heating, cooling, and climate considerations.
- Operational considerations and methods of control.

Evaluate Airflow & Air Distribution Requirements

Occupant comfort should be a primary concern when evaluating airflow and air distribution requirements, just as it should be when designing traditional overhead air distribution systems. ASHRAE standard 55 relates to temperatures and velocities within the occupied zone that have a direct impact on occupant comfort. The standard defines the occupied zone as being from the floor to 67” high for standing occupants, and floor to 43” high for seated occupants. Per standard 55, vertical temperature gradient within that zone should be 5.4°F or less, and average room airflow velocities should be maintained at 50 fpm or less. The ASHRAE Handbook says that discharge air temperature from overhead air devices should be no greater than 15°F higher than the space design set point, or the space will no longer comply with the stratification limits.

By handling the sensible load within the space, chilled beam technology provides a way to decrease the volume of supply air required to the minimum ventilation rate. The outdoor air that is supplied, though, must be treated for temperature and humidity and be supplied at temperatures of roughly 54°F - 57°F for cooling and 65°F - 68°F for heating. The dew point must be low enough to handle the latent load in the space; sometimes this is as low as 40°F to result in a comfortable RH within the space, typically around 50% for most applications. When sizing, assume the secondary (induction) air is equal in temperature and RH with that maintained within the occupied zone, unless specific functions or conditions within the space would dictate otherwise. Supply airflow rates should be high enough to control internal humidity loads, and to maintain discharge velocities that are high enough to help prevent condensation in cooling and stratification in heating. The velocity should also be sufficient to ensure that the supply air jet stays attached to the ceiling and does not drop prematurely into the occupied zone. Jets of two parallel beams should not collide at high velocities. When exposed active beams or passive beams are utilized, the space between the beam and the ceiling should be adequate to allow for secondary air to enter the beam from above.

Window exposures that result in drafts due to radiation and downward convective air movement in cold seasons, heat generating devices, and other factors can cause convective flows that have an impact on air jet direction and air distribution; these are forces that should be considered during design. In addition, spacing and distribution of chilled beams within a space should be such that avoids the risk of excessive jet collision that will cause or exacerbate draft conditions. Passive beams should be located in a space in such a way that they are not directly over workstations where the occupants will remain stationary. The temperature and velocity of the descending air stream could cause occupant discomfort. Heating should not be attempted with passive chilled beams; it is ineffective due to stratification.

Evaluate The Structural & Architectural Design

When deciding the type of air distribution system to utilize in design, it is important to understand the current layout of the space and the potential for future space design and use modifications, such as the addition or removal of partition walls or a change in occupancy that would result in an increase or decrease in load. If future fluctuations in load are anticipated, adaptable chilled beams should be used and piping systems should be designed for ease of those load and zone changes. Whether installed in lay-in or exposed ceilings, chilled beams must be supported independently from the ceiling. If they are to be installed in a ceiling grid, once supported independently, they can be positioned to the grid. When necessary, attention should be taken to comply with seismic codes and safety recommendations. Structure must be considered for support and safety, and understanding what features must also be installed in the ceiling such as lighting, sound, and fire suppression equipment should be considered in order to maintain architectural consistency. In some cases, multiservice chilled beams are the most appropriate choice when all of these factors must be considered.

Evaluate the Type of System(s) Available for Design

Determining whether or not chilled beams are an appropriate design solution requires knowledge of the operating temperatures and efficiencies of equipment and systems that will be providing air and water to the beams. If a chilled beam system will be handling sensible cooling only, a two-pipe system is sufficient; if heating will be required, the best practice for most designs would be to use a four-pipe system. As with any system requiring water, supply water piping must be insulated to prevent condensation and decrease energy loss.

Water temperatures being supplied to chilled beams are more moderate than those being supplied to many other types of equipment such as air handlers, fan coil units, and terminal units; for cooling, the supply water temperature should be at least 55°F (or more than 2°F above dew point) and for heating it should, at the extreme, be absolutely no higher than 140°F. Standard chilled water supply temperatures are frequently 40°-45°F and hot water temperatures are often 180°F. With the varied temperatures water must be supplied either by equipment dedicated to serve only the chilled beam system (chiller/boiler), or the system must be designed with a secondary loop supplying tempered water to the beams.

Selection & Design Guidelines

In addition to water temperatures and piping system type, pumps, storage tanks, air separators, strainers, valves, and other specialty items must be sized to handle the flow rates and pressure drops associated with supplying water to the chilled beam system. Major on-site piping and duct modifications should be avoided or carefully supervised by the engineer to ensure that associated pressure changes do not exceed the capabilities and efficiencies of the supplying fans and pumps.

Evaluate Heating, Cooling, & Climate Considerations

While chilled beam designs are appropriate and can offer energy savings and comfort in many conditions, designing with this technology may not be ideal in some conditions. It is important to evaluate heating, cooling, and climate specific requirements before designing with chilled beams.

Some regions have moderate climates with such low temperature variation throughout the year that outdoor designs are frequently used. In these climates, open courtyards, atria, outdoor entry ways, and large window exposures are common. Given the fact that chilled beam designs result in chilled water coils being directly exposed to in-space temperature and humidity, drastic changes in space conditions that produce large temperature swings or increases in latent load should be avoided. It is imperative to understand the impact of architectural design on temperature and load requirements, and whether or not the use of chilled beams can accommodate these conditions.

During construction it is sometimes necessary to turn on air conditioning systems to provide comfortable conditions for workers and/or to maintain equipment and materials within the space. During construction and building commissioning, water should not be supplied to chilled beams until the dew-point temperature within the space is below the chilled water supply temperature. Furthermore, if the space(s) is still “open” to outside construction and subject to humidity and temperature swings, alternate methods of conditioning the space should be considered until the building envelope is sealed.

The table below gives general design condition recommendations. Please note that these may vary based upon climates, code, and design requirements.

Important to Note

- Supply water temperatures must be maintained above the specific design space dew point temperature to prevent condensation. If chilled beams are installed in areas where space humidity and temperature are not readily controllable, supply water temperatures should be elevated. Passive beam water supply temperatures should be maintained at least 1°F - 2°F above room dew point temperature.
- Chilled beam chamber pressure is dependent upon manufacturer design; effective length, nozzle size, and velocity controller (HVC) position are selectable design variables. It is important to note changes in flow and capacity as these options are selected.
- In heating mode, measure room temperature at the ceiling.

Evaluate Operational Considerations & Methods of Control

Control system selection involves determining the type of controls necessary and available, for example: BMS or thermostat control, occupancy sensors, two or three way water valves, condensate or dew-point sensors, etc.. When chilled beams are operable, if control of the space dew-point temperature cannot be guaranteed, sensors and controls are even more important and should be integrated into the design accordingly.

Condensate sensors attached to chilled water supply pipes sense moisture and signal the valve to the coil to close, discontinuing water flow until it can no longer sense moisture. Although this is a cost effective method of control, over-sensitivity to moisture may occur causing the valve to remain closed for much of the operating time of the beam, yielding the potential comfort and energy benefits of the beam less effective. Controlling occupant’s ability to open the windows may reduce this risk; window sensors and controls may be utilized in designs with operable windows, especially in high humidity environments, where the beam being exposed to large volumes of untreated air could cause condensation.

RECOMMENDED DESIGN VALUES

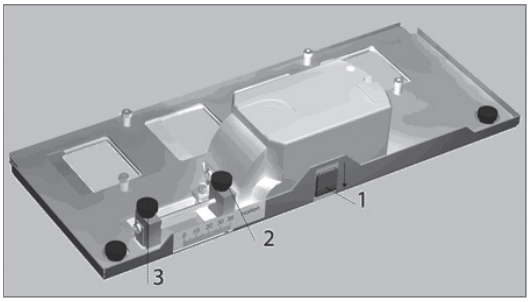
	Active Cooling	Active Heating	Passive Cooling
Room Air Temperature	73° - 77°F	68° - 72°F	73° - 77°F
Supply Air Temperature	55° - 66°F	55° - 66°F	N/A
Water Inlet Temperature	1-2° Over Space DPT	95° - 130°F	+2° Over Space DPT
Duct Pressure Level	.28 - .48” WC	.28 - .48” WC	N/A
Water Flow Rate	.32 - 1.58 GPM	.16 - .63 GPM	.32 - 1.58 GPM
Sound Pressure Level	< 35dB(A)	< 35dB(A)	N/A
Cooling/Heating Capacity, Floor Area	25 - 38 BTU/hr/ft²	13 - 19 BTU/hr/ft²	25 - 38 BTU/hr/ft²
Cooling/Heating Capacity, Effective Unit Length	260 - 416 BTU/hr/ft	156 - 260 BTU/hr/ft	260 - 416 BTU/hr/ft
Outdoor Airflow Rate, Effective Unit Length	3.2 - 7.8 CFM/ft	3.2 - 7.8 CFM/ft	N/A

* *If chilled beams are installed in areas where space humidity and temperature are not readily controllable, supply water temperatures should be elevated.*

Space Conditions & Control

Zone control of chilled beam systems is accomplished similarly to that of traditional VAV overhead air distribution systems; similar components are required, such as space thermostats or other temperature sensors linked with the BMS, a hot water and/or chilled water control valve, and the chilled beams that will be serving the space. Instead of varying the volume of air being provided to control space temperature as with many traditional systems, the thermostat or BMS can be linked to the water valves and temperature is controlled via variable water flow. Control can be as simple as on/off operation of two-position water valves but, where more precise space temperature control is necessary, the use of proportional valves should be considered in design.

To maintain optimal comfort and air quality, air quality control may be regulated independently of temperature control. Demand-control ventilation is accomplished by use of either CO_2 sensors or occupancy sensors that can modulate supply of outdoor air. Chilled beams are well-suited for demand-based ventilation, as they are most commonly designed utilizing DOAS units and one of their primary uses is to supply the minimum ventilation rate. Adaptable chilled beams that are equipped with the *Krueger by Halton* Air Quality control, especially those that are equipped with an electric actuator, work well in a demand-control ventilation system. When the CO_2 sensor or room occupancy sensor signals the need for more outdoor air, the HAQ would open and allow more outdoor air to be diffused directly into the space without impacting the volume of air being diffused from the primary nozzles.



Motorized HAQ

COOLING

During the cooling season, the primary air supply to active chilled beams provides partially cooled air that has been sufficiently dehumidified at the supply unit. Once at the beam, the sensible capacity is achieved by the utilization of the chilled beam coils within the space. In cooling mode, if the outdoor air temperature reaches the balance temperature where the need for secondary cooling no longer exists (usually around 55°F), the quantity of air being supplied is sufficient to satisfy sensible loads, and the climate is such that the outdoor air being supplied has low moisture content, “free cooling” is possible. In such cases, the DOAS unit can supply untreated outdoor air directly to the chilled beams and no secondary treatment is required, so the water valves may remain off. Due to heat gains within the space, even though the primary air temperature is low enough to be supplied directly to the beams, it still may be necessary to provide chilled water to the beams to meet the sensible load within the space. Additional heating or cooling depending on the needs of each zone may then be provided by modulating hot and chilled water valves serving the zone.

HEATING

During the heating season, if chilled beam systems are used, primary air does not have to be heated completely at the unit; once at the beam, the full sensible capacity is achieved through heat transfer in the chilled beam coil. A major advantage of this is that in buildings that have multiple zones with varying sun exposure or interior spaces that remain warmer throughout the day, much of the heating load can be handled solely by the primary air supply being warmed at the unit. Space heating, then, can be provided on an as-needed basis by utilizing hot water as required or turning the valves off and relying solely on the tempered primary air.

TWO-PIPE VS FOUR-PIPE IN HEATING

Two-pipe and four-pipe systems operate very similarly during conditions of peak load for either heating or cooling. Where the two system’s characteristics differ is how intermediate seasonal conditions and varying loads are handled. Utilizing a two-pipe system to provide chilled and hot water to chilled beams is possible by using a changeover valve to switch from chiller to boiler supply; changeover can be complicated and is not ideal. The main benefit of this system is a reduction in first-cost investment. However, they are far less capable of handling varying loads, present particular difficulties in seasonal building maintenance, require highly trained mechanical staff to perform changeover, require changing the action of room temperature controls, and therefore, ultimately are far more costly to operate than a four-pipe system. When utilizing chilled beams in a heating design it is best to utilize a four-pipe system. Four-pipe systems are a better option, as designing this way provides availability of heating and cooling in all seasons, there is no seasonal changeover requirement, and there is much more flexibility and functionality in control. Varying sun exposures throughout buildings, four-season climates where temperatures may vary widely during fall and spring, and variable loads based on occupancy and use throughout the building are all factors that make a four-pipe system best practice of design in most cases.

For example: one DOAS unit may be supplying a space with northern exposures, southern exposures, and interior spaces. When a two-pipe system is used and the system has been changed over for heating, the beams only have hot water supply available. To satisfy the load of the interior and northern zone, the primary air being provided would be secondarily heated by the hot water in the chilled beam coils. During periods of high sun exposure, the southern zone would operate with the hot water supply valve closed. Modern construction often involves large areas of glass, frequently double glazed windows which reduce reverse transmission and increase the heating effect of radiant energy. Modern technology often results in spaces with high internal heat gains due to mechanical equipment, computers, lighting, and other sources. In situations such as these, the primary air being supplied to the chilled beam system may not be low enough in temperature to overcome the heat load. Overcooling would occur in zones with less solar or internal heat gain if the temperature of the primary air being

Space Conditions & Control

supplied were reduced. Conversely, perimeter spaces with low sun exposure, large quantities of glass, or a poor building envelope may be exposed to cooler temperatures, while people and processes on the interior create internal heat gains that necessitate cooling.

A four-pipe system offers the means to provide cooling in zones with high heat gains while simultaneously providing heating in other zones and does not require major maintenance efforts for changeover.

Two-Pipe Systems

When utilizing a two-pipe system to provide heating and cooling, a changeover system must be used. Changeover temperature is the outdoor temperature point at which heat gains in each space within a design can be accommodated for by the combination of the cool outdoor primary air and the transmission loss. At or above the changeover temperature, additional cooling is required; below this temperature, no additional means are required to cool outdoor air. In order to calculate changeover temperature the room with the lowest changeover point (t_r) should be identified. This is typically the room in the zone with the hottest temperatures due to exposures resulting in solar heat gains or internal heat gains.

To estimate the changeover temperature (at sea level):

$$t_{co} = t_r - \frac{q_{is} + q_{es} - 1.1Q_p\,(t_r - t_p)}{\Delta q_{td}}$$

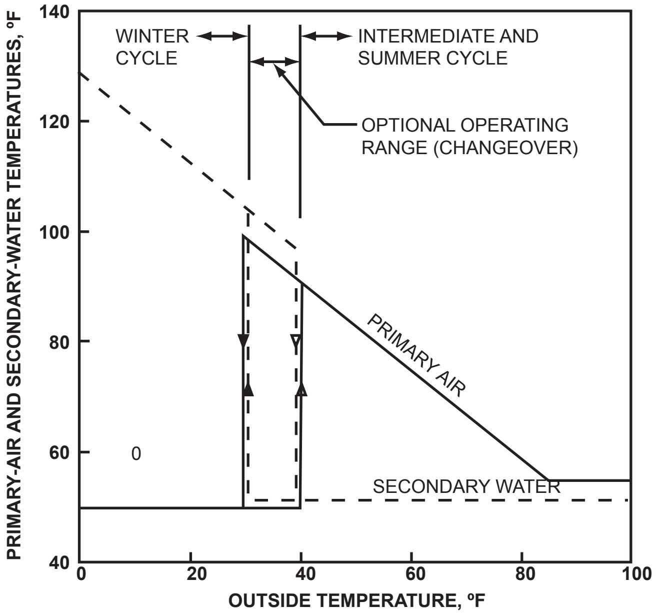
Where:
t_{co} = Temperature of changeover point, °F.
t_r = Room temperature at time of changeover, normally 72°F.
t_p = Primary-air temperature at unit after system is changed over, normally 56°F.
Q_p = Primary-air quantity, cfm.
q_{is} = Internal sensible heat gain, Btu/h.
q_{es} = External sensible heat gain, Btu/h.
Δq_{td} = Heat transmission per degree of temperature difference between room and outdoor air.

* *Carrier, 1965 - FROM ASHRAE Handbook 2012 - HVAC Systems and Equipment, Chapter 5.*

When the outdoor temperature reaches the changeover temperature, changeover should occur. Chilled water should be supplied to chilled beam systems in Spring, Summer, and Fall, being changed over to hot water for Winter only. When cool outdoor conditions are experienced in the Spring and Fall, chilled water flow to the beams should be turned off and primary air supply temperatures should be elevated to prevent over-cooling in spaces that have lower cooling loads. In non-changeover designs, when cool conditions are experienced the same method should be employed. In addition, to maintain warmer temperatures during cool evenings and nights, 100% return air may be utilized as primary supply to the unoccupied space.

Changeover temperatures and conditions should be well communicated with building owners and facility maintenance personnel to prevent efficiency losses and potential damage if changeover is neglected.

CHANGEOVER TEMPERATURE



Temperature variation for a system operating with changeover, indicating the relative temperature of the primary air and secondary water throughout the year and the changeover temperature range. The solid arrows show the temperature variation when changing over from the summer to the winter cycle. The open arrows show the variation when going from the winter to the summer cycle." – ASHRAE Handbook 2012 – HVAC Systems and Equipment Ch 5.

Four-Pipe Systems

Understanding the changeover point is important for designs utilizing a four-pipe system, but mostly from the perspective of control. A control sequence may need to be in place that allows the ability to signal for chilled water flow to beams in spaces with high sun exposure, even when supplying low temperature primary outdoor air.

Four-pipe systems are practical and adaptable for climates with varying loads, are efficient, have low operating costs, and are simple to understand and operate. Heating and cooling is operated in sequence by the same thermostat, with the supply of hot and cold water varying based on demand. A modulating control valve on the inlet of the coil is capable of allowing variable flow of either hot or cold water to the chilled beam coil, based upon the signal from the thermostat or air temperature sensor within the space that calls for either heating or cooling. Hot and cold water are never provided simultaneously. If no additional heating or cooling is required the flow is shut off.

Design Procedures

ACTIVE CHILLED BEAM DESIGN PROCEDURES

Although passive beams have no active airflow supply, it is still important to understand temperature, humidity, and air movement within the space for proper selection of either active or passive chilled beam systems.

Determine the ventilation requirement. Reference ASHRAE Standard 62-2004 - Minimum fresh air flow rate.

$$Q_{oz} = R_pP_z + R_aA_z$$

Where:

Q_{oz} = Air volume required to meet minimum outdoor airflow rate.
R_p = Outdoor airflow rate required per person.
P_z = Zone population based on the largest number of people expected to occupy the zone during typical usage.
R_a = Outdoor airflow rate required per unit area.
A_z = Zone floor area of the zone in ft².

Find the required supply dew-point temperature to remove the latent load. Reference ASHRAE Humidity Control Design Guide (ASHRAE, 2007).

$$q_L = C_lQ_s\Delta W$$

Where:

q_L = Latent load in Btu/h
C_l = Air latent heat factor in Btu/hcfm
Q_s = Supply air volume
ΔW = Difference in humidity ratio between the supply air and room condition. in $lb_{m,wv}$ / $lb^{m,DA}$

Determine the supply air volume. Required supply air volumes are the maximum required by code for ventilation and the volume required to handle the latent load.

$$Q_s = \max[Q_{oz}, Q_L]$$

Where:

Q_s = Supply air volume sensible capacity.
Q_{oz} = Air volume required to meet minimum outdoor airflow rate.
Q_L = Air volume required to control latent load.

Calculate the sensible cooling capacity of the supply air.

$$q_{S,air} = \rho c_pQ_{air}\Delta t_{air}$$

Where:

$q_{S,air}$ = Supply air sensible capacity.
ρc_p = Air volume required to meet minimum outdoor airflow rate.
Q_{air} = Air volume required to control latent load.
Δt_{air} = Difference between supply air and room control temperatures.

Calculate the sensible cooling required from the water side. The sensible cooling required from the water side is equal to the total sensible cooling capacity required less the sensible cooling capacity of the supply air.

$$q_{S,hydronic} = q_{total\text{-}q} - q_{s,air}$$

Where:

$q_{S,hydronic}$ = Supply water sensible capacity.
$q_{total\text{-}q}$ = Total capacity.
$q_{s,air}$ = Supply air sensible capacity.

Select and lay out chilled beams with careful consideration to the design considerations previously mentioned, including airflow patterns that will avoid draft, providing enough capacity to satisfy sensible loads, and ensuring occupant comfort.

Krueger's engineering selection program is an excellent tool for beam selection.

Theoretical Background / Principles of Operation

HEAT TRANSFER PRINCIPLES

Thermal energy, also known as heat, moves from higher temperature regions to lower temperature regions. This is known as “Heat Transfer”, and occurs by means of conduction, convection, radiation, or a combination of any of the three. When considering heat transfer theory, it is important to understand that although the methods may differ there are other factors that apply to all three, such as:

- Heat lost or gained may be expressed in “BTU”, or British Thermal Units, which is the amount of energy required to raise one pound of water one degree Fahrenheit. Coefficients used to estimate the value of the heat transfer include:
  - K-Factor – The Thermal Conductivity Factor is the measure of a material’s ability to transfer heat. Materials which transfer heat readily have high K-Factors; that is, they are highly conductive. The K-Factor is a measure of heat transfer in BTU/h that will pass through 1 sq. ft. of material, 1” thick with a 1°F temperature difference between the two surfaces.
  - C-Factor – The Thermal Conductance Factor is the measure of heat transfer in BTUs which will pass through 1 sq. ft. of material with a 1°F temperature difference between the two surfaces. The C-Factor for a material 1” thick would be equal to the K-Factor of the same material; the C-Factor of the same material at three inches thick would be 1/3 of the K-Factor.
  - R Value – The Thermal Resistance Value is the measure of the ability of a material to slow heat flow. The higher the R Value, the better the insulating properties of a material and the less conductive the material is. This is measured as the reciprocal of conductance. To determine R Value, divide the thickness of an insulator by its K-Factor (R=thickness/K) or calculate the reciprocal of C (R=1/C)
  - U Coefficient – “U” is the overall coefficient of conductivity, determined by adding the C-Factors of various materials and any applicable calculated C-Factors of air spaces. Higher U-Factors indicate higher conductivity, and thus lower resistance and poorer insulation values. $U=C_1+C_2+C_3+\ldots C_n$.
- The greater the difference in temperature (expressed as ΔT), the greater the amount of heat transferred.
- Time and surface area are directly proportional to the amount of heat that is transferred.
- Thermal resistance of the materials involved in heat transfer has an impact on the rate of heat transfer.

Conduction

Thermal conduction is the mechanism of heat transfer that occurs due to molecular movement within a material, without any movement of the material itself. Conduction is the only method by which energy may be transferred through a solid. The rate of conduction heat transfer is expressed with the following equation:

$$q = K\left(\frac{(t_{S1} - t_{S2})A_C}{L}\right)$$

Where:
q_s = Rate of conductive heat transfer.
K = Thermal conductivity of conductive surface.
L = Thickness of conductive surface.
A_C = Area of conductive surface.
t_{S1} = Temperature on one side of surface.
t_{S2} = Temperature on opposite surface.

Convection

Thermal convection is the transfer of heat that occurs via a similar mechanism to conduction, but with the transfer of energy being between the surface and a fluid in motion. The two types of convection are forced convection whereby the motion of the fluid is caused by an external force such as a fan, pump, wind, etc., and free/natural convection whereby the motion of the fluid is caused by buoyant forces such as when colder air falls and warmer air rises. The rate of convection heat transfer is expressed with the following equation:

$$q = H_cA_s(t_s-t_\infty)$$

Where:
q_s = Rate of conductive heat transfer.
H_C = Heat transfer coefficient in BTU/(h•ft²)°F.
A_S = Area of surface.
t_S = Temperature of surface.
t_∞ = Fluid temperature.

Radiation

Thermal radiation occurs when matter emits thermal radiation at its surface, in the form of photons of varying frequency. Radiation differs from conduction and convection, as both of the aforementioned methods of heat transfer require a material substrate whereas radiation requires no medium for photon transport and, in fact, can be impeded or prevented if the two surfaces cannot “see” each other. The net energy exchange rate is dependent upon the relative size, orientation and shape, temperature, and emissivity and absorptivity of the two surfaces.

Each method of heat transfer has an influence on an individual’s perception of heating and cooling comfort.

On the following page are some equations commonly used for heating and cooling load calculations in a space.

Theoretical Background / Principles of Operation

EQUATIONS

Total Heat Transmission, Structure with Multiple Skin Materials

$$U = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3} + \dots \frac{1}{R_n}$$

Where:
U = Material conductivity.
R = Thermal resistance.

Heat Transfer through a Window or Wall

$$Q = U * A (t_1 - t_2)$$

Where:
Q = Btu/H.
U = Material conductivity.
A = Area in feet.
t_1 & t_2 = Temperatures in °F

Exterior Surfaces, Cooling

$$Q = U * A * \Delta T$$

Where:
Q = Heat load in Btu/H.
U = Material conductivity.
A = Area in feet.
ΔT = Temperature difference across the component under consideration.

Heating and Non-exterior Cooling

$$Q = U * A * \Delta T$$

Where:
Q = Heat load in Btu/H.
U = Material conductivity.
A = Area in feet.
ΔT = Temperature difference in °F between indoors and outdoors across the component under consideration, taking into account the combined effect of radiation, time lag, storage and temperature.

Sensible Heating and Cooling Load

$$Q = 1.08 * cfm * \Delta t$$

Where:
Q = Heat load in Btu/H.
1.08 = Constant for density at sea level cfm is the volume of conditioned air.
cfm = Volume of airflow calculated by area in square feet. x velocity in feet per minute.
Δt = Temperature difference between the supply air and the room control temperature.

Latent Cooling Load

$$Q = 0.68 * cfm * GR$$

Where:
Q = Load in Btu/H.
0.68 = Latent load constant.
cfm = Volume of airflow calculated by area in square feet x velocity in feet per minute.
GR = Difference between absolute humidity between indoor humidity/area and outdoor humidity/area.

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PSYCHROMETRICS

The term “Psychrometrics” relates to the understanding and use of an instrument (psychrometer) to determine atmospheric humidity by the reading of two thermometers; one of these readings would come from a thermometer with a bulb or “wick” that is kept moist, the other a standard or “dry” bulb reading. Data regarding psychrometrics can be found on a psychrometric chart, which includes the following properties for moist air:

- Dry Bulb Temperature
- Wet Bulb Temperature
- Relative Humidity, or “RH”
- Dew Point Temperature
- Humidity Ratio
- Total Heat (Enthalpy)
- Specific Volume

To truly understand the study of psychrometrics, you must understand the terms involved.

DRY BULB TEMPERATURE

Dry bulb temperature is the temperature of a substance as read by a common thermometer. This is an indication of the sensible heat – the type of heat that causes a change in temperature as heat is added or removed but causes no change in state. Latent heat, on the other hand, causes a change of state but involves no change in temperature. The changes in state associated with latent temperature are:

- Freezing - Removal of heat resulting in a state change from liquid to solid.
- Melting - Addition of heat resulting in a state change from solid to liquid.
- Vaporization - Addition of heat resulting in a state change from liquid to vapor.
- Condensation - Removal of heat resulting in a state change from vapor to liquid.

Note that a substance requires the same amount of heat for each change of state; if one BTU is required to freeze one pound of water, one BTU is also required to melt one pound of ice. The same principle applies for boiling and condensing. This rule is consistent for all substances. On a psychrometrics chart, dry bulb is shown by vertical lines originating from the “x” axis on the bottom of the chart. More detail on sensible and latent heat is contained at the end of this article.

WET BULB TEMPERATURE

Wet bulb temperature is the temperature of a substance as read by a thermometer that has a wet wick over its sensing bulb, and is used to measure the water content of moist air. The drier the air, the more water will evaporate from the wick, which lowers the reading on the thermometer. For example, according to the National Climate Data Center’s data on annual average wet-bulb temperatures, between 1996-2010 the average wet-bulb temperature in a very humid climate such as Honolulu Hawaii was 69.5. In contrast, the average wet-bulb temperature in a more arid climate such as Las Vegas, Nevada, was 50.8. If the relative humidity of the air is 100%, the air is saturated and the dry bulb and wet bulb temperatures will be equal. Slanted similarly to enthalpy lines (yet not exactly parallel), wet bulb lines on a psychrometric chart originate from where the dry bulb lines intersect the saturation line, and slope down and to the right.

RELATIVE HUMIDITY

Relative Humidity is the ratio of water vapor pressure in a given sample of air to the water vapor pressure that saturated air at the same temperature can hold. Achieving 30-35% RH for heating conditions and 45-60% for cooling conditions yields optimal space comfort conditions.

$$\text{RH} = \frac{\text{Amount of Moisture Air IS Holding}}{\text{Amount of Moisture Air CAN Hold}}$$

The 100% RH line is the saturation line. Relative humidity lines at less than saturated conditions fall below and to the right of the saturation line.

DEW POINT TEMPERATURE

Dew Point Temperature (DPT) is the temperature to which air must be cooled before condensation is possible. As heat is removed from air, the relative humidity of the air increases until it reaches 100%, or saturation. The temperature at which this occurs is the dew point. At saturation, the dew point, dry bulb, and wet bulb temperatures are equivalent. If warmer air (air that still contains moisture) is passed over a surface that is below the air’s dew point temperature, the moisture in the air will condense on the surface. Understanding the interaction between dew point and surface temperature is important in determining and preventing problems that may be associated with condensation occurring in an HVAC system. Dew point temperatures are shown on the saturation line.

If the surface temperature of an object is equal to or lower than the DPT, the surface will form condensation. This is an important concept to understand in the design of chilled beams; if the surface of the coil and associated piping is 55° as a result of the chilled water temperature, and the DPT of the space is 60°, condensation may occur.

HUMIDITY RATIO

Humidity ratio (W), also known as “specific humidity”, is the actual weight of water vapor per pound of dry air and is expressed in pounds or grains. 7,000 grains is equal to one pound of water.

$$\text{Humidity Ratio} = \frac{\text{Pounds of Moisture}}{\text{Pounds of Dry Air}}$$

Humidity ratio lines on a psychrometric chart originate at the vertical axis on the right side and run horizontally across.

ENTHALPY

Enthalpy refers to the total heat of a substance, expressed in British Thermal Units (BTU) per pound. If air is moist, enthalpy indicates the total heat in the air and water vapor mixture and is shown in BTU per pound of dry air. Dry air at 0ºF has a total enthalpy of 0 BTU/lb. Enthalpy values are found on a psychrometric chart on a scale above and left of the saturation line. Lines with constant enthalpy slope down and to the right

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and appear to be, although they are not precisely, parallel to the wet bulb lines.

SPECIFIC VOLUME

Specific Volume (SpV) is the reciprocal of density. Density is expressed in unit of mass per unit of volume. Specific density is expressed as cubic feet of air-water vapor mixture per pound of dry air. D= M/V where SpV= V/M where D = density, M = mass, V= volume, and SpV = specific volume. Lines of specific volume slope up and to the left of their origin, the horizontal axis. See TABLE 1 for specific volume correction factors at altitude.

Understanding how air density, for which the constant is 1.08 at sea level (SEE TABLE 2), relates to mass and volume is helpful in understanding many of the above properties that are calculated according to air masses, densities, and volumes. It is also helpful in understanding the impact of altitude on the properties of air. Standard psychrometric charts are calculated at sea level, but are often provided with the adjustments for a specific altitude in locations that frequently encounter designs at higher altitude conditions. In the heat load calculation formula, the air density ratio at altitude should be multiplied by the air density constant 1.08.

BTU / h = 1.08(ADR)*CFM*∆T

TABLE 1: VOLUME CORRECTIONS AT ALTITUDE

Altitude in Feet	Volume Correction Factor
0	1.00
1600	1.05
3300	1.11
5000	1.17
6600	1.24
8200	1.31

SENSIBLE AND LATENT HEAT

As previously discussed, sensible heating is that which causes an increase in temperature, whereas latent heat only causes a change in state. Latent and sensible are terms also used in reference to heating and cooling capacities. Total capacity is the sensible capacity, which is the capacity required to lower the temperature without effecting the moisture content of the air, along with latent capacity, which involves the capacity to remove the moisture from the air. As latent heat increases, moisture content increases. For example, water heated to 212ºF will maintain that temperature even as heat is added. The temperature will not increase, but the water will vaporize. In regards to cooling: the continued removal of latent heat from the water at the freezing point will result in a decrease of moisture content and the change in state from liquid to solid, but will not lower the sensible temperature any further.

Sensible heat factor is the ratio of sensible heat to total heat. Utilizing a psychrometric chart, you can determine that the enthalpy for return air entering a cooling coil at 76ºF at 50% RH is 28.7 by plotting the status point defined by the two parameters of temperature and humidity. The enthalpy for the resultant saturation temperature of 55ºF @ 80% RH is 21.1. Subtracting 21.1 from 28.7 determines the total heat in this example - 7.6. Plotting the wet bulb temperatures, you can determine that the enthalpy at the wet bulb intersect at 60ºF is 26.5. To determine sensible heat, subtract the saturation temperature enthalpy 21.1 from the wet bulb enthalpy 26.5. Finally, to determine the sensible heat factor, divide the sensible heat 5.4 by the total heat 7.6, which results in a sensible heat factor of 0.71. Sensible heat factors are generally higher than .5 because cooling processes typically remove more sensible than latent heat. Sensible processes are typically shown in horizontal paths on the psych chart, vs. latent which are typically shown in vertical paths. Most of the processes that involve both result in angled or diagonal paths.

TABLE 2: AIR DENSITY RATIO AT ALTITUDE

Altitude (Ft)	Air Density Ratio (Altitude/Sea Level)	Temperature (ºF)	Barometric Pressure (mm Hg)	Barometric Pressure (in Hg)
0 ft (Sea Level)	1.0	59.0	750.0	29.53
1000	.9702	55.4	722.7	28.45
2000	.9414	51.9	696.3	27.41
3000	.9133	48.3	670.9	26.41
4000	.8862	44.7	646.4	25.45
5000	.8598	41.2	622.7	24.52
6000	.8342	37.6	599.8	23.62
7000	.8094	34.1	577.8	22.75
8000	.7853	30.5	556.6	21.91
9000	.7619	26.9	536.1	21.11
10,000	.7392	23.4	516.3	20.33
11,000	.7172	19.8	497.3	19.58
12,000	.6959	16.2	478.9	18.85
13,000	.6752	12.7	461.1	18.16
14,000	.6551	9.1	444.0	17.48

Changes in air density alter the physical and thermodynamic properties of air/water ratios in a mixture. Note that the air density is reduced by approximately 3.6% every 1,000 feet above 2,000 feet of altitude.

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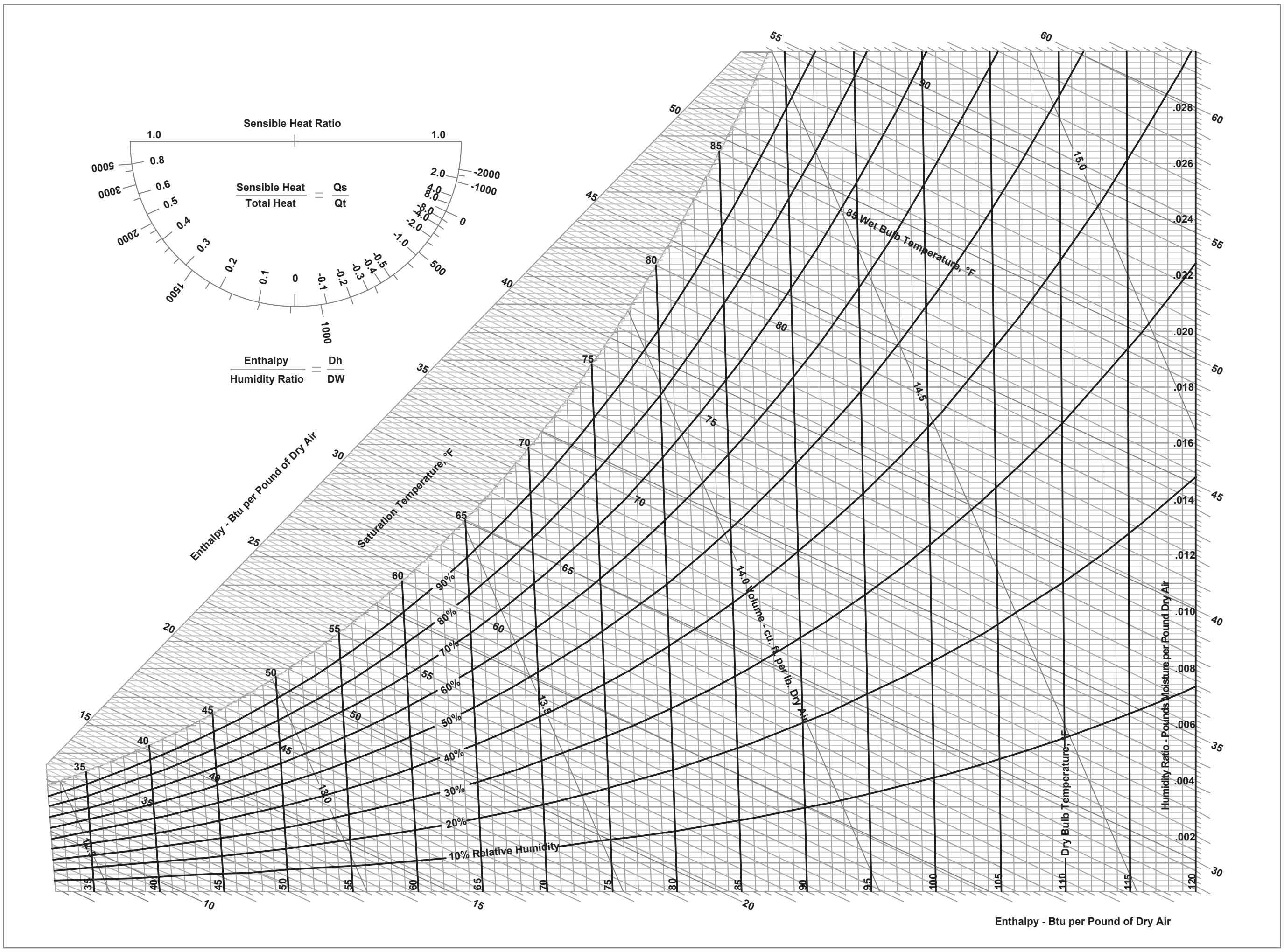
PSCHYCROMETRIC CHART



INDUCTION

Bernoulli's principle states that "the pressure of a fluid [liquid or gas] decreases as the speed of the fluid increases." Within the same fluid high-speed flow is associated with low pressure, and low-speed flow is associated with high pressure. When applied to the concept of chilled beams, this principal is key to understanding entrainment and induction.

Primary air is supplied to the pressurized plenums of active chilled beams. The plenum air is expressed through small nozzles along the plenum of the chilled beam at high velocity. This high velocity primary airflow passing the water coil creates a drop in pressure that facilitates the induction of room air across the coil, where it is then passed through the slots in the side of an active chilled beam as "mixed" air. The low volume of high pressure air distributed through the jets entrains low pressure around it, resulting in a larger volume of low pressure air being mixed with the small volume of high pressure air and being distributed into the space.

The airflow rate of an active chilled beam depends upon the effective length, the chilled beam chamber pressure, and the nozzle size. Increasing the velocity of the air jet increases induction; therefore, utilizing the same airflow rate in a beam of a given length but reducing the nozzle size will increase the induction ratio. This is one way to increase the capacity of a beam of a given length and airflow; however, the pressure and sound increase as well to varying degrees. To a lesser degree, changing the position of the *Krueger by Halton* Velocity Controller does the same thing by increasing the discharge velocity of the mixed air through the face of the diffuser. Utilizing these methods to increase induction air and capacity may also result in the need to decrease primary air quantities. It is important to satisfy the minimum ventilation requirement, maintain pressures that can be handled by the designed equipment, and meet sound requirements; following recommended methods of design will ensure appropriate attention to these factors for ideal system design. There is no "set" induction ratio for all chilled beams. In general, induction ratios tend to vary between 3:1 and 8:1 depending on product design, manufacturing, and application.

The volume of induced air is equal to the volume of the primary air multiplied by the induction ratio minus 1.

$\mathrm{CFM}_{\mathrm{induced}} = \mathrm{CFM}_{\mathrm{primary}}*(\mathrm{Induction\ Ratio} - 1)$

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The sensible cooling capacity of the coil may be expressed as such:

$$q_{coil} \leq \frac{T_{room} - EWT}{\left(\frac{1}{1.08*CFM_{induced}} + \frac{1}{500*GPM}\right)}$$

For most appropriate beam designs, to achieve the energy efficiency, sound levels, and sensible comfort that chilled beams are intended for, the sensible cooling capacity of the coil (q_{coil}) will be much less, rather than equal to, the calculated “potential” capacity. High capacity beam selections that strive to achieve much greater capacity with fewer beams frequently result in over-cooling, improper mixture, and due to elevated air flows - higher NC levels; this may completely defeat the intended energy efficiency and comfort control design purposes of chilled beams.

References

ASHRAE Standard 62-2004 - Minimum Fresh Air Flow Rate

ASHRAE Humidity Control Design Guide (ASHRAE, 2007)

*Carrier, 1965 - FROM ASHRAE Handbook 2012 - HVAC Systems and Equipment Chapter 5.

ASHRAE Handbook 2012 – HVAC Systems and Equipment Chapter 5.