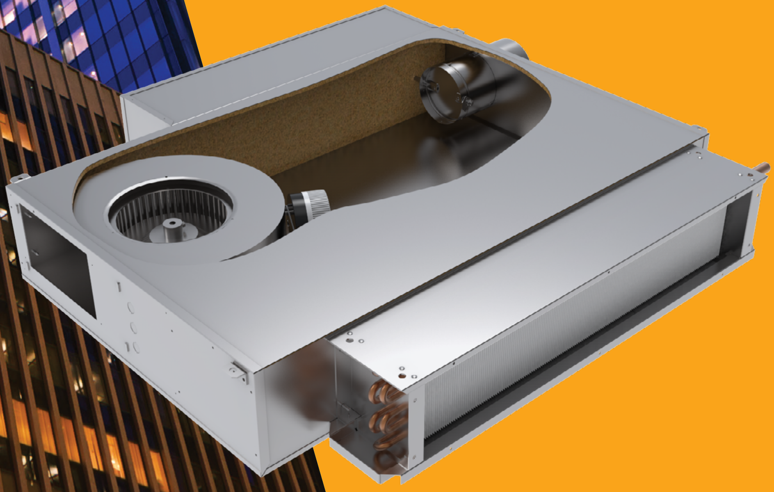
Redefine your comfort zone.™

DLSC

DESIGN AND APPLICATION GUIDE
sensible cooling fan terminals



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"The dimensional and performance data presented herein are subject to change. Please contact you local Titus representative for any updates."

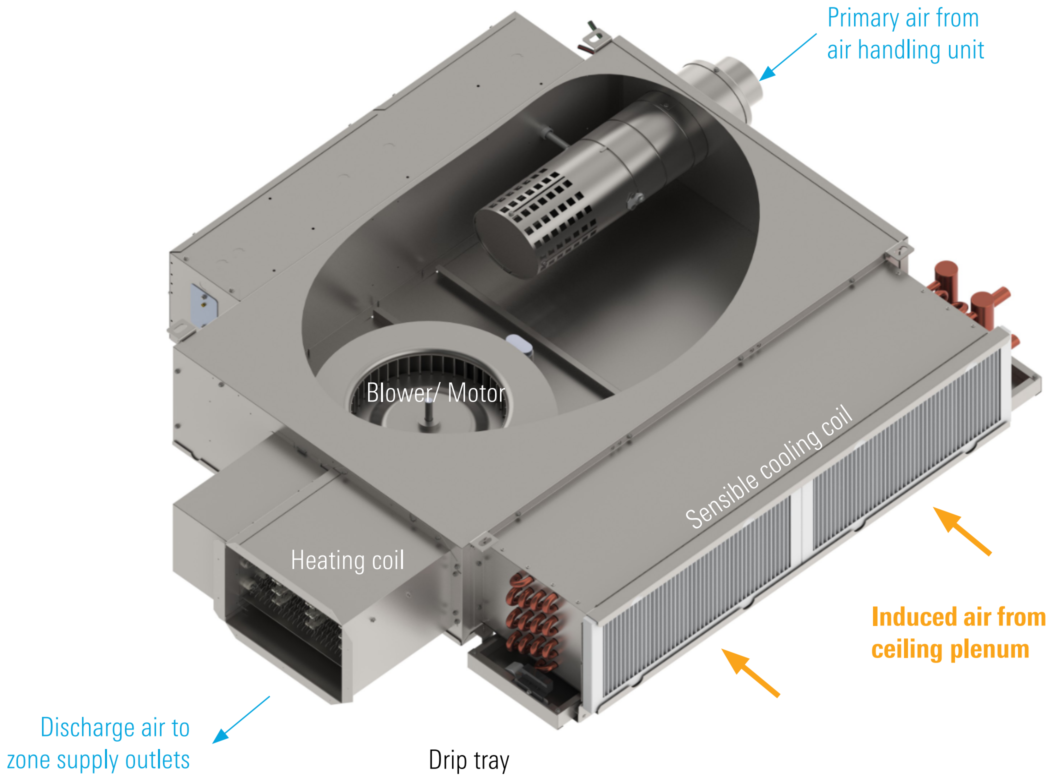


Figure 1: Diagram of a DLSC sensible cooling fan terminal

Like other air-water HVAC devices, sensible cooling fan terminals are intended to provide local zone cooling by means of a sensible heat transfer coil. Sensible cooling processes involve the addition or removal of heat without affecting the mixture's moisture content. As such, the cooling process must result in a final dry bulb temperature that remains above the air mixture's dew point temperature.

These terminals are supplied chilled water at a temperature at or above the room dew point to eliminate the possibility of condensate formation on their coil surfaces. They also incorporate pressure independent volume flow rate controllers which modulate the flow rate of pre-treated primary air supplied from a central air handling unit.

DLSC terminals (see figure 1) are essentially series type fan terminal units fitted with a sensible cooling coil on their induced air inlet. Although, it should never be required when the unit is properly applied, a drip try is provided beneath the coil. Primary air conditioned at the air handling unit is then ducted to the unit's primary air inlet. This primary air is tasked with providing ventilation as well as satisfying the zone's entire latent cooling requirement as the sensible cooling coil provides no moisture removal.

Plenum air drawn across the cooling coil supplements any sensible cooling provided by the unit's primary air. The motor/blower within the terminal may be configured to maintain either a constant or variable delivery of supply air to outlets within the zone the terminal serves. A heating coil (hydronic or electric) is often provided on the unit discharge to provide heat to the zone. Alternatively, an additional row on the induction side heat transfer coil may be dedicated to heating.

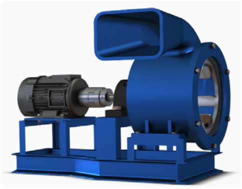
The primary air volume regulator may also be configured to provide constant or variable volume airflow but should never allow the primary airflow rate to drop below that required for sufficient space humidity control.

The information and advice that follows is intended to provide the designer of these air-water systems a keen understanding of their operation and application. Additional state of the art sizing and selection software is available through your local Titus representative and on our website www.titus-hvac.com.

The use of water as a heat transfer medium offers considerable energy savings and potentially more efficient space usage. Figure 2 illustrates the fact that a one inch diameter water pipe can transport the same amount of cooling energy as an air duct two-hundred and fifty (250) times its cross-sectional area!

Figure 2: Cooling energy transport capacity of water versus air

In addition, the transport energy required to deliver a similar amount of space cooling with air is more than six times that of doing so with water (see figure 3).



550 CFM = 1 ton

$$BHP = \frac{CFM \times SP}{6{,}356 \times FAN_{EFF}}$$

Fan BHP to move 10 tons of cooling = 5.2

4 GPM = 1 ton

$$BHP = \frac{GPM \times HD}{3{,}960 \times PUMP_{EFF}}$$

Pump BHP to move 10 tons of cooling = 0.8

Figure 3: Energy to transport cooling, air versus water

In consideration of this, air-water systems are designed to maximize the zone sensible cooling provided by their integral water coils while reducing ducted primary airflow rates to that required for proper space ventilation and latent cooling (humidity control).

Conventional all-air systems perform the zone latent and sensible cooling within their central air handling unit. Air-water systems allocate most of the sensible cooling to chilled water coils mounted in zone terminal units.

Decoupled sensible cooling systems offer HVAC energy, space usage, occupant comfort and operational cost benefits over conventional variable air volume (VAV) systems. Some of these benefits are identified below:

Space related benefits:

- Reduced primary (ducted) air requirements enable reduction of primary air ductwork capacities and cross sections by 50% or more versus all-air systems.
- Reduced central equipment capacities afford smaller mechanical room footprints that can result in 5 to 15% of additional usable floor space.

Comfort and IAQ benefits:

- Higher supply air temperatures during cooling operation contribute to increased occupant comfort levels.
- Minimal airflow throttling results in less noticeable changes to room air motion, air diffusion performance and acoustical consistency within the space.
- Employment with a DOAS (100% outside air system) ensures consistent ventilation rates and space humidity levels are maintained.
- Dry sensible cooling coils eliminate bacterial, fungal, or mold growth associated with condensation in fan coils, VRF terminals and other unitary products.

Energy Efficiency and Operational Benefits:

- Using water as the primary zone heat transfer medium often reduces annual transport costs by 40 to 60%.
- Higher supply water temperatures may afford more efficient use of water side economizers and increased opportunities for free-cooling and geothermal cooling.
- Higher condenser water temperatures result in higher chiller operational efficiencies.
- Dry-coil operation does not require upstream filtration (International Mechanical Code 2015, section 307.2)

SENSIBLE COOLING FAN TERMINALS AS A COMPLIMENT OR SUBSTITUTION FOR ACTIVE BEAMS

Sensible cooling fan terminals may be used as a substitute for active chilled beams or may compliment them in perimeter zone applications. Since active beams and sensible cooling fan terminals rely on similar primary air and chilled water temperatures, they can be supplied by the same air and water distribution systems. This enables each type of terminal to be used in applications to which it is best suited.

While active beams generally require less maintenance, sensible cooling fan terminals allow heating during unoccupied periods to be accomplished without having to operate the air handling unit that serves them. Conversely, active beams rely on their primary air supply to support induction across their heat transfer coil.

Table 1 illustrates some comparisons between active beams and sensible cooling fan terminal (SCFT) units. Both rely on similar water and primary air service temperatures. The fan terminals require less inlet pressure than the beams which rely on duct pressure to support their room air induction function. While beams are also inherently quieter, SCFT's eliminate the need for run out chilled and hot water piping beyond the zone terminal unit. Active beams require chilled and/or hot water piping connections be made to each individual device.

Attribute	Active Chilled Beams	Sensible Cooling Fan Terminals
Typical Unit Height (In.)	10	10 to 17
Chilled Water Supply Temperature	56 to 59^{0}F	56 to 59^{0}F
Primary Air supply Temperature	53 to 62^{0}F	48 to 62^{0}F
Typical Primary Airflow Rate (CFM/ft^2)	0.4	0.4
Max. Interior Space Coverage Area (ft^2/Unit)	300 to 400	1000 to 1500
Max. Perimeter Space Coverage Area (ft^2/Unit)	150 to 200	450 to 600
Typical Space NC Levels	25 to 30	30 to 40
AHU Operation During Unoccupied Heating	Required	Not Required

Table 1: Comparison of active beams to sensible cooling fan terminals

On the other hand, the downstream ductwork required by sensible cooling fan terminals is typically larger than that required with an all-air system since their discharge air temperature will rarely be cooler than 60°F. Active beam systems deliver primary air directly to the point of discharge and do not require downstream ductwork for the delivery of their discharge mixture of primary and reconditioned room air.

One advantage SCFT's have versus active beams is that their induction airflow rate can be modulated independent of their primary airflow rate. Active beams induce room air at a relatively consistent ratio to their primary airflow rate. As such, they may over cool the space when their primary air is delivered below room temperature and there is no demand for cooling (zone chilled water valve is closed). SCFT's modulate their fan airflow rate (during periods when their sensible cooling coil is inactive) to induce a sufficient volume of warm plenum air to offset the cooling delivered by their primary air.

Sensible cooling fan terminals also have an advantage when serving areas where the indoor humidity level cannot be sufficiently managed. Figure 4 shows a DLSC terminal serving an atrium area where the space humidity level cannot be assured due to uncontrolled infiltration. The terminal and its sensible cooling coil are located within a controllable humidity area and ducted to conventional supply outlets serving the atrium.

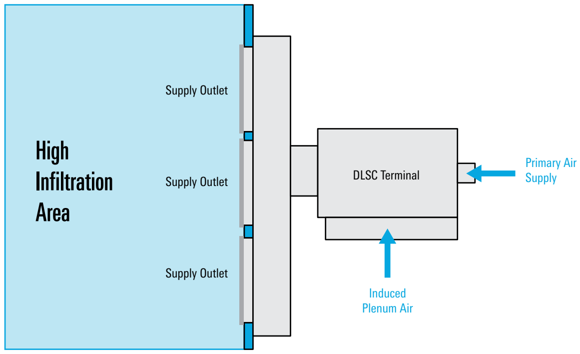


Figure 4: DLSC terminal serving a high infiltration area

One of the major elements affecting thermal energy transfer and occupant comfort is the space humidity level. Psychrometrics use thermodynamic properties to analyze conditions and processes involving moist air. A detailed study can be found in the ASHRAE Fundamentals Handbook. This chapter is a brief overview of HVAC psychrometric terms and processes and how they can best be applied to air-water system performance.

Atmospheric Air (the air that we breathe), contains many gaseous components including water vapor and contaminants. Dry Air is atmospheric air with all moisture removed and is used only as a point of reference. Moist Air is a combination of dry air and water vapor and is considered equivalent to atmospheric air for this discussion.

The psychrometric chart is a graphical representation of the thermodynamic properties of moist air. There are several charts available to cover all common environmental conditions at various altitudes.

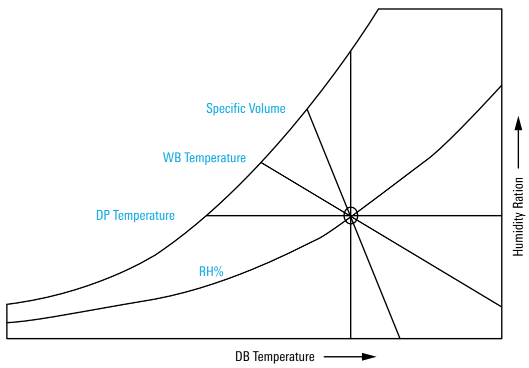


Figure 5: Psychrometric chart

Psychrometric terms relevant to air-water system design

Dry-bulb temperature (T_{DB}) is the temperature of air that may be measured using a standard thermometer. It can also be referred to as sensible temperature.

Wet-bulb temperature (T_{WB}) is measured using a 'wetted' thermometer. The combination of the dry and wet-bulb temperature of an air mixture can be plotted on the psychrometric chart to determine its moisture content.

Relative Humidity (RH) is the moisture content of an air mixture expressed as a percentage of that corresponding to fully saturated air at the same dry bulb temperature. The room air relative humidity level for optimum space comfort is 30-35% for heating conditions, and 45-60% for cooling conditions.

Absolute humidity (W) is the vapor content of an air mixture. It is described in terms of "lbm-moisture per lbm-dry air" or "grains of moisture per lbm-dry air" (where there are 7,000 grains of moisture per pound of water). It may also be referred to as moisture content or humidity ratio.

Dew Point Temperature (T_{DP}) is the temperature below which vapor begins to condense and separate from the air. It is also known as the saturation temperature and corresponds to an air mixture that is at 100% relative humidity.

Specific Volume (v) is the reciprocal of air density and is described in terms of cubic feet per pound of dry air (ft^3/pound mass). An increase in air temperature will result in a decrease in density and an increase in its specific volume. A decrease in atmospheric pressure (that generally accompanies an increase in altitude) will decrease air density while increasing its specific volume. Higher altitudes require larger motors and blowers to move the same effective mass, due to this related increase in specific volume.

Enthalpy (h) is the total heat content within an air mixture. Enthalpy is dependent on both the dry-bulb temperature and moisture content of the air and is described in terms of Btu/pound mass of the mixture.

Once two of the afore mentioned properties of the air have been defined, values for all of the other properties can be obtained from the psychrometric chart.

Psychrometric processes in HVAC systems

Four important psychrometric processes performed by HVAC systems are illustrated in figure 6 and discussed below.

The diagram on the upper left side of figure 6 illustrates a sensible heat transfer process. Shown as a horizontal path drawn from left to right on the psychrometric chart, sensible heating is a process that raises the dry-bulb temperature of air without changing its moisture content. Conversely, sensible cooling is the removal of heat without affecting the absolute humidity (W) or dew point temperature of the air mixture and is represented on the chart as a horizontal path moving from right to left. It should be noted that sensible cooling and heating processes do affect the relative humidity of the air mixture. Note that sensible heat processes have no effect on the moisture content (dew point temperature or humidity ratio) of the mixture but affect changes to the wet bulb temperature, the relative humidity, enthalpy and specific volume of the mixture.

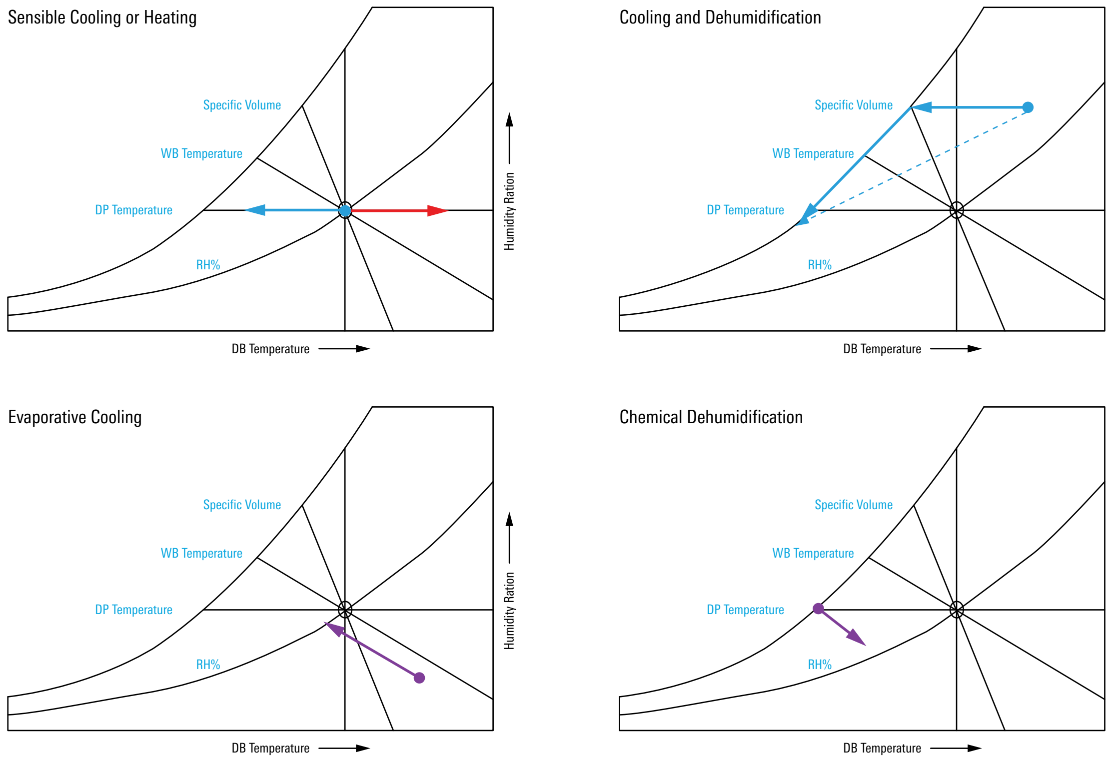


Figure 6: Common psychrometric processes

Latent cooling processes occur both within the air handling unit and the room itself.

- Latent processes are shown as vertical paths on a psychrometric chart but never really occur in HVAC systems. Instead, humid air introduced into the air handling unit is typically cooled to a dry-bulb temperature below its dew point which lowers both its dry-bulb temperature and moisture content before delivery to the space. This is actually a two- stage process. Moisture removal does not occur until the mixture reaches its dew point temperature. Further cooling (below the dew point) results in moisture removal in the form of condensation. This process, referred to as **cooling and dehumidifying** and is modeled as a diagonal line in the upper-right hand illustration of figure 6.
- Latent cooling also occurs when relatively dry air is introduced to a room with a higher moisture content. The drier air absorbs moisture as it passes through the space. This is not only important to maintaining thermal comfort but also to assuring that condensation does not form on the DLSC's sensible cooling coil surfaces.

Evaporative cooling

Dry climates like those in the desert Southwest may utilize evaporative cooling techniques to reduce the dry-bulb temperature of the outdoor air. Water sprayed into the dry outdoor air evaporates and is absorbed as water vapor by the dry air mixture. This process lowers the dry-bulb temperature of the air while increasing its moisture content. It is modeled by the diagonal line in the lower left-hand illustration of figure 6.

Chemical dehumidification

Air-water systems applied in high humidity applications may be supplied by air handling units that are fitted with desiccant dehumidification provisions. These often involve the use of a rotating wheel with a solid core made of materials (activated alumina, silica gels, zeolites, etc.) that adsorb moisture from the supply air stream leaving the cooling/dehumidifying coil, transferring it to the return air stream. The process (shown in the lower right-hand diagram of figure 6) occurs at a relatively constant enthalpy and results in moisture transfer from the supply air in exchange for sensible heat transferred from the return air.

THERMAL COMFORT BASICS

ASHRAE Standard 55-2020 *Thermal Environmental Conditions for Human Occupancy* analyzes the factors that contribute to occupant thermal comfort and establishes comfort guidelines for indoor occupancy. The Standard defines the occupied zone (see figure 7) as "the region normally occupied by people within a space, generally considered to be between the floor and 6 ft. level above the floor, more than 3.3 ft. from outside walls/ windows or fixed heating, ventilation, or air-conditioning equipment and 1 ft. from internal walls." The Standard leaves the designation of the occupied zone height to the designer as most commercial HVAC applications involve stationary occupants that are primarily seated. In that case, the height of the occupied zone is often considered to be 42 inches or 1.1 m.

Standard 55-2020 recommendations are based on ASHRAE's Comfort Tool which allows the user to input various operative temperatures and relative humidity levels as well as local air speeds. It also factors in the occupant clothing (clo) levels, a metabolic rate appropriate to the application and predicts the percentage of the population expected to express thermal dissatisfaction under those conditions.

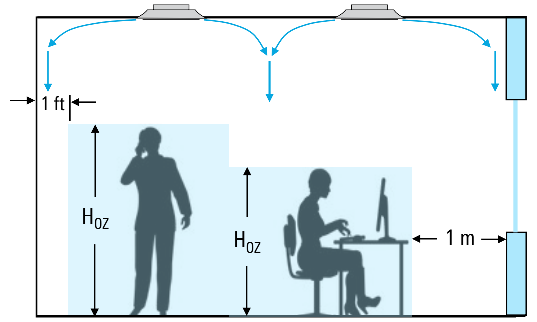


Figure 7: ASHRAE Standard 55 definition of occupied zone

Figure 8 has used the Comfort Tool to establish comfort windows for a metabolic level of 1.1 which is representative of light office work. It is also based on an average occupied zone air speed of 40 FPM. The tool has been used to predict windows that are expected to produce 90% thermal satisfaction for male (1.0 clo) and female (0.65 clo) occupants. These are clothing levels that are again representative of that worn by office workers. The Comfort Tool can be downloaded from the ASHRAE website (www.ashrae.org).

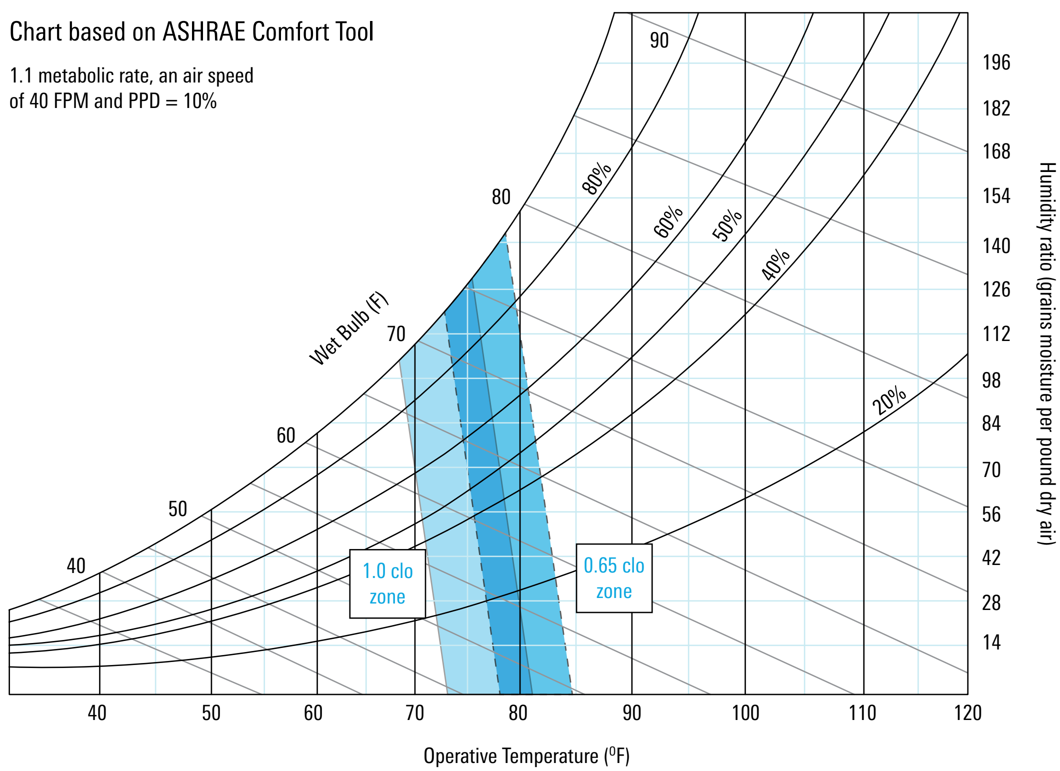


Figure 8: ASHRAE Standard 55-2020 thermal comfort recommendations

DLSC sensible cooling fan terminals are available in eight basic sizes. Sizes 1, 2, 3 and 5 are fitted with a standard capacity sensible cooling coil on the inside induction port. Sizes A, B, C and E are longer units which accommodate a larger sensible cooling coil. All units can be provided with 2, 4, or 6 row sensible cooling coils as well as 1 or 2-row hydronic or electric discharge heating coils.

All unit sizes other than sizes 5 and E feature low profile casings whose height does not exceed eleven inches. These units are ideal for applications such as Washington, DC where the ceiling plenum height is limited by local building height restrictions.

Basic dimensions for all DLSC units are presented in figure 9 below:

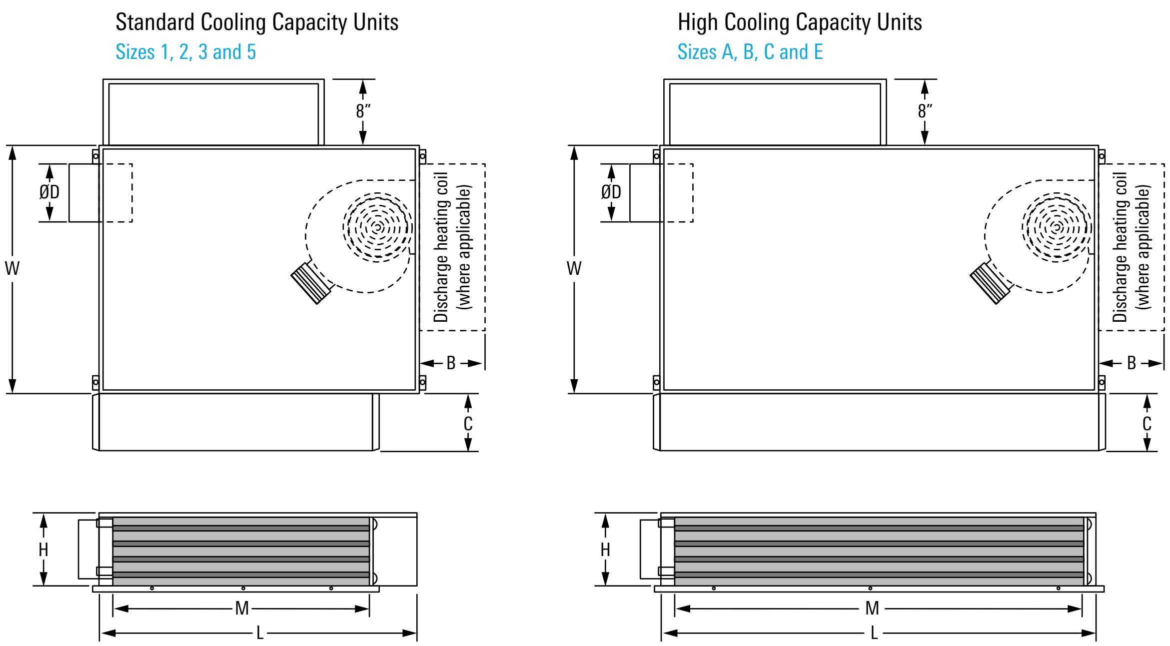


Figure 9: DLSC unit casing dimensions

<table>
<tr><th rowspan="2">Unit Size</th><th colspan="5">Basic Unit Dimension (Inches)</th><th colspan="4">Dimension C
Induction Coil Rows</th><th colspan="5">Nominal inlet size (ØD) Availability</th></tr>
<tr><th>L</th><th>W</th><th>H</th><th>M</th><th>B</th><th>2</th><th>4</th><th>6</th><th>8</th><th>4</th><th>6</th><th>8</th><th>10</th><th>12</th></tr>
<tr><td>1</td><td rowspan="2">48 1/8</td><td rowspan="2">32</td><td>8 5/8</td><td rowspan="3">38 3/4</td><td rowspan="8">9 1/2</td><td rowspan="8">7 1/8"</td><td rowspan="8">9 1/4"</td><td rowspan="8">11 1/2"</td><td rowspan="8">13 5/8"</td><td>x</td><td>x</td><td></td><td></td><td></td></tr>
<tr><td>2</td><td>9 1/2</td><td>x</td><td>x</td><td>x</td><td></td><td></td></tr>
<tr><td>3</td><td>40 1/8</td><td>26</td><td>11</td><td></td><td>x</td><td>x</td><td>x</td><td></td></tr>
<tr><td>5</td><td>46 1/8</td><td>35</td><td>17</td><td>44 3/4</td><td></td><td>x</td><td>x</td><td>x</td><td>x</td></tr>
<tr><td>A</td><td rowspan="4">66</td><td rowspan="2">32</td><td>8 5/8</td><td rowspan="4">62 3/4</td><td>x</td><td>x</td><td></td><td></td><td></td></tr>
<tr><td>B</td><td>9 1/2</td><td>x</td><td>x</td><td>x</td><td></td><td></td></tr>
<tr><td>C</td><td>26</td><td>11</td><td></td><td>x</td><td>x</td><td>x</td><td></td></tr>
<tr><td>E</td><td>35</td><td>17</td><td></td><td>x</td><td>x</td><td>x</td><td>x</td></tr>
</table>

Primary air controller

Each DLSC terminal is also provided with a primary air control valve. This primary valve is fitted with a damper and flow sensor and is controlled by a pressure independent DDC (direct digital control) volume flow rate controller. This flow controller compensates for changes in inlet pressure that might result from other system dampers opening or closing, maintaining the precise airflow required for zone conditioning at all times. This controller may be furnished by Titus or the temperature control contractor.

The controller allows minimum and maximum flow limits to be set and modulates the primary airflow rate between these limits according to the space temperature and/or dehumidification needs. Further discussion of the primary airflow modulation is included in the subsequent application chapter.

Table 2 below lists the recommended minimum and maximum primary airflow rates for each size primary air connection. The maximum flow rates, based on a 2,000 FPM inlet velocity will never likely be employed as primary airflow rates for these terminals are quite low. The minimum flow rates (based on an inlet velocity of around 350 FPM) are those which can be achieved with reasonable control accuracy. It is recommended that inlet velocities between 500 and 1,200 FPM be employed.

Primary Air CFM ranges					
Nominal inlet diameter (in.)	4	6	8	10	12
Min. CFM	30	50	100	150	200
Max. CFM	175	400	700	1,100	1,575
Recommended Range	40-120	100-240	200-450	325-675	475-1,000

Table 2: Primary airflow ranges for DLSC terminals

Blower/motor assembly

The blower/motor in the DLSC terminal operates at all times the air handling unit is energized. It delivers a mixture of primary and reconditioned plenum air to the supply outlets it serves.

Each DLSC terminal is fitted with a forward curved direct drive blower driven by a high efficiency EC motor. The motor is internally isolated to minimize vibration. Titus EC motors include an in-line inductor which not only reduces harmonic distortions but also increases the motor efficiency. Each size DLSC offers motors for the following voltage and phase power supplies:

1. 120 volts/1 phase
2. 208 volts/1 phase
3. 240 volts/1 phase
4. 277 volts/1 phase

EC motors can be programmed to provide either a constant or variable volume fan airflow delivery. Table 3 defines the motor horsepower, full load amperage, rated airflow and recommended fan minimum and maximum airflow rates for various sizes of DLSC terminals.

Additional guidance regarding the DLSC fan operation, control and sequencing is provided in Chapter 5 of this document.

Unit Size	Fan Motor HP	Full Load amps (FLA) Motor Voltage (1 phase)				Rated (maximum) fan CFM	Recommended fan CFM range	
		120	208	240	277		Min.	Max.
1	1/3	5.0	3.3	2.8	2.6	750	200	650
2						800	200	675
3	1/2	7.7	5.0	4.3	4.1	1,150	350	975
5	3/4	9.6	7.9	6.8	5.5	2,000	500	1,700
A	1/3	5.0	3.3	2.8	2.6	825	200	700
B						925	200	800
C	1/2	7.7	5.0	4.3	4.1	1,325	350	1,125
E	3/4	9.6	7.9	6.8	5.5	2,050	500	1,750

Table 3: DLSC fan characteristics and operational recommendations

Sensible cooling and heating coils

The heat transfer coils mounted on the induction port of DLSC terminals can be configured with 2, 4 or 6 rows of chilled water tubing. Coil connections can be located on either the primary inlet or discharge side of the DLSC unit (if discharge electric heaters are specified, the coil connections must be on the primary inlet side of the unit).

A drip tray is provided beneath the sensible cooling coil to catch any condensation that could occur if the zone dew point temperature rises, however, the units should never be supplied chilled water cooler than the design room dew point temperature as this could also lead to moisture carryover within the unit. Condensation prevention methods are discussed in Chapter 5.

DLSC terminals serving perimeter areas will typically require some heating capabilities. This may be provided by adding an additional row to the induction port heat transfer coil or by a separate hydronic or electric heating coil on the unit discharge. Discharge side hydronic coils may be either one or two rows.

Figure 10 illustrates a recommended piping configuration for each connection to DLSC hydronic coils. It includes an automatic water flow control valve as well as isolating valves that allow the coil to be removed in case it needs replacement. A cleanable strainer on the supply side can also double as a drain fitting.

All hydronic coils should be provided with a drain which allows its fluid to be evacuated in the event it needs flushed or replaced. A vent that allows air purging should also be provided above the coils' return connection. Drains and vents are standard on all Titus cooling and hydronic heating coils.

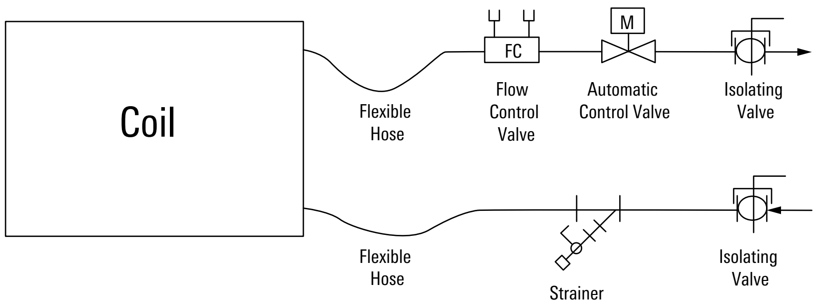


Figure 10: Hydronic coil piping

Inlet sound attenuators (optional)

DLSC terminals may be furnished with a three (3) foot long sound attenuator mounted on the inlet of their sensible cooling coil. These attenuators reduce the inlet escape noise from the fan and thus affect the unit radiated sound levels. They do not, however, affect discharge sound levels

Casing lining options

There are numerous casing liner options for DLSC terminals. The standard lining is 1/2" EcoShield™ which is composed of recycled denim fragments. This lining contains no harmful irritants or chemicals, is treated with an EPA registered antimicrobial inhibitor and meets all requirements of NFPA 90A and UL 181.

Available alternative liners are also available:

- 1/2" fiberglass lining (matte or foil faced)
- 1" EcoShield™ or fiberglass lining (matte or foil faced)
- 3/8" closed cell (non-erosive) foam liner
- 1/2" Ultra-Loc EcoShield™
- 1/2" Ultra-Loc foil faced EcoShield™ perforated inner liner

The closed cell and Ultra-Loc liners have no erosive surfaces exposed to the airstream. As the absorptive material is not directly exposed to the airstream, its sound attenuation properties are diminished. As such, use of these liners is commonly limited to healthcare and other applications where insulation particle erosion is prohibited

DLSC terminals are air-water zone cooling devices that rely on chilled water delivered at temperatures that will not result in condensation formation on their integral cooling coil. This means **all zone ventilation and latent cooling must be accomplished by pre-treated primary air** ducted from an air handling unit. While the minimum zone ventilation rate varies according to application, ASHRAE Standard 62.1 *Ventilation for Acceptable Indoor Air Quality* prescribes outdoor airflow rates for most non-residential applications. The primary airflow rate required for zone latent cooling can be calculated using equation 4.1 where LHG is the zone latent heat gain (Btu/h) and $W_{ROOM} - W_{PA}$ is the humidity ratio differential between the room and primary air expressed in grains of water per pound of dry air.

$$SCFM_{LAT} = \frac{LHG}{0.69 \times (W_{PA} - W_{ROOM})} \quad \text{equation 4.1}$$

The actual DLSC primary airflow rate is then the greater of the ventilation and latent airflow rate.

This primary air is typically delivered cooler than the space it serves and thus contributes to the zone sensible cooling as well. The remainder of the sensible cooling is provided by the DLSC terminal's integral cooling coil.

Table 4 illustrates the relationship between zone ventilation and latent airflow requirements for various office and educational facility spaces. Latent cooling loads are based on the default occupancy (at 185 Btu/h-person) plus typical moisture gains associated with infiltration in perimeter spaces. The right side of the table presents primary airflow rates that would be required to handle the latent load based on various humidity ratio differentials (ΔW) between the room and primary air. All CFM values are based on standard air and will vary with altitude. Note that only the latent airflow requirements in bold blue text represent those which are within 10% of the mandated space ventilation rates according to ASHRAE Standard 62.1 – 2019.

Application	Occupant density[1] #/1000 ft²	Ventilation Rate[2]			Latent Load[3] Btu/h-ft²	Latent Airflow Requirement (CFM/ft²) $W_{ROOM} - W_{SUPPLY}$ (grains)					
		R_P CFM/ea.	R_A CFM/ft²	Total CFM/ft²		10	12	14	16	18	20
Office Buildings											
Interior Offices	5	5.0	0.06	0.09	1.0	0.14	0.12	*0.10*	*0.09*	*0.08*	*0.07*
Perimeter Offices	5	5.0	0.06	0.09	3.3	0.47	0.39	0.34	0.29	0.26	0.24
Conference Rooms	35	5.0	0.06	0.24	7.0	1.01	0.85	0.72	0.63	0.56	0.51
Reception Areas	35	5.0	0.06	0.21	6.0	0.87	0.72	0.62	0.54	0.48	0.43
Educational Facilities											
K-12 classrooms											
Ages 5 - 8	25	10.0	0.12	0.37	5.4	0.79	0.66	0.56	0.49	0.44	*0.39*
Ages 9 plus	35	10.0	0.12	0.47	7.3	1.06	0.88	0.75	0.66	0.59	*0.53*
Libraries	10	5.0	0.12	0.17	2.7	0.39	0.32	0.28	0.24	0.21	*0.19*
Media Centers	10	10.0	0.12	0.22	2.7	0.39	0.32	0.28	*0.24*	*0.21*	*0.19*
Lecture Classrooms	65	7.5	0.06	0.55	12.8	1.86	1.55	1.33	1.16	1.03	0.93

NOTES:

1. Occupancy rates default values taken from ASHRAE Standard 62.1-2019 Ventilation for Acceptable Air Quality, table 6.1
2. Minimum ventilation rates are based on ASHRAE Standard 62.1-2019 Ventilation for Acceptable Air Quality, table 6.1
3. Typical space latent loads based on default occupancy plus typical infiltration (where applicable)

Value presented in ***bold italics*** indicate latent airflow rates within 10% of the ASHRAE mandated minimum space ventilation rate.

Table 4: Comparison of space ventilation and latent cooling requirements

AIR HANDLING UNIT CONSIDERATIONS

The air handling unit is the interface between the outdoor and indoor thermal environments. It must be capable of conditioning and delivering air at the necessary temperature, flow rate and moisture content required to provide acceptable thermal comfort and air quality for the inhabitants of the spaces it serves.

Figure 11 illustrates a conventional mixing-type air handling unit consisting of a filtration bank, an energy recovery wheel, mixing damper, cooling/dehumidifying coil and supply/ exhaust blowers. Outdoor air is drawn through the energy wheel where heat - and often moisture - is transferred to an equal amount of exhaust air. The outside air is then mixed with return air before it enters the cooling coil. The mixture is cooled to a desired dry bulb temperature and delivered at an airflow rate capable of removing sensible heat gains in the spaces it serves. In the case of all-air systems, an off-coil temperature of 50 to 52°F is commonly used to minimize the system's design airflow rate. As this leaving air temperature is typically below the mixture's dew point temperature (commonly 60 to 70°F), a significant amount of moisture is also removed in the form of condensate.

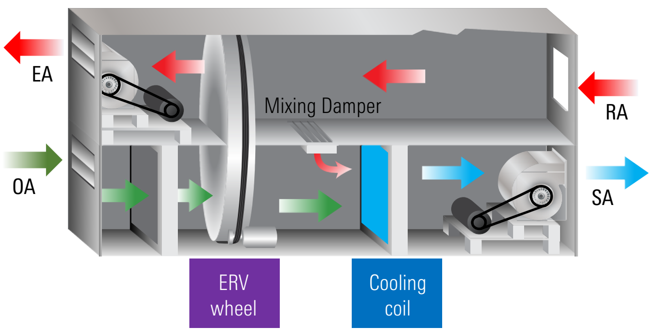


Figure 11: Mixing type air handling unit

Most chilled water air handling units are capable of supplying primary air at 49 to 50°F dew point temperatures which results in humidity ratios 10 to 15 grains lower than the spaces they serve. ASHRAE recommends that space relative humidity levels be maintained between 40 and 60% for optimal occupant comfort. At sea level and a 75°F dry bulb room temperature, this equates to a humidity range of 52 to 78 grains. While convention has been to design all-air systems for 50% RH (65 grains at a dry bulb temperature of 75°F), air-water systems perform more efficiently with a slightly higher humidity level as it increases the room to primary air humidity ratio differential resulting in a lower design primary airflow rates.

Figure 12 illustrates a DOAS type air handling unit. Although DLSC terminals are often referred to as DOAS (dedicated outdoor air) fan terminals, they may be served by either DOAS or mixing type air handling units. DOAS air handling units utilize 100% outside air and exhaust all of their return air and any indoor contaminants it might convey without mixing them with the outside air. Note that the DOAS unit depicted is configured to deliver primary air at a sufficiently low dew-point temperature in order to offset all of the zone latent heat gains. DOAS units that supplement variable refrigerant flow (VRF) and condensing fan coil devices are typically designed to deliver outside air directly to the space at room neutral temperature and humidity levels. All of the sensible and latent cooling is then performed by zone condensing cooling coils.

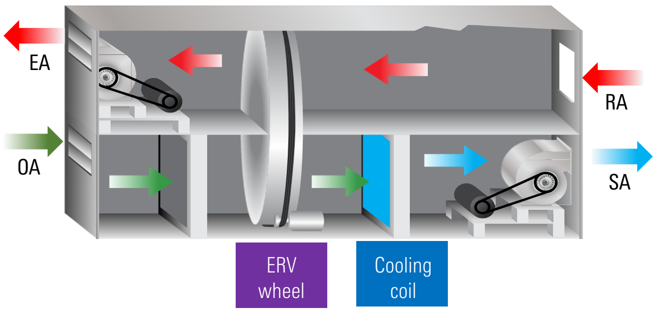


Figure 12: DOAS type air handling unit

In either case (mixing or DOAS air handling units), the cooling coil should pretreat the primary air delivery to a 51°F dew point or lower to limit the primary airflow rate required to provide adequate zone latent cooling. When air is delivered at that dew point, the primary airflow rate required to satisfy the zone latent cooling demand will usually be greater than that required to ventilate the zone. As such, the latent cooling demand almost always determines the air-water system's primary airflow rate.

Table 4 also illustrates that air-water systems may operate more efficiently in certain applications when mixing type air handling units are used. A mixture of outside and recirculated return air is usually less expensive to condition than outside air during near peak design conditions. As the coils within the DLSC terminals provide most of the space sensible cooling, the primary airflow rate can generally be reduced to that required to provide adequate space dehumidification.

The use of zone sensible cooling devices may not be effective for high occupancy applications where the latent cooling airflow rate is significantly higher than the ventilation airflow requirement. Assembly areas like concert halls, auditoriums (including the lecture classrooms detailed in table 4) and religious facilities often have latent loads that approach their sensible cooling requirements. In other cases, such as K-12 classrooms, increasing the humidity ratio differential between the room and supply air is recommended. However, achieving depressed humidity ratio differentials (more than 10 to 15 grains) often requires the use of passive desiccant technologies such as that shown in figure 13 which may significantly increase the air handling unit cost.

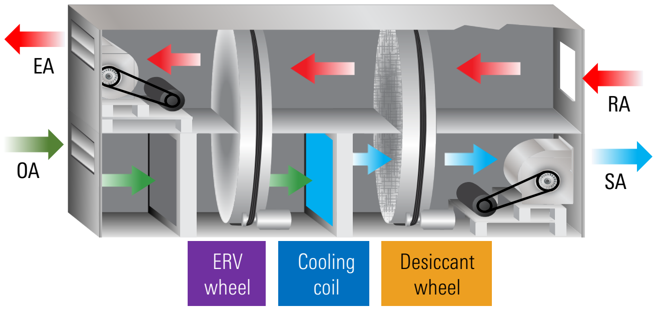


Figure 13: DOAS air handling unit with passive desiccant dehumidification

Other DOAS air handling units may employ **active desiccant** dehumidification. In these devices, the return air is superheated before entering the desiccant wheel. This significantly increases its moisture absorption capacity and results in very low supply air dew point temperatures. Those dew point temperatures are far lower than that required by air-water systems and will thus not be further discussed in this document.

Comparing air-water and all-air VAV systems

All-air variable air volume (VAV) systems have been the predominant air distribution system for commercial buildings for over forty years. These systems deliver air to the space at a constant supply air temperature (and humidity ratio) while throttling their supply air volume flow rate in accordance with the space cooling demand. Space ventilation requirements mandate a minimum outdoor air supply rate that limits the terminal units' airflow turndown. They may also be fitted with electric or hot water heating coils that allow them to reheat the air to prevent over cooling of the spaces they serve.

Figure 14 compares the use of a SCFT system to an all-air VAV system in a typical office building.

Figure 14 calculations are based on the following design conditions:

1. *Room air design conditions are 75°F and 50% RH which corresponds to a humidity ratio (W_{ROOM}) of 65 grains per pound of dry air.*
2. *Primary air is delivered at 55°F and 80% RH which corresponds to a humidity ratio (W_{PA}) of 51 grains per pound of dry air.*
3. *Interior and perimeter ventilation airflow rate based on an occupant density of 10 persons per 1,000 ft² floor area.*
4. *Interior and perimeter design latent airflow rates are based on latent cooling loads of 2 and 4 Btu/h-ft² respectively.*
5. *Interior and perimeter sensible cooling airflow rates are based on average space design loads of 12 and 36 Btu/h-ft², respectively.*

While the primary airflow requirements for space ventilation and latent cooling are the same for both systems, the supply airflow rate to provide the perimeter area design sensible cooling in the VAV system is nearly three times that required for space latent cooling and ventilation.

As the latent cooling performed by the air handling unit is proportional to its sensible cooling, a significant amount more moisture is removed at the VAV system air handling unit than would be necessary to maintain the desired space humidity level.

The sensible cooling performed by the coils within the DLSC terminals allow the ducted airflow rate to be reduced to exactly that which is required for space humidity control resulting not only in lower cooling transport costs, but also refrigerant plant energy savings reflected by the reduced latent cooling.

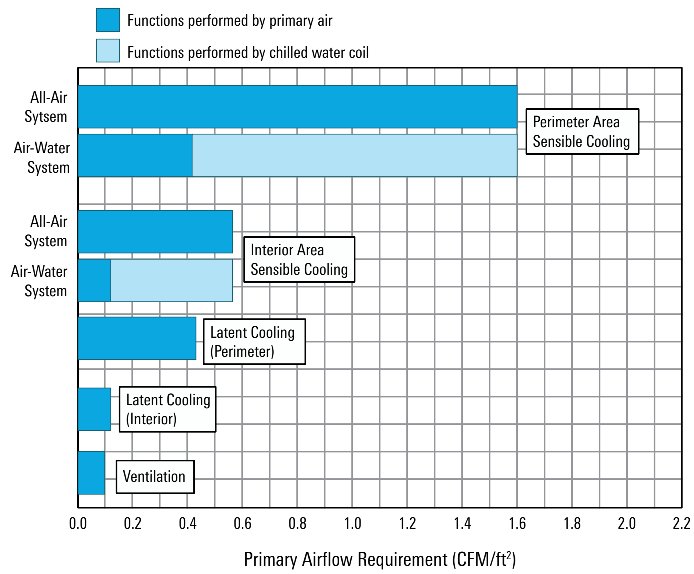
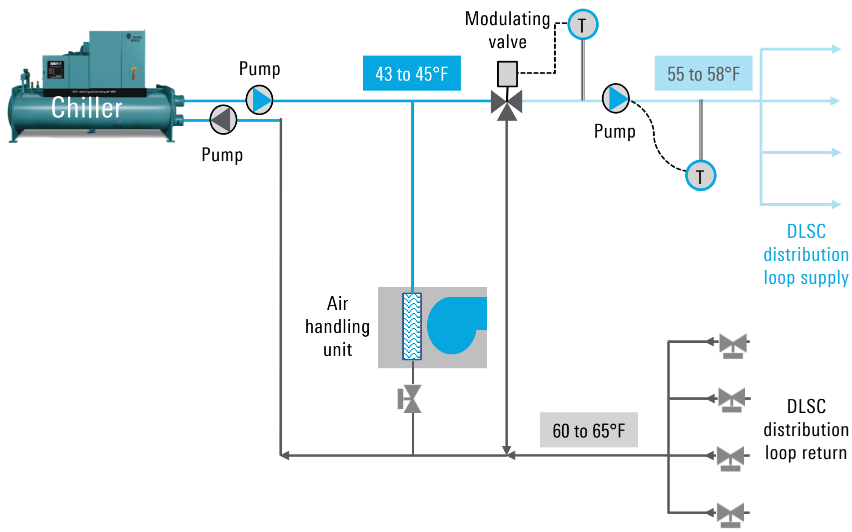


Figure 14: Comparing air-water to all-air VAV in an office building

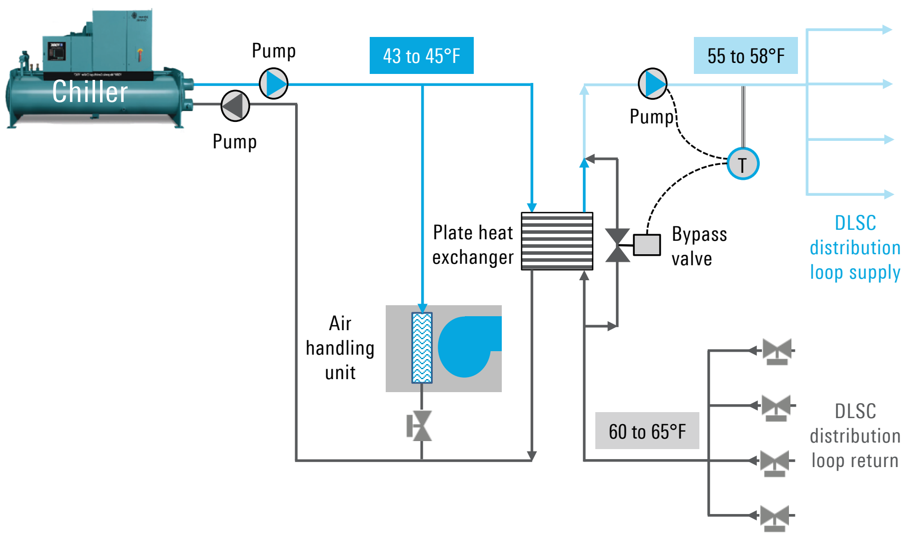
DLSC SYSTEM CHILLED WATER SOURCING

Sensible cooling fan terminals rely on chilled water delivered at a temperature that will not result in condensation on their cooling coil surfaces. This is typically 12 to 15°F warmer than that delivered to the cooling/dehumidifying coil in the AHU coil that is designed to condense moisture from its entering air. As such, some method of deriving appropriate temperature chilled water for each device must be devised. This can be achieved by either a shared chiller or by dedicating separate chillers for each function.

Figure 15 illustrates two applications where a single chiller is used to supply chilled water to both the AHU and the DLSC terminals' supply water loop. This may be accomplished with an open or closed loop secondary water circuit.



1. Open loop configuration



2. Closed loop configuration

Figure 15: DLSC application with single shared chiller

Figure 15a) illustrates the use of a three-way modulating valve to blend water returning from the DLSC loop with low temperature make-up chilled water to maintain the required delivery temperature to the DLSC terminals it serves. The supply water temperature of the secondary chilled water loop is monitored, and a modulating valve is positioned to allow an appropriate amount of low temperature water to pass into the DLSC circuit to offset heat removed by its return loop.

Figure 15b) represents a closed loop configuration where a plate heat exchanger is employed to extract heat from the DLSC return loop and transfer it to the primary chilled water return loop. Again, the secondary loop's supply water temperature is monitored and actuates a three-way valve that assigns an appropriate flow rate of secondary return water to bypass the heat exchanger. There the heat absorbed in the DLSC water circuit is transferred to the primary chilled water loop and subsequently returned to the shared chiller.

The latter option may be preferred in applications such as K-12 schools as it assures that no low temperature chilled water can possibly flow into the secondary chilled water loop. It may also be favored when applied in cold climates that require glycol additives in the primary chilled water loop. The use of the heat exchanger eliminates glycol entering the secondary water loop which would reduce the chilled water's heat transfer capacity.

In both cases, the chilled water flow rate through each DLSC's sensible cooling coil is varied by a zone chilled water valve in response to a signal from the zone temperature controller. Two-way valves which provide on-off or time modulated control of the zone chilled water flow rate are typically employed. Variable volume pumping systems adjust the total circuit flow rate according to the pressure loss detected.

Alternatively, the air handling unit and DLSC terminals may be served by dedicated chilled water sources. Figure 16 illustrates this application. Chiller 1 delivers moderate temperature (55 to 58°F) chilled water to the DLSC terminals while chiller 2 delivers low temperature chilled water to the air handling unit to support its cooling and dehumidification process. Chiller energy savings of 15 to 20% may be realized by using a dedicated chiller for the DLSC supply loop.

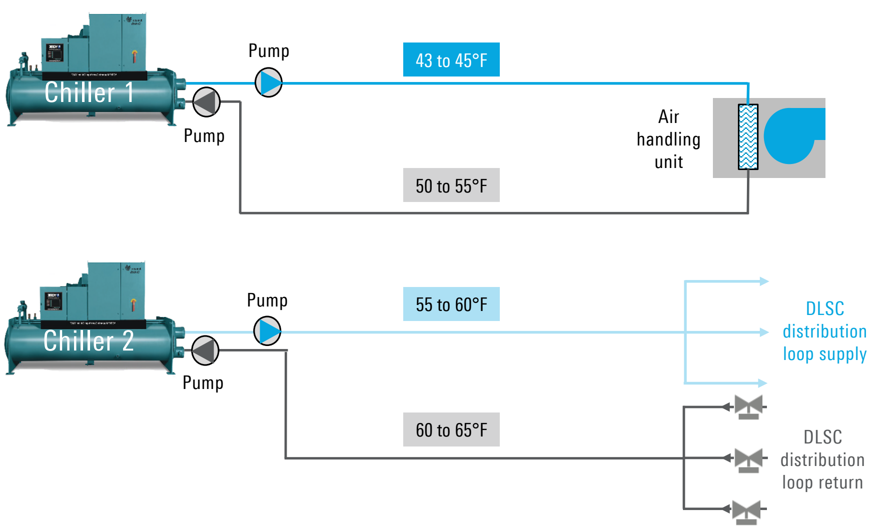


Figure 16: DLSC application with dedicated chilled water source

In this latter case, the air handling unit could have also employed direct expansion (DX) coils for cooling and dehumidification which would eliminate the need for the low temperature chiller (designated ad chiller number 1 in figure 16).

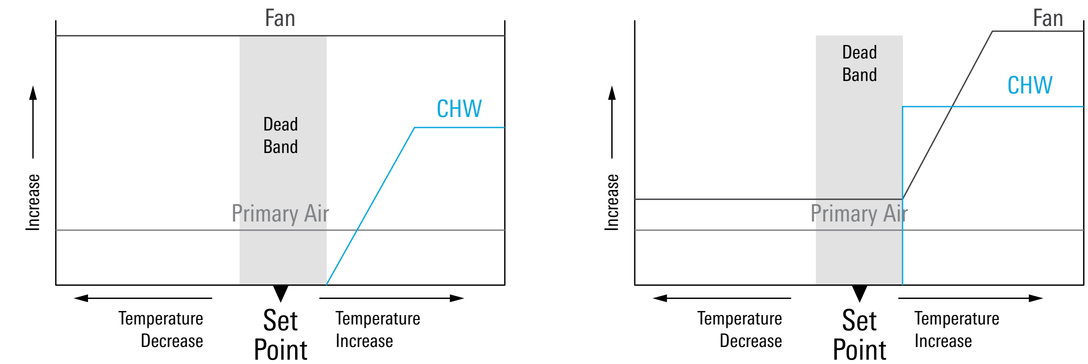
DLSC operational objectives should be tailored to the control of the ventilation and sensible/latent cooling of the zone they serve. Minimum ventilation rates for most occupancies are established by ASHRAE Standard 62.1-2019, although certain applications may be governed by other ventilation standards.

The unit's primary airflow rate must also be sufficient to remove space moisture gains and assure condensation does not form on the unit's sensible cooling coil. This involves maintaining the induced air dew point temperature at or below the zone's chilled water supply temperature. **Although DLSC terminals are provided with a protective drip tray beneath their cooling coil, they are never intended to be used in a condensing mode.**

The sensible cooling and heating (where applicable) provided by the DLSC terminals must be varied to maintain acceptable thermal comfort of the zone occupants across a wide range of space load conditions. In order to accomplish this, a zone temperature sensor must monitor the space heating/cooling demand and vary the terminal's supply airflow rate and/or temperature accordingly.

DLSC OPERATIONAL SEQUENCES

Figure 17 illustrates the typical occupied operating sequence for a DLSC terminal configured to provide zone cooling only. Note that the DLSC fan speed may be either constant or variable. In cases where the fan speed is maintained constant, the flow rate of chilled water passing through the terminal's induction side heat transfer coil is modulated in accordance with the space temperature offset from its setpoint. If the variable speed option is employed, a call for cooling will first result in the zone chilled water valve opening to its design flow rate. The DLSC fan speed will then be ramped up (from its minimum) accordingly by a demand for additional cooling. **Note that the minimum fan speed (table 3) may be greater than the primary airflow rate.**



Constant fan speed operation

1. AHU and DLSC fans remain on
2. Primary airflow rate maintained constant
3. Zone temperature increase above set point
   a. DLSC fan speed remains constant
   b. CHW valve modulates to maximum
4. Zone temperature drops below set point
   a. Zone chilled water valve remains closed
   b. DLSC fan speed remains constant

Variable fan speed operation

1. AHU and DLSC fans remain on
2. Primary airflow rate maintained constant
3. Zone temperature increase above set point
   a. Zone chilled water valve opens fully
   b. DLSC fan speed modulates to maximum
4. Zone temperature drops below set point
   a. Zone chilled water valve remains closed
   b. DLSC fan speed remains at minimum

Figure 17: Occupied operational sequences for DLSC (cooling only)

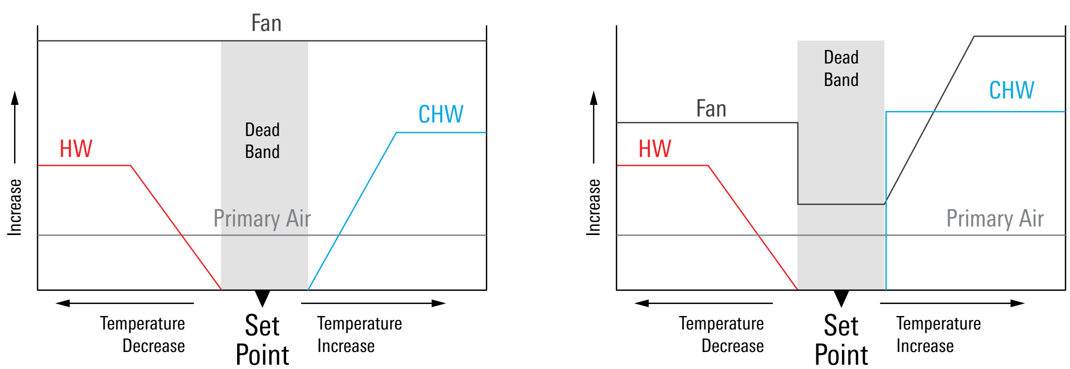
Figures 18 and 19 detail the operation of DLSC terminals with hot water and electric coils, respectively. Again, both constant and variable fan speed options are shown. DLSC electric heating coils are mounted on the unit's discharge and may be either staged (one or two stages) or time proportional (Titus Lynergy™) SCR controlled.

The cooling sequences for the constant speed options in figures 18 and 19 are identical to those shown in figure 17. **Note that in the variable fan speed cases, the fan operates at an increased air flow rate during heating** while the hot water flow rate or the electric heat is modulated according to the offset from the desired zone set point. This is done to assure that the leaving supply air temperature does not exceed the room temperature by more than 15°F, in accordance with ASHRAE recommendations for overhead forced air heating.

Table 6-4 of ASHRAE Standard 62.1-2019 Ventilation for Acceptable Indoor Air Quality assigns Zone Air Distribution Effectiveness (E_Z) factors to various room air distribution configurations. The amount of outside air that must be delivered to a zone is inversely proportional to the air distribution configuration's E_Z factor. The table stipulates that the overhead supply of air less than 15°F above the average room temperature qualifies for an E_Z factor of 1.0. Air supplied warmer than that is awarded an E_Z of only 0.8.

This means the zone ventilation (outdoor) air delivery under the latter circumstance (E_Z = 0.8) must be 25 percent higher than when the zone ventilation effectiveness factor is equal to 1.0. This can result in significant ventilation air conditioning energy use, especially in cold climate areas.

It is also recommended that warm supply air jets discharged along a wall or window achieve a vertical projection to 4.5 feet above the floor (approximately the height of the occupied zone for a stationary sitting occupant). Doing so requires significantly higher airflow rates when higher supply air to room temperature differentials are employed.



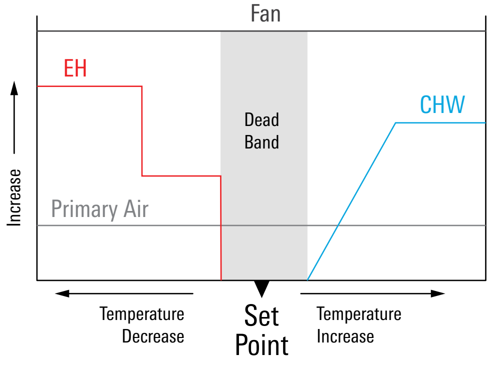
Constant fan speed operation

1. AHU and DLSC fans remain on
2. Primary airflow rate maintained constant
3. Zone temperature increase above set point
   a. DLSC fan speed remains constant
   b. CHW valve modulates to maximum
4. Zone temperature drops below set point
   a. DLSC fan speed remains constant
   b. CHW valve remains closed
   c. HW valve modulates open

Variable fan speed operation

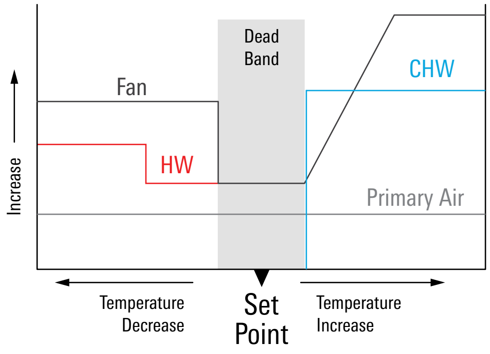
1. AHU and DLSC fans remain on
2. Primary airflow rate maintained constant
3. Zone temperature increase above set point
   a. DLSC fan speed increases
   b. CHW valve modulates to maximum
4. Zone temperature drops below set point
   a. CHW valve remains closed
   b. DLSC fan speed increases to heating max
   c. HW valve modulates open

Figure 18: Occupied control sequences for DLSC (cooling with hydronic heat) fan terminals



Constant fan speed operation

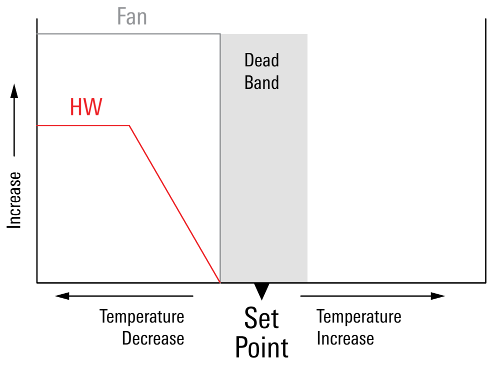
1. AHU and DLSC fans remain on
2. Primary airflow rate maintained constant
3. Zone temperature increase above set point
   a. DLSC fan speed remains constant
   b. CHW valve modulates to maximum
4. Zone temperature drops below set point
   a. DLSC fan speed remains constant
   b. CHW valve remains closed
   c. Electric heat energized



Variable fan speed operation

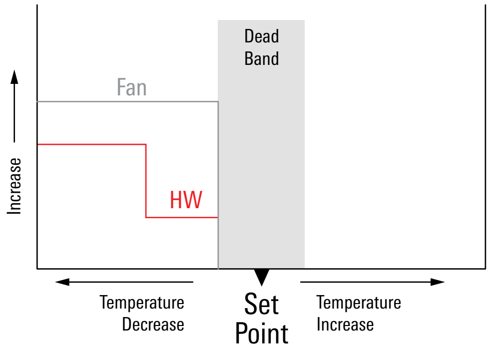
1. AHU and DLSC fans remain on
2. Primary airflow rate maintained constant
3. Zone temperature increase above set point
   a. CHW valve opens fully
   b. DLSC fan speed increases (to max.)
4. Zone temperature drops below set point
   a. CHW valve remains closed
   b. DLSC fan speed increases to heating max
   c. Electric heat energized

Figure 19: Occupied control sequences for DLSC (cooling with electric heat) fan terminals



Unoccupied mode, HW heat

1. 1. AHU fan remains off
2. Zone temperature increase above set point
   a. DLSC fan remains off
   b. CHW valve remains closed
3. . Zone temperature drops below set point
   a. DLSC fan energized to max. heat speed
   b. HW valve modulates open



Unoccupied mode, electric heat

1. 1. AHU fan remain off
2. Zone temperature increase above set point
   a. DLSC fan remains off
   b. CHW valve remains closed
3. Zone temperature drops below set point
   a. DLSC fan energized to max. heat speed
   b. Electric heat energized (and staged)

Figure 20: Unoccupied mode control sequences for DLSC fan terminals

Unoccupied set back operation and system restart

During unoccupied hours when the air handling unit is off, DLSC terminals operate only when heating is required. The chilled water valve remains closed while the fan and hot water or electric heating coil are sequenced as shown in figure 20 to maintain the desired night set back temperature.

It is important that the chilled water valve remain closed whenever the air handling unit is not running because there will be no zone humidity control and condensation could form on the unit's cooling coil. Upon the air handling unit being restarted, it is recommended that the primary air delivery be restored for a sufficient period to bring the space humidity level under control before recommencing the chilled water supply to the DLSC terminal unit coils.

Constant versus variable fan operation

High efficiency EC motors provide the DLSC system designer the option of varying the fan speed during cooling operation. This can result in significant fan energy reductions, especially in milder climates where heating requirements are minimal. Unlike conventional PSC motors whose operational efficiency drops markedly as their speed is reduced, EC motors maintain similar efficiencies throughout their speed range.

Figure 21 illustrates a typical power consumption curve for various percentages of the fan rated CFM. The fan rated CFM for the various unit sizes can be obtained from Table 3 (page 11).

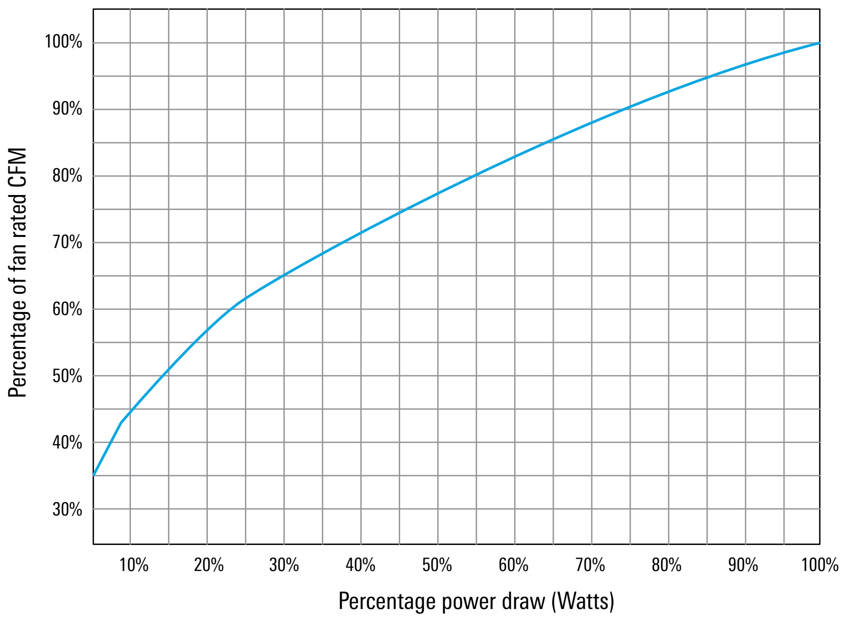


Figure 21: Power consumption of EC motors at reduced speeds

For example, a size 3 DLSC delivering 600 CFM (60% of its rated maximum airflow) during cooling operation would be estimated to do so at about 25% of its maximum rated power consumption! The same unit operating at 80% of its rated airflow would be expected to do so at 55% of its maximum (rated) power consumption.

Although the variable speed alternative is obviously preferred form an energy standpoint, there are other factors that should be considered when deciding on a fan speed operation and control method. Among these are acoustics and occupant thermal comfort. Table 5 lists the relative advantages and disadvantages of each scenario.

Effect	Constant fan speed operation	Variable fan speed operation
Energy	Higher fan energy costs but easier to control. Variable water flow rates reflect load diversity and result in lower pumping costs.	Fan energy reductions during off-peak cooling due to high efficiency EC motors. Higher pumping costs due to constant volume chilled water flow rates.
Acoustics	Higher noise levels coincide with higher fan speed, but constant fan speeds result in a consistent acoustical environment.	Reduced fan speeds during off peak cooling operation result in lower noise levels, but results in a less consistent acoustical environment.
Thermal comfort	Consistent air motion due to constant volume supply airflow rate. Higher cooling supply air temperatures less likely to result in dumping.	Variable volume supply airflow rate during cooling can result in insufficient throw and potential dumping.

Table 5: Comparison of fan speed operation options

Demand control ventilation with DLSC fan terminals

DLSC fan terminals can also support demand control ventilation (DCV) in conference and meeting areas where occupancy levels vary considerably. This involves reducing the design primary airflow rate during periods of reduced occupancy. When DCV is employed, it is sound practice to provide an override that prevents the primary airflow rate from dropping below that necessary for controlling the space dew point temperature.

Figure 22 below illustrates the control of a DLSC fan terminal configured for demand control ventilation. Zone temperature and humidity sensors provide input regarding the space dry-bulb temperature and relative humidity. The zone controller uses this information to control the zone chilled and hot water (where applicable) valve flow rates. It also calculates the space dew point temperature. A zone CO_2 sensor monitors the space occupancy. The controller then adjusts the primary airflow rate accordingly while continuing to monitor the space dew point. Upon a rise in the space dew point temperature that exceeds the chilled water supply temperature, the controller restores the primary airflow rate to its original design value until the design space dew point temperature is restored. The fan speed can be either constant or variable as shown in figures 18 through 20.

Alternatively, the zone chilled water supply temperature could be raised in accordance with a sensed space dew point temperature rise. This method will be further addressed in the Condensation and Risk Prevention section that follows.

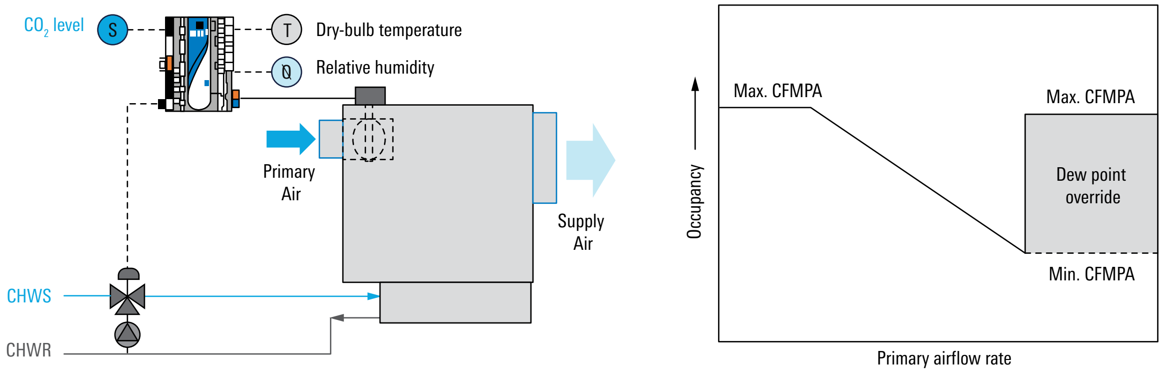


Figure 22: Demand control ventilation with DLSC fan terminals

CONDENSATION RISK AND PREVENTION

When the DLSC unit's primary airflow rate and moisture content is sufficient to keep the dew point temperature of induced air passing over the DLSC unit cooling coil at or below its chilled water supply temperature, condensation should never be an issue. In fact, the drying effect of the air velocity passing over the coil surfaces should allow for dew point rises up to 3°F above the chilled water temperature to occur before condensation will form. In any case, DLSC terminals are provided with a drip tray to catch any condensation that might fall from their heat transfer coil surfaces. These trays are not intended to be piped and again, **DLSC terminals should never be intended to operate in a condensing mode.**

In applications where there is a risk of condensation, there are several prevention strategies that may be considered.

1. As previously mentioned, conference and meeting areas where demand control ventilation is employed should consider having a room dew point override that resets the primary airflow rate upward in the event the dew point rises above the DLSC's chilled water supply temperature. Upon restoration of the space dew point, control is relinquished back to the space temperature and CO_2 or motion sensors. Figure 23 illustrates such a control sequence.

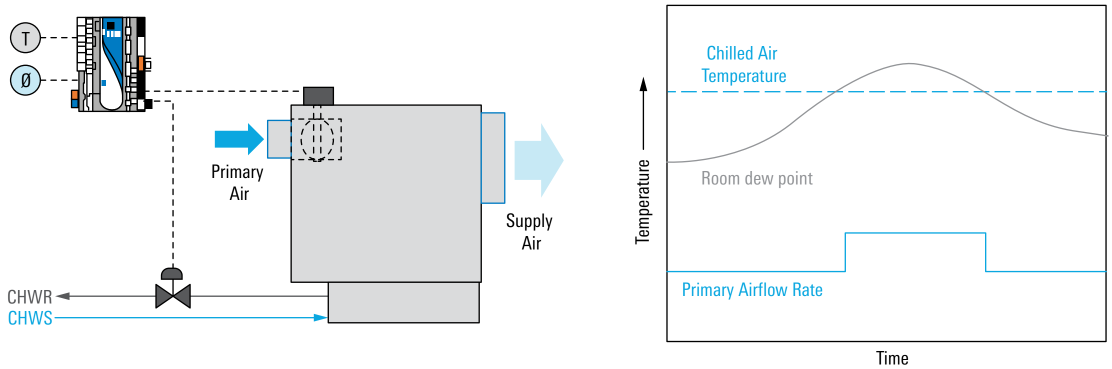


Figure 23: Primary airflow purge

2. For classroom applications and other instances involving operable windows, the space dew point temperature can be monitored and the coil's chilled water supply temperature reset upward (see figure 24) to maintain a constant differential between the dew point and chilled water supply temperatures. This can also be done on a multiple zone basis by polling dew point temperatures in a few key spaces and resetting the chilled water supply temperature according to the highest value polled.

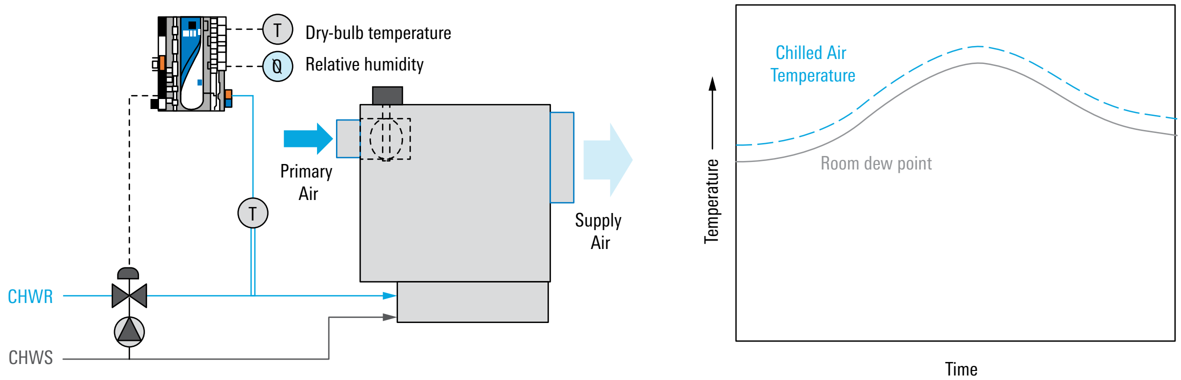


Figure 24: Zone chilled water temperature modulation

DLSC selection and sizing is typically driven by the zone's sensible cooling requirement. The terminal unit(s) fan speed and sensible cooling coil capacity combine to augment the primary air's contribution to deliver the zone's design sensible cooling.

Tables 7 and 8 detail the standard and high-capacity DLSC units' induction coil sensible cooling capacities and water side pressure losses at various chilled water flow rates. These tables are based on an 18°F temperature differential between the induced air and chilled water supply temperature. The data in the tables can be revised for other temperature differentials by multiplying the table values by the applicable correction factor from table 6.

ΔT between entering chilled water and induced air (°F)							
Correction factor	12	14	16	18	20	22	24
	0.67	0.78	0.89	1.00	1.11	1.22	1.33

Table 6: Induction coil cooling ΔT correction factors

The temperature of the induced air leaving the induction cooling (T_{LCC}) can be estimated by equation 6.1:

$$T_{LCC} = T_{IA} - \frac{1000 \times MBH_{CORR}}{60 \times \rho_{IA} \times C_P \times CFM_{IA}} \qquad \text{equation 6.1}$$

where,

T_{IA} = temperature of the induced air entering the coil, °F

MBH = MBH of sensible heat removed by the coil

ρ_{IA} = density of the induced air, lbm/ft^3

C_P = specific heat constant for air (0.24 Btu/lbm -°F)

CFM_{IA} = Induced airflow rate, ft^3/min

The chilled water temperature (T_{LCW}) leaving the cooling coil can be estimated by equation 6.2:

$$T_{LCW} = T_{ECW} + \frac{1000 \times MBH_{CORR}}{500 \times GPM_{CW}} \qquad \text{equation 6.2}$$

where,

T_{ECW} = temperature of the induced air entering the coil, °F

GPM_{CHW} = chilled water flow rate, GPM

Finally, the DLSC supply air temperature (T_{SUPPLY}) can be determined using equation 6.3:

$$T_{SUPPLY} = \frac{(CFM_{PA} \times T_{PA}) + (CFM_{IA} \times T_{LCC})}{CFM_{FAN}} \qquad \text{equation 6.3}$$

where,

T_{SUPPLY} = Temperature of the supply air leaving the DLSC terminal, °F

CFM_{PA} = Primary airflow rate, ft^3/min

T_{PA} = Temperature of the primary air, °F

CFM_{IA} = induced airflow rate, ft^3/min

T_{LCC} = temperature of the induced air leaving the sensible cooling coil, °F

CFM_{FAN} = fan airflow rate (CFM_{PA} + CFM_{IA}), ft^3/min

DLSC coil water flow rates should be sufficient to assure turbulent water flow but not result in excessive coil water pressure losses (over 10 feet of water). The range of flow rates in the tables reflect these lower and upper limits.

Unit size	Cooling rows/circuits	Induced air SCFM	MBH cooling at various chilled water flow rates (GPM)									
			1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
1, 2	2 rows 2 circuits	150	2.2	2.2	2.2	2.3	2.3	2.3	2.3	2.3	2.3	2.3
		275	2.9	3.1	3.1	3.2	3.3	3.3	3.3	3.4	3.4	3.4
		400	3.5	3.8	3.9	4.1	4.2	4.2	4.3	4.3	4.4	4.4
		525	4.0	4.4	4.6	4.8	5.0	5.0	5.1	5.2	5.2	5.3
		650	4.5	4.9	5.2	5.4	5.6	5.7	5.8	5.9	6.0	6.0
		ΔP water (ft.)	0.7	1.2	1.9	2.7	3.7	4.8	6.0	7.3	8.8	**10.4**
	4 rows 3 circuits	150			2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7
		275			4.2	4.3	4.3	4.3	4.3	4.4	4.4	4.4
		400			5.5	5.6	5.7	5.8	5.9	5.9	5.9	6.0
		525			6.6	6.8	6.9	7.1	7.2	7.3	7.3	7.4
		650			7.5	7.8	8.0	8.2	8.3	8.4	8.5	8.6
		ΔP water (ft.)			1.3	1.8	2.5	3.2	4.0	5.0	6.0	7.1
	6 rows 4 circuits	150				2.7	2.7	2.6	2.7	2.6	2.6	2.6
		275				4.6	4.6	4.6	4.7	4.7	4.7	4.7
		400				6.3	6.3	6.4	6.5	6.5	6.5	6.6
		525				7.7	7.9	8.0	8.1	8.2	8.2	8.3
		650				8.9	9.2	9.4	9.5	9.6	9.7	9.8
		ΔP water (ft.)				1.2	1.7	2.2	2.7	3.3	4.0	4.8
3	2 rows 2 circuits	325	3.9	4.2	4.3	4.5	4.5	4.6	4.7	4.7	4.7	4.8
		475	4.9	5.4	5.6	5.8	6.0	6.1	6.2	6.3	6.4	6.4
		625	5.7	6.3	6.7	7.0	7.2	7.4	7.6	7.7	7.8	7.9
		775	6.4	7.1	7.6	8.0	8.3	8.5	8.7	8.9	9.0	9.1
		925	6.9	7.7	8.3	8.8	9.2	9.4	9.7	9.9	10.0	10.2
		ΔP water (ft.)	0.8	1.4	2.2	3.2	4.3	5.6	7.0	8.6	**10.3**	**12.2**
	4 rows 3 circuits	325			5.0	5.1	5.1	5.1	5.2	5.2	5.2	5.2
		475			6.4	6.6	6.7	6.8	6.9	6.9	7.0	7.0
		625			7.6	7.9	8.1	8.3	8.4	8.5	8.6	8.7
		775			8.6	9.0	9.3	9.5	9.7	9.9	10.0	10.1
		925			9.4	9.9	10.3	10.6	10.9	11.1	11.3	11.4
		ΔP water (ft.)			1.5	2.1	2.9	3.7	4.7	5.7	6.9	8.2
	6 rows 4 circuits	325				5.5	5.5	5.5	5.5	5.5	5.5	5.5
		475				7.3	7.5	7.5	7.6	7.7	7.7	7.7
		625				9.0	9.2	9.4	9.5	9.6	9.7	9.8
		775				10.4	10.7	11.0	11.2	11.3	11.5	11.6
		925				11.5	12.0	12.3	12.6	12.8	13.0	13.2
		ΔP water (ft.)				1.4	1.9	2.5	3.1	3.8	4.6	5.5
5	2 rows 2 circuits	475	3.2	3.3	3.4	3.5	3.6	3.6	3.6	3.7	3.7	3.7
		725	3.8	4.1	4.3	4.4	4.5	4.6	4.7	4.7	4.8	4.8
		975	4.4	4.8	5.1	5.2	5.4	5.5	5.6	5.7	5.7	5.8
		1,225	4.9	5.4	5.7	6.0	6.2	6.3	6.4	6.5	6.6	6.7
		1,475	5.3	5.9	6.3	6.6	6.8	7.0	7.2	7.3	7.4	7.5
		ΔP water (ft.)	1.3	2.3	3.6	5.2	7.0	9.0	**11.4**	**13.9**	**16.8**	**19.8**
	4 rows 3 circuits	475				4.8	5.2	5.4	5.6	5.6	5.8	5.9
		725				5.7	6.3	6.7	7.0	7.2	7.4	7.5
		975				6.5	7.3	7.8	8.2	8.5	8.8	9.0
		1,225				7.1	8.1	8.7	9.3	9.6	10.0	10.2
		1,475				7.6	8.7	9.5	10.1	10.6	11.0	11.3
		ΔP water (ft.)				3.3	4.5	5.8	7.3	9.0	**10.8**	**12.8**
	6 rows 4 circuits	475				8.2	8.2	8.2	8.2	8.2	8.2	8.2
		725				10.9	11.2	11.4	11.5	11.6	11.7	11.8
		975				13.2	13.7	14.1	14.4	14.6	14.8	15.0
		1,225				15.1	15.8	16.4	16.9	17.2	17.5	17.8
		1,475				16.5	17.5	18.3	18.9	19.4	19.9	20.2
		ΔP water (ft.)				2.2	2.9	3.8	4.8	5.9	7.1	8.4

NOTE : Table is based on an 18°F temperature difference between the entering air and chilled water **Bold type** indicates water side pressure drop > 10 ft.

Possible laminar flow region

Table 7: Induction coil sensible cooling capacities of standard DLSC terminals

Unit size	Cooling rows/circuits	Induced air SCFM	MBH cooling at various chilled water flow rates (GPM)									
			1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0
A, B	2 rows 2 circuits	150	3.0	3.3	3.4	3.4	3.5	3.5	3.4	3.5	3.5	3.8
		275	4.4	4.7	4.9	5.2	5.3	5.3	5.5	5.5	5.6	6.1
		400	5.3	5.8	6.1	6.4	6.6	6.7	6.9	7.0	7.1	7.8
		525	5.8	6.7	7.1	7.0	7.3	7.6	7.5	7.8	8.0	9.0
		650	5.9	7.3	7.7	7.1	7.6	8.0	7.5	8.0	8.3	9.7
		ΔP water (ft.)	1.1	1.9	2.9	4.2	5.6	7.3	9.1	**11.2**	**13.5**	**15.9**
	4 rows 3 circuits	150			3.9	4.4	4.1	4.1	4.1	4.1	4.0	4.0
		275			6.5	8.0	6.8	6.8	6.9	6.9	7.0	7.0
		400			8.4	11.5	9.0	9.3	9.3	9.4	9.5	9.5
		525			9.6	15.0	10.9	11.4	11.4	11.5	11.5	11.5
		650			10.1	18.4	12.3	13.2	13.2	13.1	13.1	13.0
		ΔP water (ft.)			0.9	1.2	1.7	2.2	2.7	3.4	4.0	4.8
	6 rows 4 circuits	150				4.3	4.3	4.3	4.3	4.3	4.3	4.3
		275				7.5	7.6	7.6	7.7	7.7	7.7	7.7
		400				10.0	10.3	10.4	10.5	10.6	10.7	10.8
		525				12.0	12.4	12.8	13.0	13.2	13.4	13.5
		650				13.4	14.1	14.6	15.0	15.4	15.6	15.8
		ΔP water (ft.)				1.8	2.4	3.1	3.9	4.8	5.8	6.9
C	2 rows 2 circuits	325	3.7	3.9	4.1	4.2	4.2	4.3	4.3	4.4	4.4	4.4
		475	4.6	4.9	5.2	5.3	5.5	5.6	5.6	5.7	5.8	5.8
		625	5.3	5.8	6.1	6.4	6.6	6.7	6.8	6.9	7.0	7.1
		775	5.9	6.5	7.0	7.3	7.5	7.7	7.9	8.0	8.1	8.2
		925	6.4	7.2	7.7	8.1	8.4	8.6	8.8	8.9	9.1	9.2
		ΔP water (ft.)	1.3	2.2	3.4	4.8	6.5	8.5	**10.6**	**13.1**	**15.7**	**18.6**
	4 rows 3 circuits	325			5.2	5.3	5.3	5.4	5.4	5.4	5.4	5.4
		475			6.9	7.1	7.2	7.3	7.4	7.4	7.5	7.5
		625			8.3	8.7	8.9	9.0	9.2	9.3	9.4	9.4
		775			9.6	10.0	10.3	10.6	10.8	10.9	11.1	11.2
		925			10.6	11.2	11.6	11.9	12.2	12.4	12.6	12.7
		ΔP water (ft.)			1.0	1.4	1.9	2.5	3.1	3.8	4.6	5.5
	6 rows 4 circuits	325				5.6	5.6	5.6	5.6	5.6	5.6	5.6
		475				7.8	7.9	7.9	8.0	8.0	8.1	8.1
		625				9.7	9.9	10.1	10.2	10.2	10.3	10.4
		775				11.4	11.8	12.0	12.2	12.3	12.4	12.5
		925				12.9	13.3	13.7	13.9	14.1	14.3	14.4
		ΔP water (ft.)				2.1	2.8	3.6	4.5	5.6	6.7	7.9
E	2 rows 2 circuits	475	5.0	5.4	5.6	5.8	5.9	6.0	6.1	6.2	6.3	6.3
		725	6.1	6.8	7.2	7.5	7.8	8.0	8.1	8.2	8.3	8.4
		975	7.1	7.9	8.5	9.0	9.3	9.6	9.9	10.0	10.2	10.3
		1,225	7.8	8.9	9.7	10.2	10.7	11.1	11.4	11.6	11.8	12.0
		1,475	8.4	9.7	10.6	11.3	11.8	12.3	12.7	13.0	13.2	13.5
		ΔP water (ft.)	1.8	3.1	4.9	6.9	9.3	**12.1**	**15.2**	**18.7**	**22.5**	**26.7**
	4 rows 3 circuits	475			7.4	7.5	7.6	7.7	7.7	7.8	7.8	7.8
		725			9.7	10.1	10.4	10.6	10.8	10.9	11.1	11.1
		975			11.7	12.3	12.8	13.2	13.5	13.7	13.9	14.1
		1,225			13.2	14.2	14.8	15.4	15.8	16.1	16.4	16.7
		1,475			14.4	15.6	16.5	17.2	17.7	18.2	18.6	18.9
		ΔP water (ft.)			1.4	2.0	2.6	3.4	4.3	5.3	6.4	7.6
	6 rows 4 circuits	475				8.1	8.1	8.0	8.0	8.0	7.9	7.9
		725				11.3	11.5	11.6	11.5	11.5	11.5	11.4
		975				14.1	14.4	14.5	14.5	14.5	14.5	14.4
		1,225				16.2	16.7	16.9	17.0	16.9	16.9	16.8
		1,475				17.8	18.4	18.7	18.8	18.8	18.8	18.7
		ΔP water (ft.)				2.9	3.9	5.0	5.3	7.7	9.3	**11.1**

NOTE : Table is based on an 18°F temperature difference between the entering air and chilled water **Bold type** indicates water side pressure drop > 10 ft.

Possible laminar flow region

Table 8: Induction coil sensible cooling capacities of high capacity DLSC terminals

HOT WATER HEATING WITH DLSC TERMINALS

When DLSC terminals are intended to serve perimeter areas, they can be fitted with discharge hydronic or electric heating coils. Tables 9 and 10 detail the heating capacities of 1 and 2-row hydronic coils respectively when added to the discharge of various DLSC models. Both tables are based on a 50°F temperature differential between the hot water and air mixture entering the coil. Capacities for other differentials can be estimated by applying the correction factors in table 11.

Unit size	Heating rows	Fan CFM	MBH heating at various water flow rates (GPM)									
			0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0
1, 2 A, B	1	200	2.6	3.4	3.7	3.8	3.9	3.9	3.9	4.0	4.1	4.2
		300	2.9	4.0	4.4	4.6	4.7	4.8	4.8	4.9	5.0	5.1
		400	3.1	4.5	5.0	5.3	5.4	5.5	5.6	5.7	5.8	6.0
		500	3.3	4.9	5.5	5.9	6.0	6.2	6.3	6.4	6.4	6.6
		600	3.5	5.2	5.9	6.3	6.5	6.7	6.8	6.9	7.0	7.2
		700	3.6	5.4	6.2	6.6	6.9	7.0	7.2	7.3	7.4	7.6
		ΔPHW (ft.)	0.03	0.1	0.2	0.3	0.5	0.7	1.0	1.2	1.5	1.9
3, C	1	350	3.4	4.7	5.3	5.5	5.7	5.7	5.8	5.9	5.9	5.9
		500	3.7	5.3	6.0	6.3	6.6	6.7	6.8	6.9	7.0	7.0
		650	3.9	5.8	6.7	7.1	7.3	7.5	7.7	7.8	7.9	7.9
		800	4.1	6.2	7.2	7.6	8.0	8.2	8.4	8.5	8.6	8.7
		950	4.2	6.5	7.6	8.1	8.5	8.7	9.0	9.1	9.2	9.3
		1,100	4.2	6.8	7.9	8.4	8.9	9.1	9.4	9.5	9.7	9.7
		ΔPHW (ft.)	0.04	0.1	0.2	0.4	0.6	0.9	1.1	1.5	1.8	2.5
5, E	1	500	5.0	7.1	8.0	8.4	8.7	8.9	9.0	9.1	9.2	9.3
		750	5.5	8.2	9.5	10.1	10.4	10.7	10.9	11.1	11.2	11.3
		1,000	5.9	9.1	10.6	11.4	11.9	12.3	12.5	12.7	12.9	13.0
		1,250	6.1	9.7	11.4	12.4	13.0	13.4	13.8	14.0	14.2	14.4
		1,500	6.3	10.1	12.0	13.1	13.7	14.2	14.7	14.9	15.2	15.4
		1,750	6.3	10.3	12.2	13.4	14.2	14.7	15.2	15.6	15.8	16.0
		ΔPHW (ft.)	0.1	0.2	0.4	0.6	1.0	1.4	1.8	2.3	2.9	3.6

NOTE : Table above based on a 50°F temperature differential between entering hot water and air mixture entering coil

Table 9: 1-row discharge hydronic heating coil performance (all sizes)

It is recommended that the differential between the warm supply air and that of the room be limited to 15°F to minimize thermal stratification and maximize zone ventilation effectiveness. While higher differentials might be required in extreme cases, they should never be allowed to exceed 25°F. Equation 6.4 may be used to calculate the minimum fan airflow rate required to limit the supply to room temperature differential for a given zone heating requirement (MBH_{ZONE}):

$$SCFM_{FAN} = \frac{MBH_{ZONE} \times 1000}{1.1 \times Max.\ \Delta T_{SR}} \qquad \text{equation 6.4}$$

Note that the fan airflow is presented in SCFM which indicates it is based on standard density air. When designing for other altitudes, SCFM values in this equation as well as equation 4.1 should be divided by the correction factor (C_{ALT}) calculated by equation 6.5:

$$C_{ALT} = [1 - (ALT \times 6.874 \times 10^{-6})]^{5.2599} \qquad \text{equation 6.5}$$

Unit size	Heating rows	Fan CFM	MBH heating at various water flow rates (GPM)									
			0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0
1, 2 A, B	2	200	4.2	5.6	6.0	6.3	6.4	6.5	6.6	6.6	6.6	6.7
		300	4.8	6.7	7.5	7.9	8.2	8.3	8.4	8.5	8.6	8.7
		400	5.3	7.7	8.8	9.3	9.7	9.9	10.1	10.2	10.3	10.4
		500	5.6	8.5	9.8	10.5	10.9	11.3	11.5	11.7	11.8	11.9
		600	5.9	9.1	10.7	11.5	12.0	12.4	12.6	12.9	13.0	13.2
		700	6.0	9.5	11.3	12.2	12.8	13.3	13.6	13.8	14.0	14.2
		ΔPHW (ft.)	0.07	0.2	0.4	0.6	0.9	1.3	1.8	2.2	2.8	3.4
3, C	2	350	5.5	7.9	8.9	9.3	9.6	9.9	10.0	10.1	10.2	10.2
		500	6.0	9.0	10.5	11.1	11.6	11.9	12.1	12.4	12.5	12.6
		650	6.4	10.0	11.8	12.7	13.3	13.7	14.0	14.3	14.5	14.6
		800	6.6	10.7	12.8	13.9	14.7	15.2	15.6	15.9	16.1	16.3
		950	6.8	11.2	13.6	14.9	15.7	16.4	16.8	17.2	17.5	17.7
		1,100	6.9	11.6	14.2	15.6	16.5	17.2	17.8	18.1	18.5	18.8
		ΔPHW (ft.)	0.1	0.3	0.4	0.7	1.1	1.5	2.0	2.6	3.3	4.0
5, E	2	500	7.6	11.2	12.8	13.7	14.2	14.6	14.8	15.0	15.2	15.3
		750	8.2	13.1	15.5	16.9	17.7	18.3	18.8	19.2	19.4	19.7
		1,000	8.7	14.4	17.6	19.4	20.6	21.4	22.1	22.6	23.0	23.3
		1,250	9.0	15.4	19.1	21.3	22.8	23.8	24.7	25.3	25.8	26.2
		1,500	9.1	15.8	20.1	22.6	24.3	25.5	26.5	27.3	27.9	28.4
		1,750	9.1	15.8	20.4	23.2	25.1	26.6	27.7	28.5	29.3	29.9
		ΔPHW (ft.)	0.1	0.5	0.8	1.2	1.9	2.6	3.5	4.5	5.6	6.8

NOTE : Table above based on a 50°F temperature differential between entering hot water and air mixture entering coil.

Table 10: 2-row discharge hydronic heating coil performance (all sizes)

ΔT between entering hot water and air entering coil (°F)							
Correction factor	40	50	60	70	80	90	100
	0.80	1.00	1.20	1.40	1.60	1.80	2.00

Table 11: Hydronic heating coil ΔT correction factors

In order to calculate the air and hot water temperatures leaving the discharge heating coil, the total amount of heat it is required to supply must first be determined. This can be calculated using equation 6.6 in which the primary air-cooling effect (MBH_{PA}) is added to the zone heating requirement (MBH_{ZONE}), then multiplied by the coil heating correction factor from table 11:

$$MBH_{TOTAL} = (MBH_{ZONE} + MBH_{PA}) \times CF_{TABLE11} \quad \text{equation 6.6}$$

The supply air and hot water temperature leaving the discharge heating coil can then be calculated by equations 6.7 and 6.8:

$$T_{LHW} = T_{EHW} - \frac{1000 \times MBH_{TOTAL}}{500 \times GPM_{HW}} \quad \text{equation 6.7}$$

$$T_{SUPPLY} = T_{MA} + \frac{1000 \times MBH}{C_{ALT} \times 1.1 \times CFM_{FAN}} \quad \text{equation 6.8}$$

Induction coil heaing

The induction coil on DLSC terminals can also be fitted with an additional row of copper tubes that are dedicated to heating. Table 12 details the heating capacity achieved by adding this extra row. The table is again based on a 50°F ΔT between their hot water supply and the induced air entering the coil. Table 11 can again be used to correct for other temperature differentials.

Unit size	Induced air CFM	MBH heat at various water flow rates (GPM)					
		0.5	1.0	1.5	2.0	2.5	3.0
1, 2	100	3.7	4.0	4.1	4.1	4.3	8.0
	200	4.4	5.1	5.4	5.5	5.6	5.5
	300	5.0	6.0	6.5	6.7	6.9	6.9
	400	5.5	6.9	7.4	7.7	8.0	8.1
	500	6.0	7.6	8.3	8.6	8.9	9.1
	600	6.4	8.1	8.9	9.4	9.7	10.0
	ΔP water (ft.)	0.3	1.0	2.3	3.9	8.0	8.6
3	150	4.6	5.2	5.5	5.7	5.7	5.7
	300	5.4	6.5	7.0	7.3	7.4	7.5
	450	6.1	7.7	8.4	8.8	9.0	9.2
	600	6.7	8.6	9.5	10.0	10.4	10.6
	750	7.2	9.4	10.5	11.1	11.6	11.8
	900	7.5	10.1	11.3	12.0	12.6	12.9
	ΔP water (ft.)	0.3	1.2	2.6	4.6	7.1	**10.0**
5	250	6.8	8.3	8.8	9.2	9.2	9.4
	500	7.7	9.9	10.9	11.5	11.9	12.2
	750	8.4	11.4	12.8	13.7	14.2	14.7
	1,000	9.0	12.6	14.4	15.5	16.3	16.8
	1,250	9.4	13.6	15.8	17.1	18.0	18.7
	1,500	9.7	14.3	16.8	18.4	19.4	20.2
	ΔP water (ft.)	0.5	2.0	4.3	7.5	**11.5**	**16.3**

Unit size	Induced air CFM	MBH heat at various water flow rates (GPM)					
		0.5	1.0	1.5	2.0	2.5	3.0
1, 2	100	4.2	4.6	4.7	4.7	4.8	4.8
	200	5.1	5.9	6.3	6.4	6.6	6.6
	300	5.9	7.1	7.7	7.9	8.2	8.3
	400	6.6	8.2	9.0	9.3	9.6	9.8
	500	7.1	9.1	10.1	10.5	10.9	11.1
	600	7.6	9.9	11.0	11.6	12.1	12.3
	ΔP water (ft.)	0.4	1.6	3.4	5.0	9.1	**13.0**
3	150	5.3	6.1	6.3	6.3	6.4	6.4
	300	6.3	7.7	8.2	8.5	8.7	8.8
	450	7.1	9.1	10.0	10.5	10.8	11.0
	600	7.8	10.3	11.5	12.2	12.6	12.9
	750	8.4	11.4	12.8	13.7	14.2	14.6
	900	8.9	12.2	13.9	14.9	15.6	16.1
	ΔP water (ft.)	0.5	1.8	4.0	7.0	**10.7**	**15.2**
5	250	7.2	8.7	9.3	9.6	9.8	9.9
	500	8.2	10.8	11.9	12.7	13.1	13.4
	750	9.0	12.6	14.3	15.3	16.0	16.5
	1,000	9.7	14.0	16.3	17.6	18.6	19.2
	1,250	10.2	15.3	17.9	19.6	20.7	21.5
	1,500	10.6	16.2	19.3	21.2	22.5	23.5
	ΔP water (ft.)	0.7	2.7	5.8	**19.1**	**15.4**	**21.3**

NOTE : Tables are based on a 50°F temperature differential between the hot water supply and the entering induced air **Bold type** indicates water side pressure drop > 10 ft.

Table 12: Heating capacities of DLSC induction coils

The temperature of the heated air leaving the induced air coil (T_{LIHC}) and the subsequent DLSC heating discharge temperature (T_{SUPPLY}) can be estimated by equations 6.9 and 6.10, where MBH_{CORR} is the MBH value from table 13 corrected for the actual entering air to entering water temperature differential multiplier from table 11

$$T_{LIHC} = T_{IA} + \frac{1000 \times MBH_{CORR}}{60 \times \rho_{IA} \times C_P \times CFM_{IA}} \quad \text{equation 6.9}$$

$$T_{SUPPLY} = \frac{(CFM_{PA} \times T_{PA}) + (CFM_{IA} \times T_{LIHC})}{CFM_{FAN}} \quad \text{equation 6.10}$$

The hot water temperature (T_{LHW}) leaving the heating coil can again be estimated using equation 6.7.

Anti-freeze additives

It is often necessary to add glycol to the chilled and/or hot water circuits serving the DLSC terminals. Doing so reduces the heat transfer capacity of the liquid mixture.

Table 13 provides correction factors for varying concentrations of ethylene and propylene glycol/water mixtures. When selecting coils served by such a mixture, the required coil capacity should be divided by the appropriate correction factor before accessing the coil performance table. For example, a coil required to provide 10 MBH of heat that is served by a 30% ethylene glycol solution would have to be selected to provide 11.23 MBH (10/0.89) due to the reduction in heat transfer capacity associated with the 30% glycol solution.

% Concentration (by volume)	Ethylene glycol			Propylene glycol		
	Freeze point, °F	F_{60}	F_{120}	Freeze point, °F	F_{60}	F_{120}
10	26	0.95	0.96	25	0.97	0.99
15	21	0.93	0.94	22	0.96	0.98
20	16	0.91	0.93	18	0.95	0.97
25	10	0.89	0.91	14	0.93	0.95
30	4	0.87	0.89	8	0.92	0.94
35	-4	0.90	0.90	2	0.90	0.90
40	-12	0.83	0.85	-6	0.88	0.91
45	-23	0.90	0.90	-16	0.90	0.90
50	-35	0.78	0.81	-29	0.84	0.87

NOTE: F60 and F120 are heat transfer correction factors for water at 60 and 120°F, respectively

Table 13: Glycol solution correction factors

Example 6.1:

A DLSC terminal with discharge hot water heat is tasked with providing 10 MBH of heat to the space it serves. The project location is Salt Lake City, Utah (elevation 4,226 feet). The mixed air temperature entering the coil is estimated to be 68°F and it is desired to limit the DLSC discharge temperature to 85°F.

What is the minimum heating fan CFM that will accomplish this?

Solution:

Equation 6.7 can be used to calculate the minimum SCFM as follows: The SCFM value must now be corrected for altitude using equation 6.5. The altitude correction for Salt Lake City is:

$$C_{ALT} = [1 - (4{,}226 \times 6.874 \times 10^{-6})]^{5.2599} = 0.855$$

$$CFM_{FAN} = \frac{MBH_{ZONE} \times 1000}{C_{ALT} \times 1.1 \times \text{Max. } \Delta T_{SR}} = \frac{10.0 \times 1000}{0.855 \times 1.1 \times 15} = 708$$

The corrected fan CFM is calculated as the SCFM / C_{ALT} or 708 CFM

Table 14 presents the maximum MBH values at various fan airflow rates (CFM) that limit supply to room temperature differentials to 15, 20 and 25°F based on a discharge coil entering air temperature of 68°F.

Maximum supply to room air ΔT (°F)	Maximum discharge hot water coil MBH @ various fan airflow rates (CFM)													
	200	300	400	500	600	700	800	900	1000	1200	1400	1600	1800	2000
15	3.7	5.6	7.5	9.4	11.2	13.1	15.0	16.8	18.7	22.4	26.2	29.9	33.7	37.4
20	4.8	7.3	9.7	12.1	14.5	16.9	19.4	21.8	24.2	29.0	33.9	38.7	43.6	48.4
25	5.9	8.9	11.9	14.9	17.8	20.8	23.8	26.7	29.7	35.6	41.6	47.5	53.5	59.4

NOTE : Table is based on a 68°F mixed air temperature entering the discharge heating coil

Table 14: Relationships between hydronic discharge heating and fan airflow rates

Note that the maximum MBH values in table 14 are based on a 17°F ΔT between the air entering and leaving the discharge heating coil since the incoming air was estimated to be 68°F.

ELECTRIC HEATING COILS

DLSC fan terminals can also be provided with electric resistance heaters mounted on their fan discharge. These coils are of open coil design (standard) but may also be provided in finned tubular construction. The information in this section is based on the standard open wire design.

UL-1995 and the National Electric Code (NEC) requirements for open coil duct heaters:

- Over temperature Protection – Duct heaters must be supplied with both primary and secondary over temperature protection.
- Airflow Interlocks – An airflow interlock must be provided to keep the heater from operating with extremely low or no airflow.
- Contactors – Contactors connected to the primary thermal cutout and airflow interlock safety circuits must be provided by the duct heater manufacturer.
- Over current Protection – For heaters drawing more than 48 amps, the duct heater manufacturer must provide some means of over current protection either built into the terminal box or a remote panel board.
- Disconnecting Means – All duct heater installations require a disconnecting means at or within sight of the heater controls.

DLSC terminals with electric heat comply with the UL and NEC standards detailed above. They are available with one or two steps as well as Synergy™ time proportional SCR controls. All include the following components:

- Automatic reset thermal cutouts (one per element)
- 80/20 nickel chrome heating elements
- Line voltage terminal block with single point electrical connection for entire unit
- Airflow safety switch
- Control transformer (line voltage to 24VAC)
- Magnetic contactor per step (on stepped control units)

Among the more popular options are:

- Interlocking (door) disconnect switch
- Main power supply fuses
- Manual reset thermal cutout
- Mercury (silent) contactors
- Dust-tight control box enclosures

Unit size	Heater kW range (min./max.) for various power supplies				
	208V/1Φ	240V/1Φ	277V/1Φ	208V/3Φ	480V/3Φ*
1	1.5 - 6.0				2.5 - 6.0
2	1.5 - 6.0				2.5 - 6.0
3	1.5 - 9.0				2.5 - 9.0
5	1.5 - 8.5	1.5 - 9.0	1.5 - 10.0		2.5 - 14.0
A	1.5 - 6.0				2.5 - 6.0
B	1.5 - 6.0				2.5 - 6.0
C	1.5 - 9.0				2.5 - 9.0
E	1.5 - 8.5	1.5 - 9.0	1.5 - 12.0		2.5 - 14.0

NOTE : *Wye type connection, requires fourth (neutral) wire

Table 15: DLSC electric heater availability chart

Table 15 identifies the available kW ranges for each DLSC size with various power supplies. These heaters can be furnished in one or two steps or with Titus Lynergy™ time proportional SCR control. When furnished with two-step control, each stage provides one-half of the coil's total kW output.

Equation 6.11, which is a modification of equation 6.8 can be used to calculate the temperature (T_{SUPPLY}) of the air leaving the DLSC terminal's discharge heating coil where T_{MA} is the temperature of the mixed air (induced plus primary) entering the coil.

$$T_{SUPPLY} = T_{MA} + \frac{3412 \text{ x kW}}{1.1 \text{ x } CFM_{FAN}}$$

equation 6.11

The temperature of the air mixture entering the discharge electric coil can be estimated by equation 6.12

$$T_{MA} = \frac{(CFM_{PA} \text{ x } T_{PA}) + (CFM_{IA} \text{ x } T_{IA})}{CFM_{FAN}}$$

equation 6.12

Electric heaters should not be sized to produce temperature differentials between their discharge air and the rooms they serve that exceed 20°F while a lesser 15°F differential is recommended. Equation 6.13 can be used to calculate the maximum kW for various fan airflow rates that produce a given supply to room air temperature differential.

$$kW_{MAX} = \frac{1.1 \text{ x } CFM_{FAN} \text{ x Max. } \Delta T_{SR}}{3412}$$

equation 6.13

Table 16 uses equation 6.13 to calculate maximum kW values at various fan airflow rates (CFM) that limit supply to room temperature differentials to 15, 20 and 25°F with an entering mixed air temperature of 68°F. Like was the case with table 12 and example 6.1, the SCFM values should be corrected for altitude where applicable.

Maximum supply to room air ΔT (°F)	Maximum discharge electric coil kW @ various fan airflow rates (CFM)													
	200	300	400	500	600	700	800	900	1000	1200	1400	1600	1800	2000
15	1.1	1.6	2.2	2.7	3.3	3.8	4.4	4.9	5.5	6.6	7.7	8.8	9.9	11.0
20	1.4	2.1	2.8	3.5	4.3	5.0	5.7	6.4	7.1	8.5	9.9	11.3	12.8	14.2
25	1.7	2.6	3.5	4.4	5.2	6.1	7.0	7.8	8.7	10.4	12.2	13.9	15.7	17.4

NOTE : Table is based on a 68°F mixed air temperature entering the coil

Table 16: Maximum kW values for given supply to room air differentials

There are other factors associated with selecting an electric heater. UL safety standards and the National Electrical Code (NEC) govern wiring, fusing and breaker sizes that must be used with DLSC terminals. These regulations will likely drive the decision regarding what line voltage and phase is to be used to power the DLSC terminals. Table 17 defines the EC motor voltages and phases that may be paired with various DLSC electric heater supply circuits.

EC motor voltage	Heater and line voltage/phase				
	208V/1Φ	240V/1Φ	277V/1Φ	208V/3Φ	480V/3Φ*
120					
208	X			X	
240		X			
277			X		X

NOTE : *Wye type connection, requires fourth (neutral) wire

Table 17: DLSC electric heater and EC fan motor pairings

The **Full Load Amperage (FLA)** is the current draw of a device when operating at its rated load and voltage. The FLA for a DLSC terminal is the sum of that for its fan motor and that of its electric heater. Table 3 in Chapter 3 lists the FLA for each DLSC unit size motor and voltage combination, while table 18 provides formulas for calculating the FLA of its electric heater (where applicable).

Heater amperage for various power supplies				
208V/1Φ	240V/1Φ	277V/1Φ	208V/3Φ	480V/3Φ*
kW x 4.808	kW x 4.167	kW x 3.610	kW x 2.776	kW x 1.203

NOTE : *Wye type connection, requires fourth (neutral) wire

Table 18: Calculation of electric heater FLA

The **Minimum circuit ampacity (MCA)** is the minimum current rating for all components (wiring, terminals, etc.) in a device's supply circuit. The MCA for a DLSC terminal is calculated as follows:

$$MCA = 1.25 \times (FLA_{MOTOR} + FLA_{HEATER}) \qquad \text{equation 6.14}$$

The **Maximum over current protection (MOP)** defines the maximum rating for over current protection devices such as circuit breakers and fusing in the supply circuit serving a DLSC terminal. Circuit breakers are available with standard ratings in 5 amp increments from 15 amps and up. The MOP of a DLSC terminal can be calculated as:

$$MOP = (2.25 \times FLA_{MOTOR}) + FLA_{HEATER} \qquad \text{equation 6.15}$$

The calculated MOP value is then subjected to the following to determine the actual required MOP and breaker size:

- If the calculated MOP is not a multiple of 5, round down to the nearest standard breaker/fuse size.
- If the rounded MOP is less than the MCA, the MOP is considered equal to the MCA, and is then rounded up to the nearest standard breaker (or fuse) size.
- The minimum MOP is 15 amps, the minimum code allowable fuse or circuit breaker allowed.

The examples that follow illustrate the MCA and MOP calculations for a given size DLC terminal with two different supply circuits. When the calculated MCA and/or MOP values exceed 25 amps, it is usually beneficial to resort to a 3-phase power supply to minimize the size of the breaker and other circuity components.

Example 6.2:

Determine the FLA, MCA and MOP for a size 3 DLSC with a 6.0 kW electric heater supplied by a 277V/1Φ circuit.

Solution:

Per table 15, a 277V/1Φ EC motor would be used. Table 3 (page 11) shows the FLA of the 1/2 HP motor to be 4.1 amps. The FLA for the heater is calculated (using table 18) to be 21.7 amps for a total unit FLA of 25.8 amps.

The MCA and MOP can now be calculated using equations 6.13 and 6.14, respectively:

$$MCA = 1.25 \times (FLA_{MOTOR} + FLA_{HEATER}) = 1.25 \times (4.1 + 21.7) = 32.3 \text{ amps}$$

$$MOP = (2.25 \times FLA_{MOTOR}) + FLA_{HEATER} = (2.25 \times 4.1) + 21.7) = 30.9 \text{ amps}$$

The rounded down MOP (30 A) is less than the MCA (32.3 A), thus the MCA will be rounded up to determine the circuit breaker size of 35 amps.

Example 6.3:

Calculate the FLA, MCA and MOP for the same DLSC (size 3) terminal supplied by a 480V/3Φ circuit.

Solution:

While the motor FLA remains the same (4.1 A), the heater FLA now becomes 1.203 x 6 kW or 7.2 amps.

The MCA and MOP for the terminal can now be calculated as:

$$MCA = 1.25 \times (FLA_{MOTOR} + FLA_{HEATER}) = 1.25 \times (4.1 + 7.2) = 14.1 \text{ amps}$$

$$MOP = (2.25 \times FLA_{MOTOR}) + FLA_{HEATER} = (2.25 \times 4.1) + 7.2) = 16.4 \text{ amps}$$

The corrected MOP (15 A) is greater than the MCA (14.1 A), thus a breaker size of 15 amps will be required.

Example 6.4:

Determine the FLA, MCA and MOP for a size E DLSC with a 2-row discharge hot water coil supplied by a 120V/1Φ circuit.

Solution:

Table 3 (page 11) shows the FLA of the 3/4 HP motor to be 9.6 amps. As there is no electric heat this is the total unit FLA. The MCA and MOP are thus calculated as follows:

$$MCA = 1.25 \times FLA_{MOTOR} = 1.25 \times 9.6 = 12.0 \text{ amps}$$

$$MOP = 1.25 \times FLA_{MOTOR} = 2.25 \times 9.6 = 21.6 \text{ amps}$$

The corrected MOP (20 A) is greater than the MCA (12.0 A), thus a 20 amp breaker will be required.

CHAPTER 7 TERMINAL AIRFLOW RATES AND ACOUSTICS

This chapter describes procedures for determining the primary and induced air flow rates required to perform the specified zone cooling and heating under design conditions. It then provides guidance regarding estimation of sound pressure levels and NC for the zone being served.

PRIMARY AND FAN AIRFLOW RATE CALCULATIONS

As previously stated, all the zone latent cooling and ventilation requirements must be satisfied by the primary air delivered from the air handling unit serving the DLSC terminal. The ventilation airflow rate can be calculated using procedures in the latest version of ASHRAE Standard 62.1, while the latent airflow rate required to maintain a given space humidity ratio can be calculated by equation 4.1.

$$SCFM_{LAT} = \frac{LHG}{0.69 \times (W_{PA} - W_{ROOM})}$$

where,

LHG = Zone latent cooling load, Btu/h

ΔW_{SR} = Humidity ratio differential between zone and primary air, grains H_2O per pound dry air

The primary airflow rate (SCFM) delivery to the DLSC terminal must then be the greater of that required for ventilation and zone latent cooling. In most cases, the latent cooling airflow requirement exceeds the ventilation rate and thus determines the DLSC's primary airflow rate.

Once the primary airflow rate has been established, its contribution to the zone sensible cooling can be calculated using equation 7.1

$$q_{PA} = 1.1 \times SCFM_{PA} \times (T_{ZONE} - T_{PA}) \qquad \text{equation 7.1}$$

Equations 4.1 and 7.1 are based on standard air ($\rho = 0.075$ lbm/ft^3) at sea level. When designing for other altitudes, their constants (0.69 and 1.1, respectively) should be multiplied by the correction factor (C_{ALT}) calculated by equation 6.8:

Fan airflow rate determination

The DLSC fan airflow rate is typically dependent upon the zone design sensible cooling load. The induction cooling coil is responsible for the specified sensible cooling less the primary air contribution (calculated by equation 7.1). The determination of the DLSC induced airflow rate will depend on the DLSC model and size chosen as well as the coil's chilled water supply temperature and flow rate. Tables 7 and 8 in Chapter 6 can be used to determine the cooling design induced airflow rate. Again, the airflow rates in the table are SCFM and should be corrected for altitude where necessary. The fan (supply) airflow rate is equal to the sum of the DLSC primary and induced airflow rates.

In the event it is desired to establish different fan airflow rates for DLSC cooling and heating operation, the fan CFM during heating operation should be sufficient to maintain an acceptable temperature differential between the coil leaving air temperature and the space they serve. ASHRAE Standard 62.1 recommends that this differential be limited to 15°F to maximize the delivery of ventilation air to the space occupants. In addition, the differential should be limited to make sure the heated air is projected to within 4.5 feet of the floor.

ACOUSTICS

Most DLSC applications will involve providing comfort to occupied areas of the building, As HVAC equipment often create a large part of the zone's background noise, acoustical considerations often limit the selection of the DLSC terminals and their discharge ductwork.

There are a few acoustical fundamentals that should be understood before attempting to analyze the zonal acoustical performance of the HVAC system.

Discharge noise enters the space through supply air outlets ducted to the fan terminal(s). This noise is created by HVAC components in the air stream and is conveyed through the supply air ductwork serving the space.

Insertion loss refers to sound absorption that occurs as the airstream passes through an HVAC component. For example, the lining inside a sheet metal duct exerts an insertion loss along the path between the DLSC and the space it serves.

Regenerated noise is that which is added to the airstream as it passes through an HVAC component such as a supply diffuser.

Radiated noise includes terminal unit and component breakout noise that is transmitted into the plenum in which then DLSC is mounted. DLSC radiated power level data includes noise that escapes through the DLSC induction coil.

Sound power levels (L_W) are a measure of sound energy output expressed in decibels compared to a reference source of 10^{-12} Watts. It is expressed in decibels (dB) and calculated by equation 7.2:

$$L_W = 10 \log (W_{SOURCE} / W_{REF}) \qquad \text{equation 7.2}$$

Terminal unit manufactures are required to catalog individual sound power level values for octave band center frequencies (2 thru 7) of 125, 250, 500, 1000, 2000, 4000 Hz.

Sound Pressure Levels (L_P) are measured directly by sound level meters at one or more points in a room. As a result, they differ from the sound power level (L_P) according to the attenuation effects of the system and surroundings in which the device is applied. Sound pressure levels are again expressed in decibels (dB).

Noise criteria (NC) values are single number ratings often used to describe room noise levels in a specific application. Figure 25 illustrates noise criteria (NC) curves that can be referenced to generate single number NC values based on individual octave band sound pressure levels (L_P). These curves were developed to represent lines of equal hearing perception in all bands and at varying sound levels.

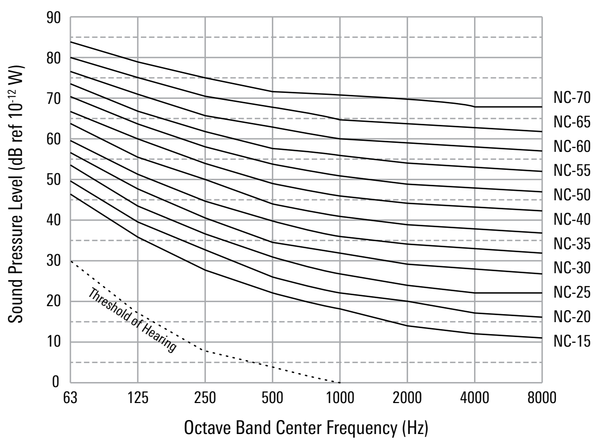


Figure 25: Noise criteria (NC) curve

NC level	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
15	36	29	22	17	14	12
20	40	33	26	22	19	17
25	44	37	31	27	24	22
30	48	41	35	31	29	28
35	52	45	40	36	34	33
40	56	50	45	41	39	38
45	60	54	49	46	44	43
50	64	58	54	51	49	48

Table 19: Tabular values for sound pressure L_P levels coincident with NC curves

Table 19 presents tabular octave band sound pressure (L_P) values that relate to the NC curves depicted in figure 25.

A product's estimated sound pressure level (L_P) performance curve is obtained by subtracting space (or other appropriate) sound attenuation effects from the unit's individual octave band discharge or radiated sound power levels (L_W). These values are then plotted the NC curve and the single number NC is reported as the highest NC value associated with the plotted individual sound pressure data points.

Office buildings	
Executive offices	NC 25 - 30
Private offices	NC 30 - 35
Open plan offices	NC 35 - 40
Conference rooms	NC 25 - 30
Public circulation areas	NC 40 - 45
Hospitals and clinics	
Private rooms	NC 25 - 30
Wards	NC 30 - 35
Operating rooms	NC 25 - 30
Laboratories	NC 35 - 40
Corridors	NC 30 - 35
Public circulation areas	NC 35 - 40
Religious facilities	NC 30 - 35
Educational facilities	
K-12 classrooms	NC 25 - 30
Lecture halls	NC 30 - 35

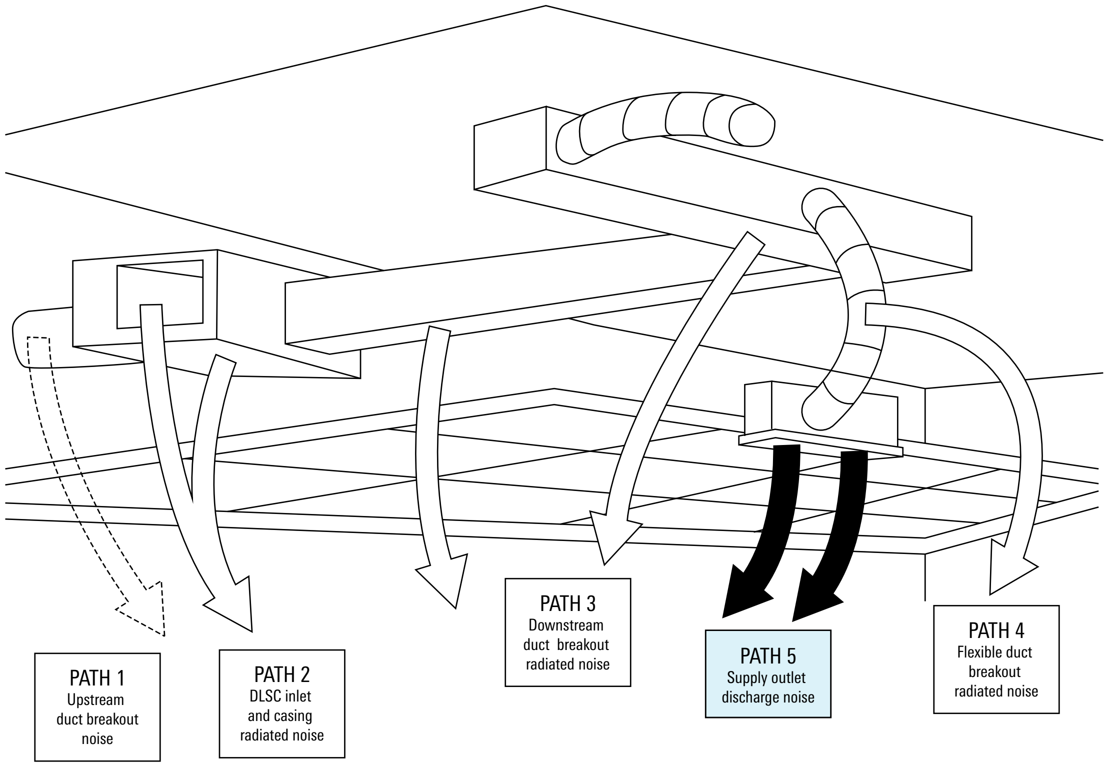
Table 20: Recommended room design NC levels

HVAC system sound paths

Figure 26 illustrates the various pathways for discharge and radiated noise transmission from a DLSC terminal. The DLSC terminal itself is both a regenerative and radiated noise source. Its connected ductwork attenuates duct-borne noise while transmitting breakout noise. When the plane wave sound passes from a small space such as a duct into a large space the size of a room, a certain amount of sound is reflected back into the duct, often significantly reducing discharge low frequency sound. This effect is referred to as end reflection.

Supply air outlets also generate noise, albeit mostly discharge noise. As such, the supply air outlets must be considered as a separate regenerative discharge sound source when calculating room sound pressure levels.

The ceiling material and plenum serve to attenuate radiated noise while the surfaces of the room absorb both discharge and radiated noise. The various sound paths are described below.



Reference AHRI Standard 885-2008

Figure 26: Sound pathways for DLSC fan terminal unit room acoustical calculations

Path 1: Upstream duct breakout noise

This is noise from upstream HVAC components that may be transmitted through the walls of ductwork feeding the DLSC terminal. Neither this nor any associated sound that may be transmitted to DLSC terminal inlet are included in the DLSC published sound power levels.

Path 2: DLSC inlet escape and casing radiated noise

This is noise that is transmitted from the DLSC terminal to the space around it. It is subsequently reduced by the absorption of the plenum surfaces, applicable ceiling tiles and the room effect.

Paths 3 and 4: Downstream ductwork radiated (breakout) noise

This is breakout noise transmitted from the surfaces of ductwork downstream of the DLSC terminal. It is also reduced by the absorption of the plenum surfaces, ceiling tiles and the room effect. These sound sources do not generally affect the overall radiated sound calculations as they are usually significantly lower than those transmitted along Path 2.

Path 5: Supply outlet discharge noise

Path 5 represents the combination of the terminal unit discharge sound power levels and noise regenerated by the supply outlets serving the space. Terminal unit discharge sound power levels are reduced by the losses of the connected downstream ductwork and other components. The end reflection loss is then applied to individual octave band L_W values to determine the corrected sound power (L_P) levels of the terminal discharge noise.

In the event the terminal serves multiple supply outlets, a flow division factor is applied to reduce the sound power levels attributed to each airflow split.

Applying terminal unit and supply outlet acoustical data

Manufacturers of HVAC terminal units test and rate their noise levels according to ASHRAE Standard 130 (2016 version as of this writing) and AHRI Standard 880-2017. These Standards document the necessary procedures for determining octave band sound power levels for the terminals.

AHRI Standard 885-2008 establishes methods for converting sound power (L_W) to sound pressure (L_P) and specifies standardized downstream system insertion losses and radiated noise transmission losses that govern the calculation of published sound pressure NC values.

Tables 21 and 22 detail the attenuation factors that AHRI Standard 885, appendix E requires to be applied to manufacturers' terminal unit discharge and radiated sound power (L_W) levels to calculate single number NC values. These values are considered to represent a "typical" application.

Terminal unit size	Discharge noise basis of adjustment	Octave band center frequency, Hz					
		125	250	500	1000	2000	4000
Small (Fan CFM < 300)	Lining reduction (based on 5 ft. of 1" lined 8 x 8 discharge duct)	2	6	12	25	29	18
	Flow division (based on a single supply outlet)	0	0	0	0	0	0
	End reflection (8" diameter flex duct)	10	5	2	1	0	0
	Flex duct reduction (8" diameter, lined flex)	5	10	18	19	21	12
	Enviromental adjustment factor	2	1	0	0	0	0
	Space effect (2400 ft^3 space, 5 ft. source to receiver)	5	6	7	8	9	10
	Total attenuation	24	28	39	53	59	40
Medium (300 to 700 fan CFM)	Lining reduction (based on 5 ft. of 1" lined 12 x 12 discharge duct)	2	4	10	20	20	14
	Flow division (based on two supply outlets)	3	3	3	3	3	3
	End reflection (8" diameter flex duct)	10	5	2	1	0	0
	Flex duct reduction (8" diameter, lined flex)	5	10	18	19	21	12
	Enviromental adjustment factor	2	1	0	0	0	0
	Space effect (2400 ft^3 space, 5 ft. source to receiver)	5	6	7	8	9	10
	Total attenuation	27	29	40	51	53	39
Large (Fan CFM > 700)	Lining reduction (based on 5 ft. of 1" lined 15 x 15 discharge duct)	2	3	9	18	17	12
	Flow division (based on three supply outlets)	5	5	5	5	5	5
	End reflection (8" diameter flex duct)	10	5	2	1	0	0
	Flex duct reduction (8" diameter, lined flex)	5	10	18	19	21	12
	Enviromental adjustment factor	2	1	0	0	0	0
	Space effect (2400 ft^3 space, 5 ft. source to receiver)	5	6	7	8	9	10
	Total attenuation	29	30	41	51	52	39

Table 21: AHRI Standard 885-2008, appendix E discharge noise attenuation factors

Radiated noise basis of adjustment	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
Ceiling plenum and space attenuation effect	16	18	20	26	31	36
Enviromental adjustment factor	2	1	0	0	0	0
Total attenuation	18	19	20	26	31	36

Table 22: AHRI Standard 885-2008, appendix E radiated noise attenuation factors

The **space effect** in table 21 is calculated using the Schultze equation (equation 7.3), where R (ft) is the distance from the outlet to the receiver and V is the volume (ft^3) of the room.

$$10 \text{ Log }(R) + 5 \text{ Log }(V) + 3 \text{ Log }(Hz) - 25 \quad \text{equation 7.3}$$

The ceiling plenum and space attenuation in table 22 combines the attenuation effects of the plenum (in which the terminal is mounted), the insertion loss of the ceiling material through which the radiated sound passes and the room absorption itself (as calculated by equation 7.3).

The environmental adjustment factor in Tables 21 and 22 is used to convert octave band sound pressure levels measured under free field conditions to that of an actual confined space which behaves more like a reverberation chamber.

AHRI-885-2008 also tabulates attenuation and transmission loss data for other applications. In addition, a Duct Discharge Calculation Spreadsheet that facilitates the discharge, radiated and overall room sound level calculations can be downloaded from the AHRI website (www.ahrinet.com).

Catalog NC value	Octave band center frequency, Hz					
	125	250	500	1000	2000	4000
15				27	28	22
20				32	29	27
25				37	34	32
30				41	39	38
35				46	44	43
40				51	49	48

Supply outlet sound power levels negligible

Table 23: Estimation of outlet regeneration octave band sound power levels

Manufacturers often catalog supply outlet sound levels in the form of a single number sound pressure NC which includes 10 dB room attenuation across their acoustical spectra. Table 23 can be used to estimate their individual octave band regenerated sound power levels. The table assumes that the published NC value occurs across octave band frequencies between 1000 and 2000 Hz. Supply outlet regenerated noise in the lower frequencies rarely affects the room sound pressure level and can generally be neglected. The table then adds 10 dB to each of the applicable octave bands to estimate the outlet's octave band sound power levels. This can then be corrected by the applicable space effect to arrive at the outlet regenerated sound pressure level.

The subsequent calculation of a single room NC value requires that the terminal's individual octave band radiated, discharge and the supply outlet regeneration sound pressure levels be added logarithmically. Equation 7.4 can be used to estimate the combined sound pressure (L_p) levels of the three applicable sound paths.

$$SPL_{ROOM} = 10 \times LOG_{10}\left(10^{SPL\ DISCHARGE/10} + 10^{SPL\ RADIATED/10} + 10^{SPL\ OUTELT/10}\right)$$

equation 7.4

The resultant room NC level is then determined by plotting the octave band sound power levels generated by equation 7.6 on the NC curve from figure 25 or deriving it from table 19.

Examples 7.1 and 7.2 that follow illustrate the room NC calculation procedure.

Example 7.1

Calculate the discharge and radiated octave band sound pressure (L_p) levels of a size 3-08 DLSC terminal with a 4- row induction cooling coil and a 2-row discharge heating coil serving three identical spaces and delivering primary and fan airflow rates of 140 and 700 CFM, respectively. Apply the discharge and radiated noise attenuation assumptions detailed in AHRI 885-2008 appendix E.

Solution:

The discharge and radiated sound power levels of the size 3-08 DLSC are presented in table 24:

Unit size	Fan CFM	Primary air CFM	Octave band sound power levels, L_W (dB)											
			Discharge (fan + primary air)						Radiated (fan + primary air)					
			125	250	500	1000	2000	4000	125	250	500	1000	2000	4000
3	800	160	73	66	63	62	60	57	65	63	60	58	50	43

Table 24: Octave band sound power levels for size 3 DLSC in examples

Discharge sound power levels can be converted to octave band sound pressure levels (see table 25 below) by subtracting the specified attenuation values summed up in table 21 (for a large fan terminal):

Conversion from discharge sound power to sound pressure	Octave band center frequency (Hz)					
	125	250	500	1000	2000	4000
Discharge sound power level (L_W)	73	66	63	62	60	57
Discharge attenuation (table 21)	29	30	41	51	52	39
Discharge sound pressure level (L_p)	44	36	22	11	8	18

Table 25: Conversion of discharge sound power levels to sound pressure

Radiated sound power levels can be converted to octave band sound pressure levels (see table 26) by subtracting the specified attenuation values from table 22:

Conversion from radiated sound power to sound pressure	Octave band center frequency (Hz)					
	125	250	500	1000	2000	4000
Radiated sound power level (L_W)	65	63	60	58	50	43
Radiated attenuation (table 22)	18	19	20	26	31	36
Radiated sound pressure level (L_p)	47	44	40	32	19	7

Table 26: Conversion of radiated sound power levels to sound pressure

Example 7.2

Given the discharge and radiated sound power levels from example 7.1, calculate the room sound pressure NC level in a space with a single supply outlet catalogued as NC 25 based on a room attenuation of 10 dB.

Solution:

The individual octave band sound pressure levels of the NC 25 outlet must first be estimated. Table 23 can be used to estimate octave band sound power levels regenerated by an NC 25 outlet. Space effect values from table 21 can then be subtracted from the sound power levels to calculate the regenerated sound pressure of the outlet (see table 27). Equation 7.4 can then be used to logarithmically add the sound pressure levels of the three defined sound sources in each octave band. The resultant room sound pressure level NC is determined using table 19 and presented in table 28.

Conversion from outlet regenerated sound power to sound pressure	Octave band center frequency (Hz)					
	125	250	500	1000	2000	4000
Regenerated sound power level (L_W)	-	-	-	27	24	22
Space effect (as calculated by equation 7.3)	-	-	-	8	9	10
Regenerated sound pressure level (L_p)	-	-	-	19	15	12

Table 27: Converting supply outlet regenerated noise to octave band sound pressure

Room sound pressure NC calculation	Octave band center frequency (Hz)					
	125	250	500	1000	2000	4000
DLSC discharge sound pressure level (L_p)	44	36	22	11	8	18
DLSC rdiated sound pressure level (L_p)	47	44	40	32	19	7
Combined DLSC sound pressure level (L_p)	49	44	40	32	19	19
Diffuser regenerated noise sound pressure level (L_p)	-	-	-	19	15	12
Combined room sound pressure level (L_p)	49	44	40	33	21	19
Room sound pressure NC level (L_p) per band*	31	34	35	33	21	20
Room sound pressure NC level	35					

Table 28: Calculating the resultant room NC level for example 7.2

Table 29 can be used to estimate maximum fan airflow rates that result in specific radiated and discharge NC levels for DLSC terminals. The NC values in this table are again derived using the attenuation factors prescribed in AHRI Standard 885, appendix E.

Unit size	Max. fan CFM for various discharge NC values					Max. fan CFM for various radiated NC values				
	NC 25	NC 30	NC 35	NC 40	MAX	NC 25	NC 30	NC 35	NC 40	MAX
1	490	575			34	450	650			34
2	750				26	600	775			30
3	675	1,025			33	350	575	875		39
5	1,075	1,450	1,900		36	575	850	1,375	1,950	40
A	550	645	800		35	440	575	675		40
B	600	800			30	590	725			32
C	800	1,000	1,250		37	575	975			34
E	1,225	1,525	1,800		36	800	1,075	1,775		37

Table based upon:

1. NC levels determined using AHRI 885-2008 Appendix E.
2. Inlet static pressure of 1.0 in.w.g.
3. Downstream static pressure of 0.25 in.w.g.
4. Primary airflow rate 20% that of rated unit fan CFM
5. Maximum NC is value at unit rated fan CFM

NC level above maximum for unit

Table 29: Maximum fan airflow rates for various NC levels

DLSC fan terminals can be configured for cooling only or to provide both cooling and heating functions. While the cooling function utilizes a chilled water sensible cooling coil mounted on the unit's induction air inlet, the heating function may be provided by either a hydronic or electric resistance heating coil. Electric coils are always mounted on the terminal unit's discharge. Although hydronic heating coils may be incorporated into the induction heat transfer coil or mounted on the unit discharge, they are more commonly mounted on the discharge to reduce the operating temperature of the fan motor during heating.

In order to size and select DLSC terminals, the design conditions and loads related to the zone the terminal(s) will serve must be identified. These include:

1. Zone sensible and latent cooling loads
2. Space heating loads (where applicable)
   a. Occupied net heating load
   b. Night setback heating load
3. Space ventilation airflow rate
4. Zone design conditions
   a. Summer operation design temperature and moisture content
   b. Winter operation design temperature
   c. Night setback (winter) design temperature

When applying hydronic cooling and heating coils in cold climates, it is often necessary to add glycol to the water supplying the coil. Doing so, reduces the heat transfer capacity of the coil. Table 13 provides correction factors for various glycol concentrations that can be applied to correct the coil capacities in tables 7, 8, 9, 10 and 12.

The examples that follow illustrate the method of selecting DLSC terminals to deliver zone sensible cooling and heating.

Example 8.1:

A 1,000 square foot interior zone of an office building in Boston, Massachusetts has the following cooling and ventilation loads:

- Sensible cooling – 11 Btu/h-ft^2
- Latent cooling – 1.5 Btu/h-ft^2
- Ventilation airflow rate – 100 CFM

The space is to be maintained at 75°F and 50% RH (humidity ratio 65 grains). Primary air is to be delivered at 55°F and a humidity ratio of 50 grains. Chilled water will be delivered at 57°F. The maximum zone NC level is 35.

Solution:

First, the primary airflow rate required to deliver the zone latent cooling must be determined. In this case the total latent cooling load is 1,500 Btu/h and the humidity ratio differential between the room and primary air is 15 grains. Equation 4.1 can be used to calculate the latent airflow rate. Note that it is expressed in SCFM and not corrected for altitude as Boston is very near sea level.

$$SCFM_{LAT} = \frac{LHG}{0.69 \times (W_{ROOM} - W_{PA})} = \frac{1.500}{0.69 \times (65 - 50)} = 145$$

Next the primary air contribution to the zone sensible cooling is calculated using equation 7.1

$$q_{PA} = 1.1 \times SCFM \times (T_{ZONE} - T_{PA}) = 1.1 \times 145 \times (75 – 55) = 3{,}190 \text{ Btu/h}$$

The sensible cooling coil will be responsible for providing the difference between the zone sensible cooling demand (11,000 Btu/h) and that delivered by the primary air (3,190 Btu/h), or 7,810 Btu/h.

Table 7 indicates that a size 2 DLSC with a six-row induction cooling coil can provide 7.9 MBH of sensible cooling at a chilled water flow rate of 3.5 GPM and an induced airflow rate of 525 CFM. The total fan airflow rate would then be the sum of the primary and induced airflow rates, or 670 CFM. Table 29 indicates this would be accomplished within the maximum NC constraint of 35.

Example 8.2:

A 900 square foot elementary school classroom in Chicago, Illinois It is to be served by a DLSC terminal with hydronic discharge heat. The maximum classroom discharge NC level is 30. The design sensible heating and cooling loads are as follows:

- Sensible cooling – 20.5 MBH
- Latent cooling – 4,500 Btu/h
- Heating (based on minimum occupancy) – 8 MBH
- Heating (based on night setback temperature) – 10 MBH
- Ventilation airflow rate – 398 CFM

The space is to be maintained at 75°F and 50% RH (humidity ratio 65 grains) during cooling operation. Primary air during cooling operation will be delivered at 55°F and a humidity ratio of 50 grains. Chilled water is delivered at 57°F and treated with 40% propylene glycol (by volume), providing freeze protection down to -6°F

The heating coil must be sized to maintain a night setback temperature of 60°F. Hot water will be delivered at 115°F. During occupied operation, the primary air will be delivered at 60°F and the supply to room air temperature differential (ΔT_{SR}) shall not exceed 15°F.

Solution:

First, the primary airflow rate required to deliver the zone latent cooling must be determined. In this case the total latent cooling load is 4,500 Btu/h and the humidity ratio differential (ΔW_{RS}) is 15 grains. No altitude correction is necessary thus equation 4.1 can be used to calculate the design latent cooling airflow rate as follows:

$$SCFM_{LAT} = \frac{LHG}{0.69 \text{ x } (W_{PA} - W_{ROOM})} = \frac{4500}{0.69 \text{ x } (60 - 50)} = 435$$

As this exceeds the ventilation rate, the latent cooling CFM determines the primary airflow rate. The primary air inlet diameter recommended for handling 435 CFM will be 8 inches according to table 3.

We can now calculate the sensible cooling contribution of the primary air using equation 7.1

$$q_{PA} = 1.1 \text{ x } SCFM \text{ x } (T_{ZONE} + T_{PA}) = 1.1 \text{ x } 435 \text{ x } (75 + 55) = 9{,}570 \text{ Btu/h}$$

The cooling coil must therefore deliver the remaining 10.9 MBH of sensible cooling. As the chilled water serving the coil is treated with 40% propylene glycol, the coil's sensible cooling delivery must be corrected for the heat transfer correction factor (F60 = 0.88) from table 13:

$$MBH_{CORR} = \frac{MBH_{COIL}}{F_{60}} = \frac{10.9}{0.88} = 12.4$$

Table 7 indicates that a size 5 DLSC with a 6-row coil can deliver as much as 13.2 MBH of sensible cooling at an induced airflow rate of 975 CFM and a chilled water flow rate of 3 GPM. The fan airflow rate is thus equal to the sum of the primary and induced airflow rates or 1,410 CFM. A quick check of table 29 confirms that the discharge NC level at 1,410 CFM is below 30.

Now we must determine that the unit size and fan CFM are sufficient for heating.

During times of minimum occupancy, the DLSC must deliver the classroom heating demand while offsetting the cooling delivered by the primary air. The primary air delivery of 435 CFM at 60°F (10°F below the design winter room temperature) results in 4.8 MBH of sensible cooling which must be added to the room heating demand (8 MBH) for a total discharge coil heating load of 12.8 MBH. This is greater than the night setback (primary air off) heating requirement of 10 MBH and thus defines the design discharge coil's heating capacity.

The temperature of the coil's entering air mixture can be calculated using equation 6.12.

$$T_{MA} = \frac{(CFM_{PA} \text{ X } T_{PA}) + (CFM_{IA} \text{ X } T_{IA})}{CFM_{FAN}} = \frac{(435 \text{ X } 60) + (975 \text{ X } 70)}{(435 + 975)} = 66.9° \text{ F}$$

Tables 9 and 10 presents DLSC hydronic discharge coil heating performance. They are based on a 50°F temperature difference between the hot water and airstream entering the coil. As the design will be based on a 48°F differential, the MBH must be adjusted slightly using the correction factor (0.96) from table 11:

$$MBH_{CORR} = \frac{MBH_{COIL}}{CF_{TABLE11}} = \frac{12.8}{0.96} = 13.3$$

Interpolation within table 9 indicates that the size 5 DLSC with a 1-row discharge heating coil and a water flow rate of 2.5 GPM delivers 13.4 MBH of heat when operating with the 1,410 CFM fan airflow required by the sensible cooling load.

The occupied heating supply air temperature can then be calculated using equation 6.8:

$$T_{SUPPLY} = T_{MA} + \frac{1000 \text{ x } MBH_{CORR}}{60 \text{ x } \rho_{IA} \text{ x } C_p \text{ x } CFM_{FAN}} = 66.9 + \frac{1{,}000 \text{ x } 13.4}{60 \text{ x } 0.075 \text{ x } 0.24 \text{ x } 1{,}410} = 75.7°\text{F}$$

Alternatively, the occupied heating fan CFM could have been reduced to deliver a supply air temperature of 85°F, which corresponds to the upper supply air to room differential limit of 15°F. Table 30 presents both the base scenario where the DLSC fan CFM during heating is kept the same as that for cooling and the alternative fan CFM based on maintaining the maximum 15°F supply to room temperature differential. The latter would incorporate a 2-row heating coil with a 1.7 GPM hot water flow rate.

Unit Size	MBH_{ZONE}	MBH_{CORR}	Scenario	CFM_{FAN}	T_{ROOM} (°F)	T_{PA} (°F)	T_{MIXED} (°F)	T_{SUPPLY} (°F)	Heating coil rows	GPM_{HW}
5	12.8	13.4	Original	1,410	70.0	60.0	66.9	75.7	1	2.5
			Alternate	505	70.0	60.0	61.4	*85.0*	2	1.7

Table 30: Alternative DLSC occupied heating operation

Figure 21 suggests that the EC motor operating at 1,410 CFM (72% of the size 5 rated CFM of 1,950) would draw 88% of its rated power while operation at the reduced 505 CFM (26% of its rated CFM) would draw only 62%, a fan energy savings of about 30%.

Example 8.3:

A DLSC terminal with electric heat is required to condition a perimeter zone in an office building in Denver, Colorado. The DLSC discharge NC level shall not exceed 35. The zone cooling and heating loads are as follows:

- Sensible cooling – 10 MBH
- Latent cooling – 550 Btu/h
- Heating (based on minimum occupancy) – 6 MBH
- Heating (based on 60°F night setback temperature) – 10 MBH
- Ventilation air rate – 45 CFM

The space is to be maintained at 75°F and 50% RH (humidity ratio 79 grains) during cooling operation. Primary air will be delivered at 55°F and a humidity ratio of 66 grains. During winter occupation, the room temperature will be at 70°F and primary air will be delivered at 60°F.

During occupied heating operation, the supply to room air temperature differential (ΔT_{SR}) should not exceed 15°F. The DLSC will operate with a single point connection delivering power at 277V (single phase).

Solution:

First, the primary airflow rate required to deliver the zone latent cooling must be determined. The altitude correction factor (C_{ALT}) for Denver's altitude of 5,285 feet can be calculated by equation 6.5:

$$C_{ALT} = [1 - (5{,}285 \times 6.874 \times 10^{-6})]^{5.2599} = 0.823$$

The airflow required to handle the zone latent load can then be determined by correcting equation 4.1 for altitude:

$$CFM_{LAT} = \frac{LHG}{C_{ALT} \times 0.69 \times (W_{PA} - W_{ROOM})} = \frac{550}{0.823 \times 0.69 \times (79 - 66)} = 75$$

This is greater than the zone ventilation airflow rate and thus determines the DLSC primary airflow rate.

Next the primary air sensible cooling contribution is calculated using a modified equation 7.1:

$$\begin{aligned} q_{PA} &= 0.823 \times 1.1 \times CFM \times (T_{ZONE} + T_{PA}) \\ &= 0.823 \times 1.1 \times 75 \times (75 - 55) = 1{,}358 \text{ Btu/h} \end{aligned}$$

The remaining sensible cooling (about 8.7 MBH) must be provided by the DLSC cooling coil. Table 8 indicates that a size C DLSC with a 4-row cooling coil can provide 8.7 MBH of sensible cooling at an induced airflow rate of 625 SCFM and a chilled water flow rate of 3.0 GPM. This induced airflow rate, however, is based on standard air at sea level and must be corrected for altitude as follows:

$$CFM_{CORR} = \frac{SCFM}{C_{ALT}} = \frac{625}{0.823} = 760$$

The fan CFM is the sum of the induced and primary airflow rates, or 835 CFM. Table 29 indicates that the size C DLSC delivering 835 CFM will result in a discharge NC well below the required NC 35.

Next, the electric heater must be selected. During occupied periods, the electric coil must be capable providing the design zone heating plus an amount required to offset any primary air cooling. The cooling delivery of the primary air during heating can be calculated, again using equation 7.1, corrected for altitude:

$$\begin{aligned} q_{PA} &= 0.823 \times 1.1 \times CFM \times (T_{ZONE} + T_{PA}) \\ &= 0.823 \times 1.1 \times 75 \times (70 - 60) = 679 \text{ Btu/h} \end{aligned}$$

Thus, the electric coil must be sized to provide at least 6.7 MBH of heat during periods of minimal occupancy. This is less than the night setback heating requirement of 10.0, thus the electric heater will be sized for setback operation.

$$kW = \frac{MBH}{3{,}412} = \frac{10.0 \times 1{,}000}{3{,}412} = 2.9$$

As electric heaters are generally available in 0.5 kW increments, a 3.0 kW heater will be selected. The temperature (T_{MA}) of the primary and induced air mixture entering the coil is calculated by equation 6.12:

$$T_{MA} = \frac{(CFM_{PA} \text{ x } T_{PA}) + (CFM_{IA} \text{ x } T_{IA})}{CFM_{FAN}} = \frac{(75 \text{ x } 60) + (760 \text{ x } 70)}{(75 + 760)} = 69.1^{\circ}F$$

The supply air temperature leaving the coil can now be calculated using equation 6.11 corrected for altitude:

$$T_{SUPPLY} = T_{MA} + \frac{3{,}412 \text{ x kW}}{C_{ALT} \text{ x } 1.1 \text{ x } CFM_{FAN}} = 69.1 + \frac{3{,}412 \text{ x } 3.0}{0.823 \text{ x } 1.1 \text{ x } 835} = 82.6^{\circ}F$$

This supply to room air temperature differential is only 12.6°F, well below the recommended 15°F maximum.

The DLSC with its 3.0 kW electric heater must be provided over current protection according to the NEC code. To determine the required circuit breaker size, we first need to calculate the full load amperage (FLA) of the terminal. Per table 3 the 277V motor FLA is 4.1 amps. The 277V (single phase) heater FLA can be calculated using the multiplier for that voltage/phase from table 18

$$FLA = FLA_{MOTOR} + FLA_{HEATER} = 4.1 + (3.61 \text{ x } 3.0) = 14.9 \text{ amps}$$

The minimum circuit ampacity (MCA) and maximum over current protection (MOP) can be calculated using equations 6.13 and 6.14, respectively:

$$MCA = 1.25 \text{ x } (FLA_{MOTOR} + FLA_{HEATER}) = 1.25 \text{ x } (4.1 + 18.6) = 18.7 \text{ amps}$$

$$MOP = (2.25 \text{ x } FLA_{MOTOR}) + FLA_{HEATER} = (2.25 \text{ x } 4.1) + 10.8) = 20.0 \text{ amps}$$

The MOP value (20 A) is standard breaker size and will not be rounded down. This is greater than the MCA (18.7 A) therefore a breaker size of 20 amps is required.

As is evidenced by the examples in chapter 8, manual selection of DLSC terminals can involve significant effort. Arriving at the best solution involves not only identifying the heat transfer capacities of the unit cooling and heating coils but also consideration of numerous performance criteria including acoustics, zone ventilation effectiveness, electrical circuitry and operating energy.

To facilitate this, Titus offers designers state of the art selection software for DLSC terminals that can be downloaded from the Titus website (www.titus-hvac.com). The Excel™ based software selects the DLSC terminals based on the following user inputs:

Project specific information

- Project name and altitude
- Room and primary air design conditions for cooling and heating (where applicable) operation
- Chilled and hot water (where applicable) supply temperatures and additives
- Electrical power supply (voltage and phase)

Zone specific information

- Zone (sensible and latent) cooling and heating loads (where applicable)
- Zone ventilation CFM requirements
- Number of DLSC terminals intended to serve the zone

Unit specific information

- Heating coil type and location (e.g., discharge or induction side hydronic coil, discharge electric coil) where applicable
- Casing liner material and construction
- Inlet and downstream pressure loss
- Primary air inlet size (may be fixed by user)

In addition, the user may specify limits regarding:

- Maximum unit height
- Maximum allowable sound levels (maximum sound pressure NC or individual octave band power levels)
- Maximum coil water side pressure loss
- Maximum heating temperature differential between the supply and room air
- Maximum fan CFM as a percentage of the unit's rated CFM capacity (85% recommended)

Based on these user inputs, the selection software presents the user with a minimum of two DLSC configurations which will provide the specified performance. It also provides performance criteria for each selection:

- Fan and primary airflow rates, as well as fan static pressure
- DLSC sensible cooling capacity, cooling coil rows, water flow rates and water side pressure loss
- Hydronic heating coil rows, hot water flow rates and water side pressure loss (where applicable)
- Electric heating coil kW and amperage requirements (where applicable)
- Motor horsepower, unit MCA and MOP for specified power supply
- Cooling and heating (where applicable) leaving air and water temperatures
- Discharge and radiated sound pressure NC levels as well as individual octave band sound power levels

Finally, the program assigns a "cost factor" to each selection it presents. The lowest cost selection is presented with a cost factor of 1.0 while the other presented selections have a higher cost factor which can be used to estimate the cost penalty represented by that selection.

Titus also offers First Cost Comparison software that allows designers to compare the first costs of air-water systems to their all-air VAV counterparts. This program, designed around office buildings, allows the user to input the size, ventilation and design load characteristics of their specific building and compare estimated first costs for different combinations or air-water and all-air VAV systems. Air-water systems that may be chosen include:

- Active chilled beams supplied by constant or variable volume primary air distribution systems
- Sensible cooling fan terminals with or without hydronic zone heating coils.
- Active chilled beams (variable or constant volume) with decoupled perimeter fin tube heating

Other user input options include:

- Building location: User chooses between 55 North American cities. Program applies altitude, outdoor design conditions and city material and labor cost indexes accordingly.
- General building information: Physical data describing the building floorplate(s).
- Interior zones: Occupancy, ventilation rates, design sensible cooling load and diversity factor as well as the systems chosen for comparison in these zones.
- Perimeter zones: Depth, amount of glass, occupancy, ventilation rates and design sensible cooling and heating loads and cooling diversity factor as well as the systems chosen for comparison in these zones.
- Air-handling unit type: User specifies whether a central mixing type or DOAS air handling unit is to be used with each system.

The program constructs detailed models of the air and water distribution systems for each alternative. The information is then used to determine the capacity of the central system equipment (air handling units, refrigeration plant, boiler, pumps, etc.) associated with each system. R.S. Means Mechanical Cost Data (including city cost multipliers for the given building location) is applied to each component of the two systems to calculate its comparative first cost.

Table 31 illustrates a typical cost comparison summary generated by the program. This building was located in Washington, DC and was 200,000 square feet and 10 floors tall. Interior spaces were served by active chilled beams while perimeter spaces used sensible cooling fan terminals with hot water discharge heating coils. Note that the four major cost categories are further subdivided and that all costs are displayed as total and cost per square foot of floor area.

Summary Table				
System Type	All-air VAV		Air/water system	
	Cost $	$/ft²	Cost $	$/ft²
Central Equipment Costs	$2,599,897	$13.00	$2,007,126	$10.04
Air handling units and exhaust fans	$1,777,938		$1,087,465	
Chillers, cooling towers and heat exchangers	$638,147		$631,522	
Boilers	$15,222		$14,999	
Pumps	$168,590		$273,140	
Air and water distribution costs	$2,162,668	$10.81	$3,322,700	$16.61
Ductwork and insulation	$835,004		$793,003	
Supply/exhaust risers and louvers	$185,518		$181,877	
Piping and insulation (includes piping risers)	$305,458		$1,093,956	
Air terminal units	$585,743		$706,376	
Supply/return grilles and diffusers	$250,945		$184,844	
Active beams	$0		$362,646	
Fin tube heating elements with covers	$0		$0	
Valves, controls and hose kits	$400,468	$2.00	$587,711	$2.94
Other costs	$364,518	$1.82	$398,255	$1.99
Fire, smoke and balancing dampers	$233,467		$224,684	
Testing and Balancing	$131,051		$173,571	
Total Installed Cost of System:	$5,527,551	$27.64	$6,315,793	$31.58

Table 31: First cost summary generated by software program

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