Related Commercial Resources

CHAPTER 55. RADIANT HEATING AND COOLING

1. APPLICATIONS

BASED on the global push towards high performance buildings, current application knowledge and readily available equipment, there are now few limitations for radiant heating and cooling systems. The past concern over condensation has now been successfully addressed, with tighter enclosures to mitigate moisture infiltration, dedicated outdoor air systems for dehumidification of incoming ventilation air, and current technologies capable of integrating and regulating all control points within radiant based HVAC systems. Additionally, radiant systems technically offer the best coupling with solar and geothermal energy systems, heat actuated cooling systems and heat pumps by enabling such systems to operate at higher efficiencies and coefficients of performance. They can also be applied using district energy and CHP principles to stand alone multi-story buildings. Furthermore, high performing buildings particularly those with above normal sensible loads due in part to the use of electronic equipment are much better served with radiant absorption of long wave energy rather than increase recirculation of chilled air which in comparison, is neither comfortable nor efficient.

2. ARCHITECTURE OF RADIANT CEILINGS

Ceiling radiant cooling panels (CRCP) and heating panels are generally built as an architectural finish product (with necessary acoustical qualities, color, and pattern), compatible with the traditional drop ceiling "tee grid" system or as a free hanging element. Typically, a copper tube is embedded into an extruded aluminum saddle which is permanently affixed to the back of an architectural metal ceiling panel or is part of a built-up panel made from linear extrusions with integral tube saddles fastened together to make different width panels. The process of how the copper coil is thermally bonded with the radiant ceiling panel is crucial especially in cooling application. Panel piping arrangements are generally in a serpentine pattern; however, parallel header arrangements are also available. Typical panel construction is illustrated in Figure 1. As installed, the "drop in" radiant panels weigh 1.6 to 2 lb/ft^2. A radiant ceiling panel also has an acoustic value. The acoustic signature can be achieved with a variety of perforations. In addition, a glass fiber blanket or non-woven sound-attenuating sheet is placed on the pack of the panel.

The lightweight construction results in a transient response "time constant" of only about 3 to 5 minutes from ambient room temperature to operating temperature once the cooling/heating fluid is applied. That means they respond rapidly to changing space sensible load conditions. Hydraulically, the ceiling panels are most frequently connected with flexible-push on coupling hoses for fast and safe installations, as illustrated in Figures 2 and 3.

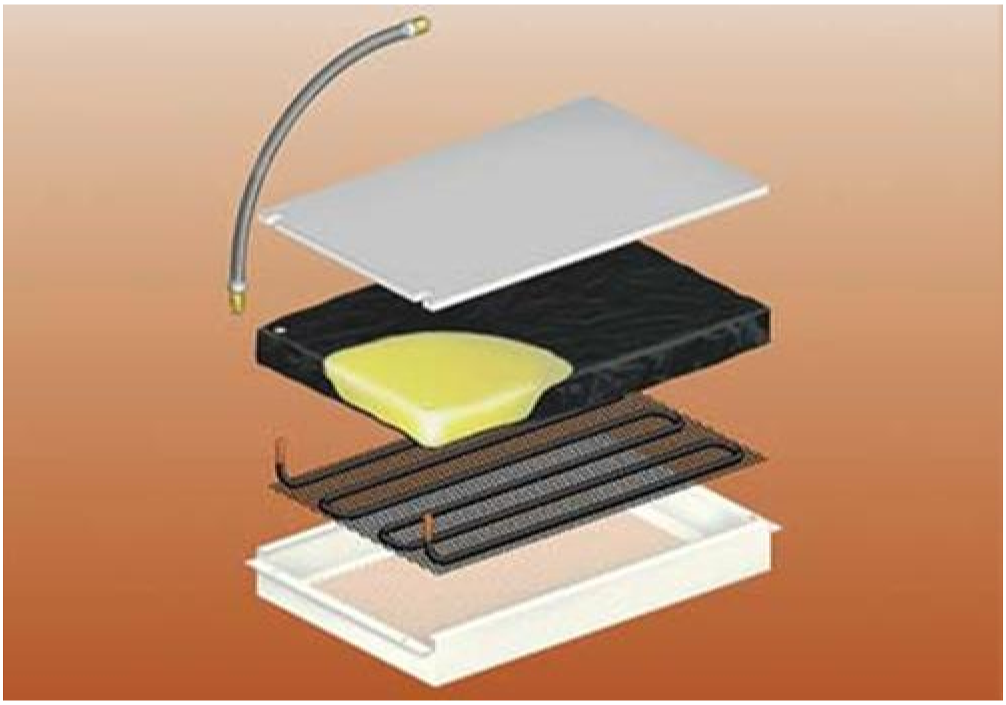

Figure 1. Typical Composition of Radiant Modular or Pan-Type Ceiling Panel

Panels can be moved aside without disconnecting the hoses, for easy access above. They can also be easily removed and reconnected for either extensive maintenance or evolving space use requirements without breaking normal threaded or sweat solder plumbing connections.

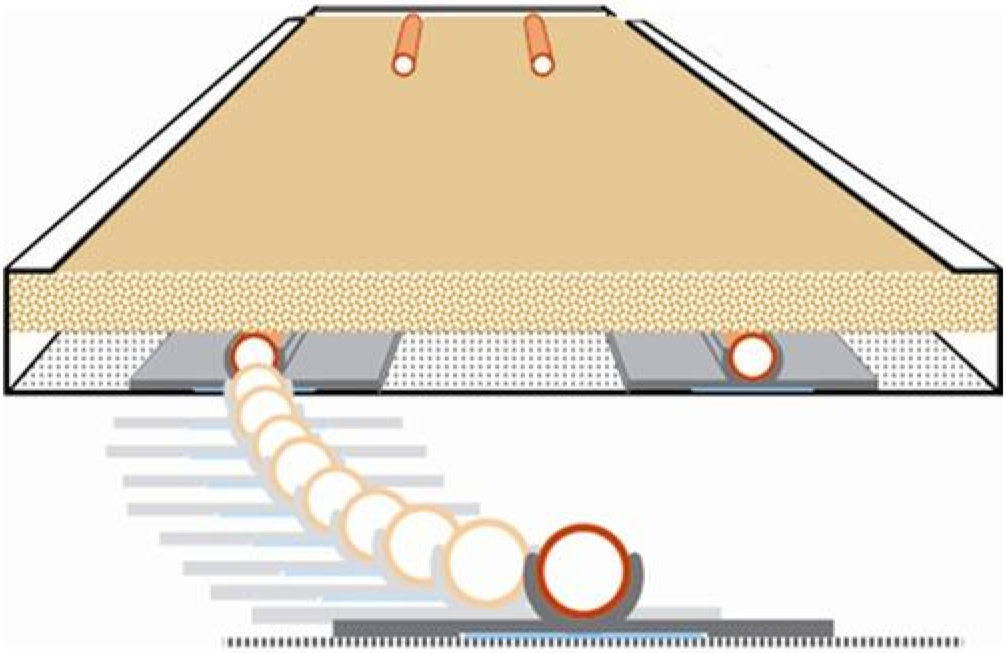

Figure 2. Cutaway View of Typical Modular Radiant Ceiling Panel

Figure 3. Back View of Drop Ceiling: Piping Configuration with Flexible Hose and Quick-Connect Fittings

Radiant ceiling panels promote architectural freedom in several ways. First, radiant cooling ceilings can be designed to be visually indistinguishable from regular ceilings to maintain aesthetic appeal. Secondly, the hardware in a radiant cooling system is smaller and more flexible. The size of the ceiling panels, the arrangement of the appliances within the ceiling, and the partitioning of the room all are flexible. Units may also be quite large assembled from linear extrusions which maybe be as long as 16 ft and widths (built of several extrusions side by side) to 24 in. or even to 48 in. wide. These are also the typical style of panels used along building perimeters for heating applications. In perimeter heating, panels are typically 6 to 24 in. wide depending on the perimeter heat loss. Due to the reduction in sensible heating and cooling loads that must be carried in the air, the new minimal airflow requirements often permit designs where common mechanical air distribution devices such as diffusers and ductwork are made much smaller or are eliminated altogether.

3. DESIGN AND DIMENSIONING

Panels are dimensioned based on building/room loads for heating or cooling and sometimes both. For heating or cooling the room design temperature and supply water temperature and flow rates determine the capacity of the panels in Btu per square feet. Once this capacity is determined it is divided into the load to determine the square feet of active panels required. The design then uses this information to layout the panels in the space. Heating panels should be placed within 3 ft of the exterior walls. Cooling panels should be more evenly distributed throughout the space with some heavier weighting to the highest area of heat gains (i.e., windows.)

COOLING

In practice, the design cooling capacity per unit panel area (Btu/h · ft^2) is determined from the panel manufacturer's catalog data. The unit panel cooling capacity can be selected from the design capacity tables provided by the panel manufacturer based on the difference between the room temperature and the mean panel surface temperature (or mean water temperature [MWT]).

HEATING

In many climates, radiant cooling ceilings can also provide heating. For best applications, the structure should be well insulated, and outside temperatures should not be extremely low. In these cases, radiators are not needed, saving costs and making more floor space available.

Because humans sense heat from a hot ceiling more quickly than from hot air blown in from ducts, surface temperatures should not exceed 95°F. The reason the surface should not exceed 95°F is to permit humans to radiate a small amount of heat from their heads that are normally at 99°F to a slightly cooler surface. If the surface temperature exceeds 95°F, humans in the space will experience discomfort and may even experience headaches. This is only if the panel is right above the head sufficient separation should allow higher temperatures of 110 to 120°F max for normal ceiling heights of 9 to 11 ft or even higher temperatures if placed at the perimeter walls where panels are not directly over the heads of occupants.

Radiant panels radiate heat to the surrounding surfaces including people, furnishings, and the cold interior surface of the window. The cold window surface, loses heat via transmission to the cold outside. Without radiant heating, the cold window surface cools the air passing over it, creating a significant convection down draft. Normally the baseboard radiators balance this action. With radiant heating, the interior surface of the glass is warmed through direct radiation, significantly reducing the convection down draft produced. The small down draft current produced, typically less than 20 to 30 fpm, creates a vacuum effect under the radiant heat panels. The low-pressure area formed draws the warm room air at the ceiling towards this area where it is further warmed by the radiant heat panels. Thus, the air becomes even more buoyant, and it has a tendency not to convect down the outer wall. The net effect is a surface boundary layer of "dead air" created very near the window. The down draft is below the minimum threshold felt by humans. Therefore, in some cases the baseboard radiators can often be eliminated, reducing capital costs and increasing usable floor area.

Another consideration in the heating mode, the convection is lower, and the ceiling's capacity is proportionally reduced. In most cases, the internal heat loads can provide sufficient heat. To prevent cold airdrop from the windows, the radiant heating portion of the ceiling needs to be installed along the perimeter of the room (typically within 3 ft of the exterior wall).

4. DESIGN ASPECTS OF RADIANT CEILING SYSTEMS

Technically speaking, a radiant cooling ceiling is simply a large heat exchanger suspended from a room's ceiling. It exchanges energy with the room via radiation and convection. Accordingly, cooling ceilings can be rated based on the temperature difference between the panel's MWT and the room design temperature. Heat transfer from the room to the ceiling surface (or the other way around in the case of heating) is a function of the average ceiling surface temperature and the room temperature. The objective of every radiant cooling ceiling developer is to get the ceiling surface temperature as close as possible to the water temperature. The smaller the temperature difference between chilled water and ceiling surface, the more efficient the system.

The overall heat exchange between ceiling panels and the flow in the water piping obeys the following equation:

$$Q = kA\,\Delta T \qquad \textbf{(1)}$$

where ΔT is the smallest possible temperature difference between the contact point of the ceiling panel and the fluid flowing in the piping. This path includes the conductance between the panel and the pipe, conductance through the pipe, and the convection from the inner pipe surface to the fluid. Thus, k is the equivalent thermal conductivity. The removed heat by the fluid flow in the piping Q should be as large as possible. Therefore, the overall k-value and the heat conducting areas A must be made as large as possible.

Panel dimensions can be chosen freely within manufacturing and building installation constraints. The only definitive design constraint to the architect is that active tubing attachment side must be flat.

5. ACOUSTIC FEATURE OF RADIANT CEILING PANELS

Room acoustics also are handled conventionally. The following addresses commonly applied strategies for improving acoustics in rooms with cooling ceilings.

ACOUSTIC INLAY MATS

Mineral fiber mats 1.18 in. thick and 2.5 lb/ft^3 are applied on the back of a perforated radiant ceiling tile to meet required acoustic values.

ACOUSTIC FLEECE

In environments such as clean rooms and hospitals where fiber rub-off is not expected under any circumstances, acoustic fleece may be used. The black fleece is bonded to the rear of the perforated ceiling panel. To prevent a decrease in the heat exchange between the heat conducting rails and the ceiling panel, the fleece is bonded between the heat-conducting rails. Acoustic fleece must have a minimum plenum height of 12 in. to be effective.

PANEL PERFORATION

The ceiling panels might feature certain perforation pattern to enhance acoustic performance or personalize the visual aspect of the ceiling. The perforation pattern is often specified with hole diameter and free area (open cross sections etc.).

6. CONTROLS

The design of control system should take into account the building, its intended use and the effective functioning of the heating/cooling system, efficient use of energy and avoiding heating/cooling the building to full design conditions when not required. This should include keeping distribution heat losses as low as possible, e.g. reducing flow temperature when normal comfort temperature level is not required. Control and operation of the system help to handle the conditioning systems with savings of operational costs and enable the maintenance of required indoor environmental conditions.

Hydronic radiant controls systems have two primary functions: controlling room temperature and preventing condensation on the ceiling surface. For proper operation and maximum energy savings, radiant ceiling systems require the use of precision electronic or direct digital controls.

TWO-PORT CONTROL VALVES

The two-port valve controls the heat loads by permitting more or less chilled water to flow through the valve and through the cooling ceiling (Figures 4 and 5). The supply water temperature stays constant. A humidity sensor closes the valve as soon as the chilled water supply temperature reaches dew point.

This type of control is very affordable and simple. The only disadvantage is dew point control. The cooling ceiling must be turned off as soon as a risk of condensation occurs.

Controls are generally applied to influence both pressure (flow) and temperature.

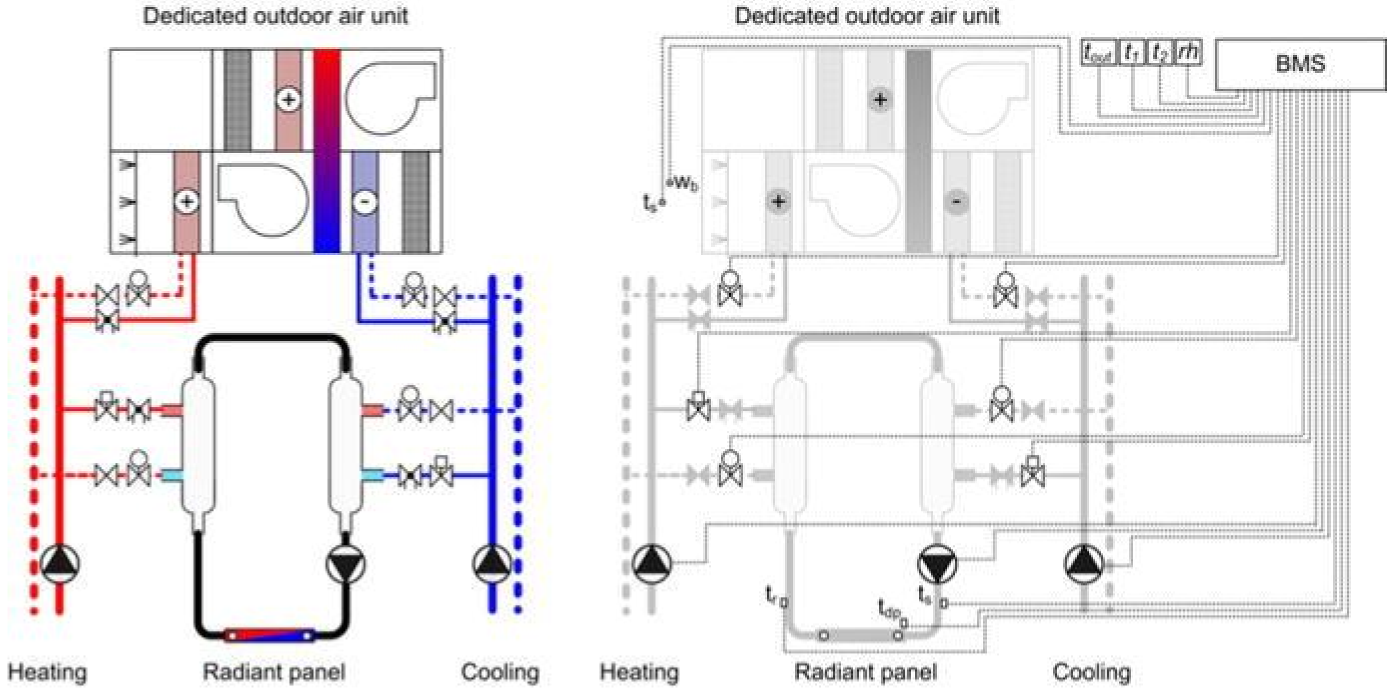

Figure 4. Typical Control Schematic for Radiant System with Injection Control Valves in Four-Pipe/Two-Pipe System

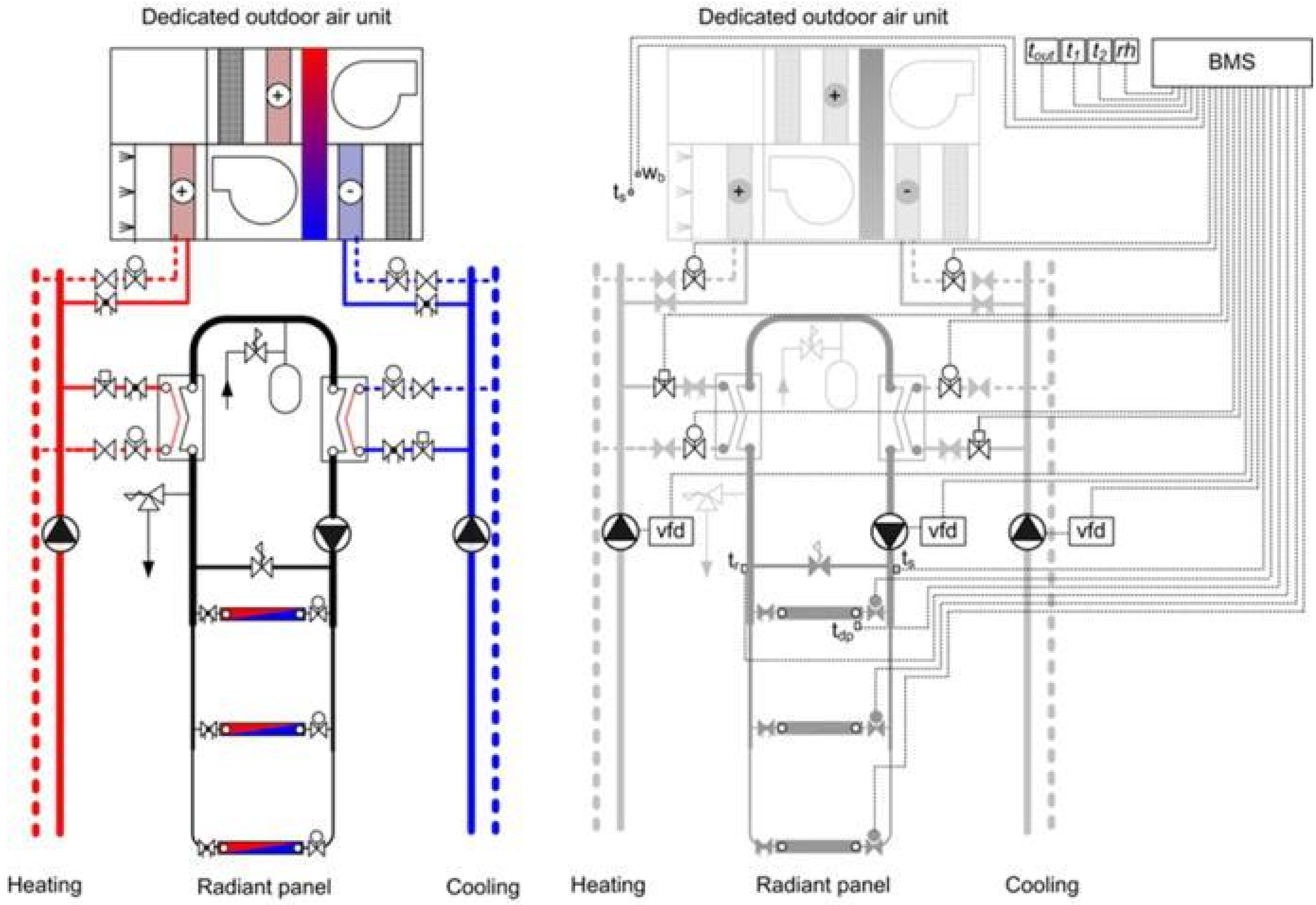

Figure 5. Advanced Control System for Radiant System with Heat Exchangers in Four-Pipe/Two-Pipe System (Some Items Removed for Clarity)

Accordingly, there are two control strategies: using (1) a two-way control valve to control water flow or (2) an injection circuit to control water temperature.

CONTROLLING WATER TEMPERATURE/INJECTION CIRCUIT

Using an injection circuit requires a circulating pump to provide a constant flow of water through the ceiling (Figures 6, 7, and 8). Depending on the heat loads, a two- or three-way valve injects more or less chilled water to the ceiling supply. The same water quantity is sent to the system return. The supply water temperature for the ceiling is controlled by the quantity of the water injected. If risk of condensation occurs, the temperature of the supply water can be raised. The ceiling loses some of its capacity but can be kept in operation.

The injection circuit ensures both effective operation and the maximum possible mean temperature difference for cooling ceilings. As each control zone needs a pump, a humidity sensor, and a controller (which constantly calculates dew point and compares it with the supply water temperature), this type of control is more expensive than the two-way control scheme.

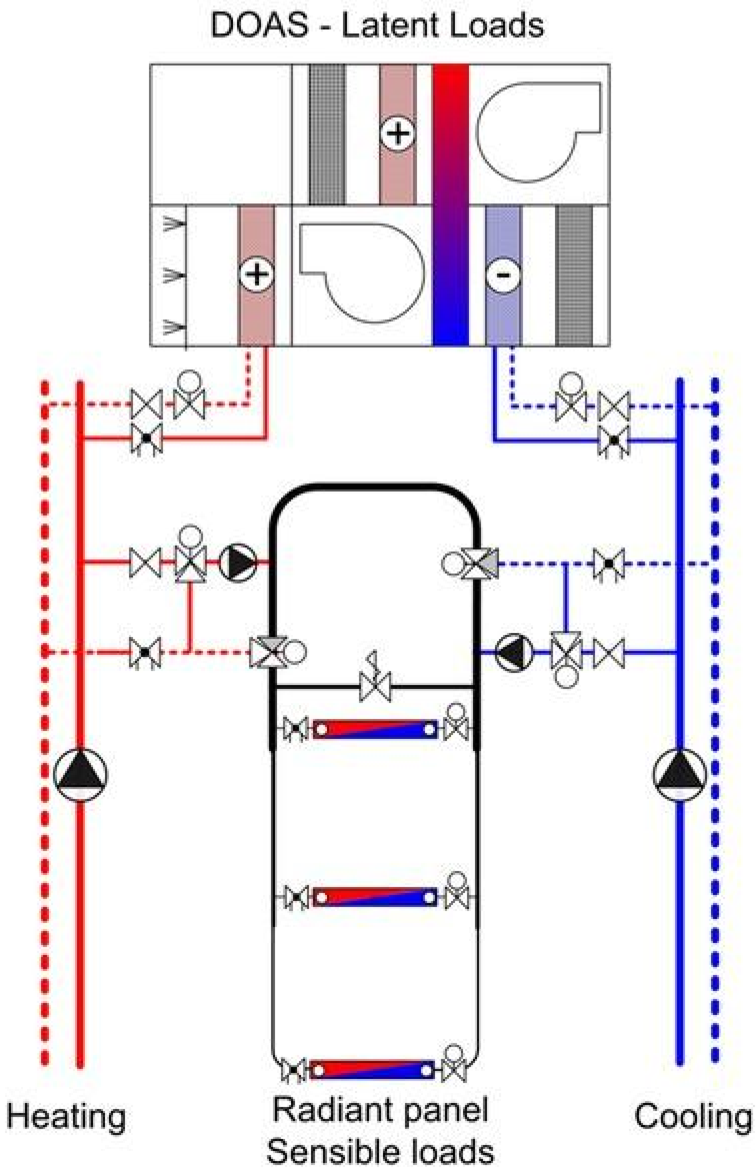

Figure 6. Secondary Pumps with Mixing/Injection Control Valves on Four-Pipe/Two-Pipe System

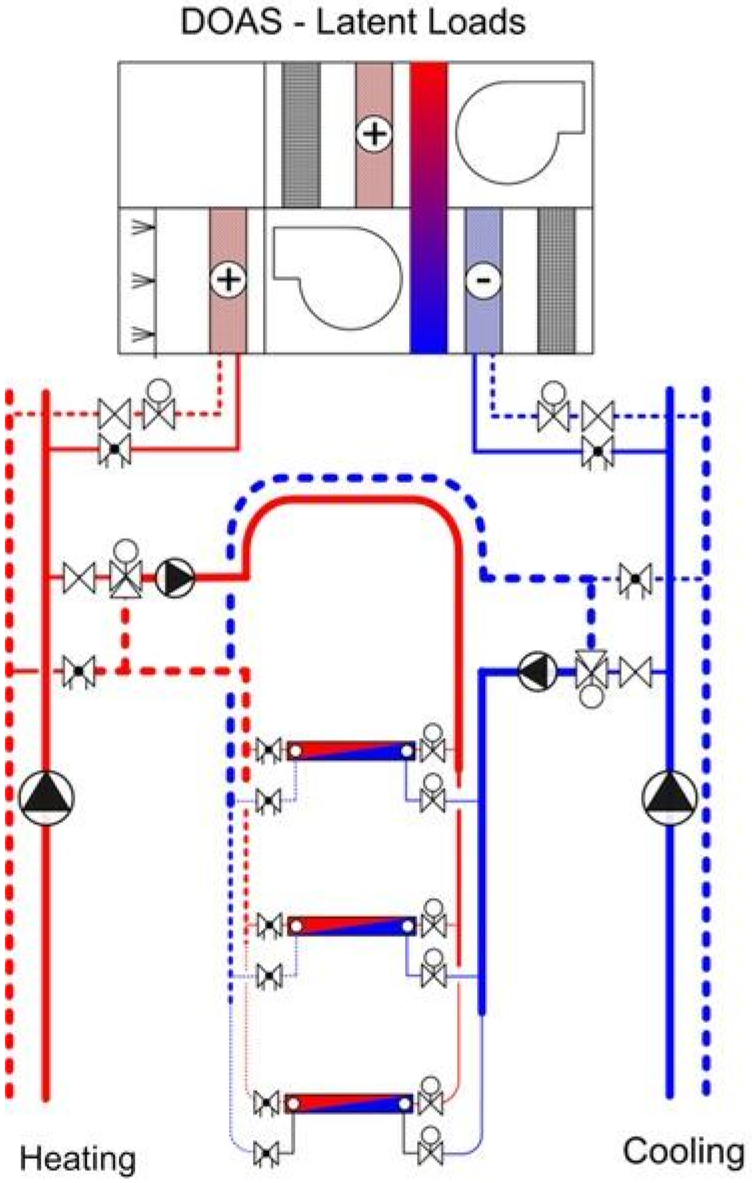

Figure 7. Secondary Pumps with Mixing/Injection Control Valves on Four-Pipe System

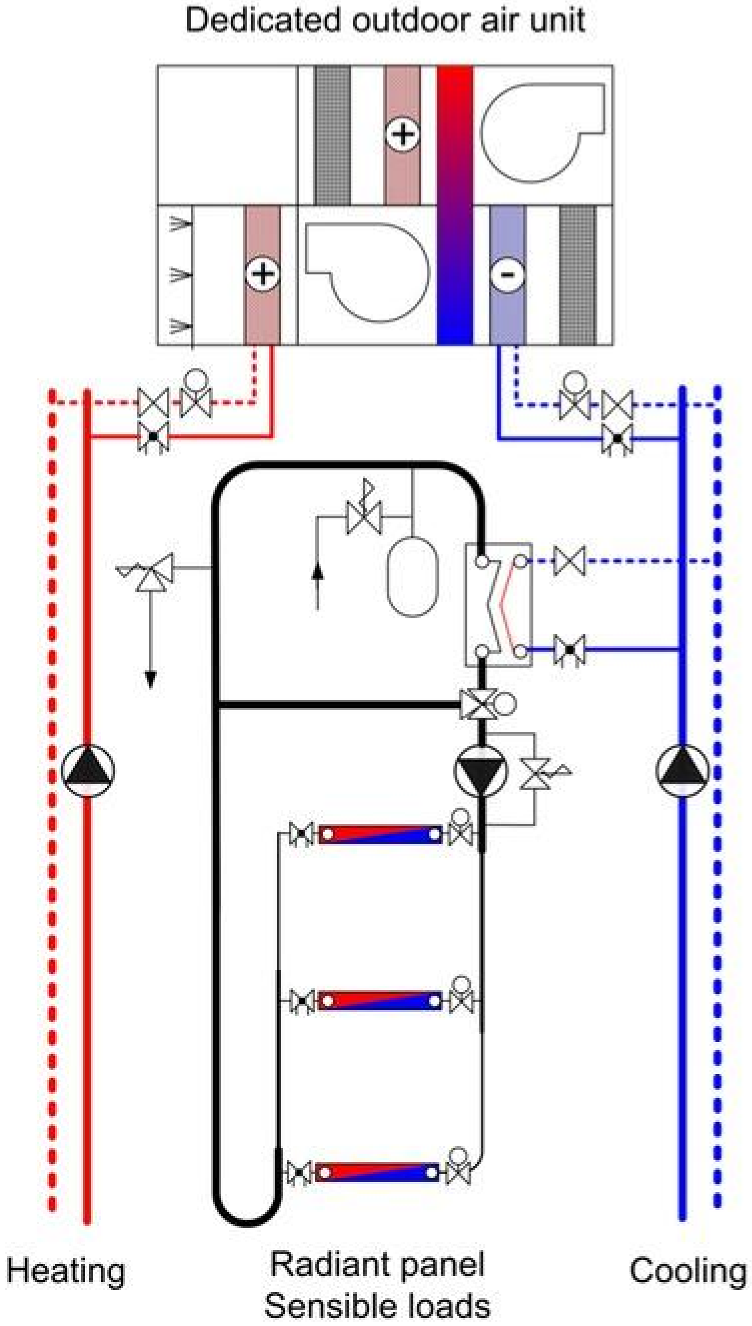

Figure 8. Two-Pipe Cooling-Only System with Heat Exchanger

ENERGY SAVINGS WITH RADIANT COOLING CEILING SYSTEMS

Buildings with radiant ceiling cooling systems, Current systems almost always require 100% outdoor air systems and tight building envelopes to manage humidity. Energy saving are realized by significant reductions in air moving power (only the outdoor make-up air is distributed to the building) and the higher evaporator temperature of the chiller supplying cool water to the chilled ceiling panels.

7. DESIGN EXAMPLES

CLASSROOM

Load calculations require that the space sensible load be 28,584 Btu/h. The space has 30 occupants.

A 30 by 30 by 10 ft classroom and a maximum occupancy of 30 people, is to be maintained at 74°F and 50% rh. From the psychometric chart, this gives a dew point temperature of 55°F and a moisture content of 0.0081 $lb_{moisture}/lb_{dry\ air}$.

Step 1. Determine the sensible and latent hourly heat gain for the room.

The sensible and latent hourly heat gains are found using accepted procedures found in the ASHRAE handbooks. For this example, assume sensible hourly heat gain = 28,584 Btu/h.

Step 2. Determine the mean water temperature required for cooling. Supply water temperature = 55°F

Assuming a temperature rise of 8°F, add half of this temperature rise to the inlet water temperature, giving 59°F.

Step 3. Determine the minimum air supply required for the room. According to ASHRAE tabulated data, the recommended air supply per person is

$$20 \text{ cfm per person} \times 30 \text{ persons} = 600 \text{ cfm}$$

Step 4. Determine the latent load capacity of the air:

$$\text{NP} \times \text{OCPL} \times 4840 \times (\text{HRODA}_2 - \text{HRIDA}) = 5631 \text{ Btu/H}$$

Step Determine the sensible cooling capacity of the primary air:

$$V \times 1.1(\text{DBIDA} - \text{DBSUP}) = 13{,}893 \text{ Btu/h}$$

where

N	=	number of occupants in the space
OCP	=	latent heat produced by the occupants
HRODA_2	=	humidity ratio of space air
HRID	=	humidity ratio of space operating design condition
V	=	volumetric flow rate of the supply air
DBID	=	dry bulb temperature of space operating design condition
DBSU	=	dry bulb temperature of supply air

30 occupants at 220 Btu/pp = 6600 Btu/pp and the moisture gain results in

$$\Delta\omega = q_L/(60 \times \rho h_{fg} \times \text{Airflow})$$

The moisture content of the space operating condition is 0.0081 + 0.0022 = 0.0103 lb/lb. From the psychometric chart, the dew point of the space operating temperature is 58°F.

The space condition is calculated by including the sensible and latent loads to the space. From this condition the dew-point temperature is shown on the psychometric chart. The panel surface operating temperature is selected about 3°F higher than this temperature.

Step 5. Determine the sensible load capacity of the air with the following equation:

$$Q_s = QpC_p(t_{room} - t_{supply})60$$

$$Q_s = (600 \text{ cfm})(0.075 \text{ lb/ft}^3)(0.24 \text{ Btu/lb}\cdot°\text{F})(74 - 55)60 = 12{,}312 \text{ Btu/h}$$

The sensible cooling required from the panels is 28,584 – 12,312 = 16,272 Btu/h.

Step 6. Select a panel surface temperature at least 3°F higher than the space operating dew-point temperature = 58 + 3 = 61°F Thus, the required temperature is higher than the MWT calculated in step 2.

Step 7. From the 61°F panel operating temperature or panel mean water temperature, derive the supply water temperature and water temperatures to and from the panels. Typically, the difference supply and return water temperature is 8°F, so the panel water supply temperature in this case is 61 – 4 = 57°F and the panel return water temperature is 61 + 4 = 65°F.

Step 8. Figure 9 shows the panel output.

Step 9. The required sensible cooling output from the panel is 16,272 Btu/h at 61°F MWT and therefore the required panel area is 16,272/26 = 625 ft^2. The classroom has a ceiling area of 30 × 30 = 900 ft^2. The ratio of ceiling panel to ceiling is 69%.

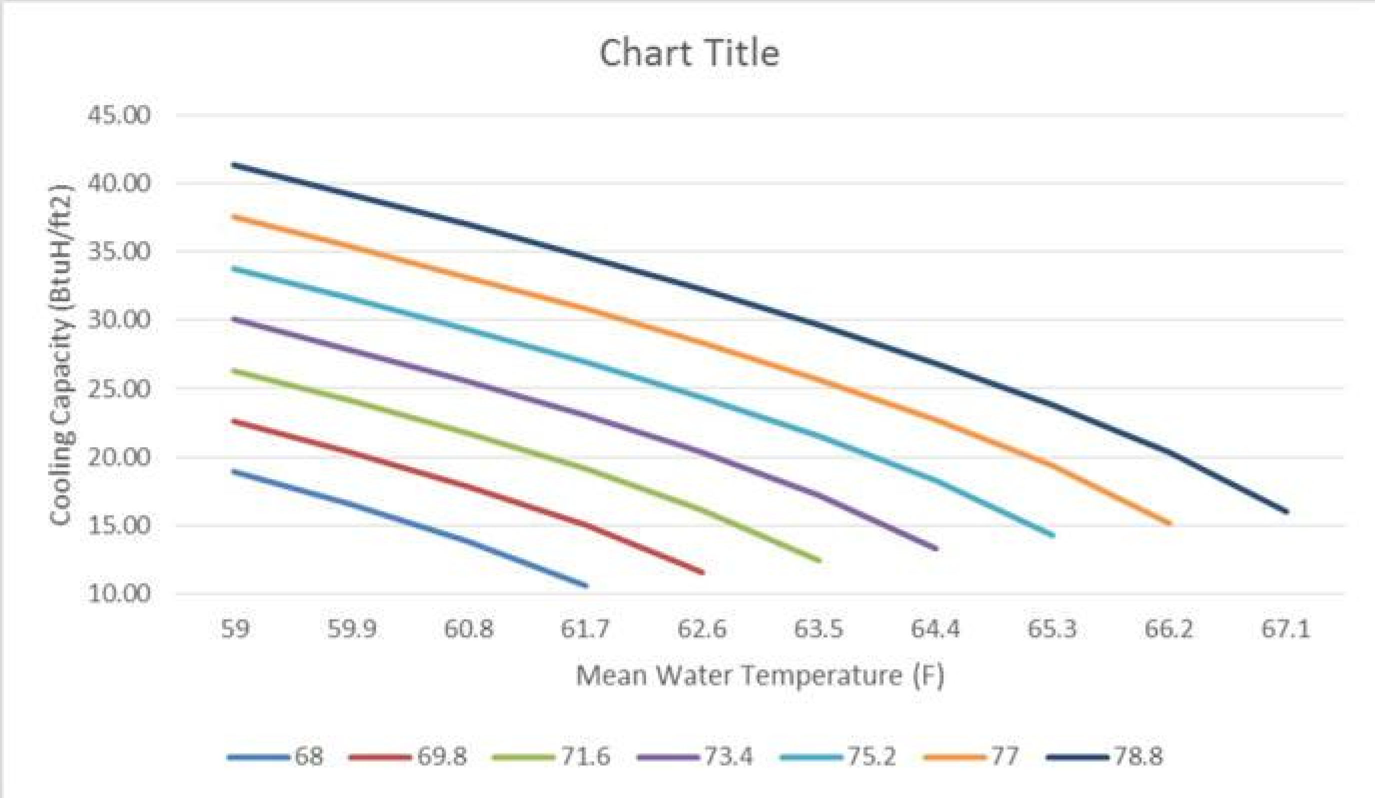

Figure 9. Panel Output for Classroom Example: 30 Btu/h·ft^2 at Room Temperature of 75.2°F and MWT of 59°F, and 26 Btu/h·ft^2 at Updated MWT of 61°F

OFFICE

The office has a sensible cooling load of 6824 Btu/h, which is obtained from the load calculations. There will be two occupants in the office.

A 13 × 16.5 × 8 ft interior office with a 2 × 4 ft T-bar ceiling and a maximum occupancy of two people is to be maintained at 75°F and 45% rh. From the psychometric chart, the dew-point temperature is 55°F and moisture content is 0.009 lb/lb.

Step 1. Determine the portion of the load to be provided by the radiant ceiling and the portion of the load to be provided by the ventilation system. As a rule of thumb, the area of radiant panels is 70% of the ceiling area of the space. For this application, the preliminary output from the radiant ceiling is 13 × 16.5 × 0.7 × 30 Btu/ft^2 (this is a preliminary output and will be reused later) = 4504 Btu/h.

Subtract the output of the radiant ceiling from the required sensible cooling load: 6824 – 4504 =2320 Btu/h.

The required volume of supply air to meet the rest of the load can be calculated as

$$Q_s = \text{Airflow} \times 1.085\,\Delta T$$

$$Q_s = 2320/1.085 \times (74 - 55) = 113 \text{ cfm}$$

Determine the sensible load capacity of the air:

$$Q_s = QpC_p(t_{room} - t_{supply})60$$

$$Q_s = 113 \text{ cfm} \times 0.075 \text{ lb/ft}^3 \times 0.24 \text{ Btu/lb·°F}(75 - 55) \times 60 = 2{,}440 \text{ Btu/h}$$

The sensible cooling required from the panels is 6824 – 2,440 = 4,383 Btu/h.

Step 2. Determine the latent load capacity of the air:

$$q_L = Qph_{fg}(\omega_{room} - \omega_{supply}) \times 60$$

$$\text{Two occupants at 250 Btu/person} = 500 \text{ Btu/h}$$

$$\text{Moisture gain} = 500/4840 \times 113 = 0.00091 \text{ lb/lb}$$

The moisture content of the space operating condition is 0.009 + 0.00091 = 0.00991 lb/lb. The dew point of the space operating temperature from the psychometric chart is 57°F.

Step 3. Determine the MWT required for cooling. Supply water temperature = 55°F. Assuming a temperature rise of 8°F, add half of this temperature rise to the inlet water temperature:

$$\text{MWT} = 55°\text{F} + 8/2 = 59°\text{F}$$

The dew-point temperature of the space operating condition is shown on the psychometric chart in Figure 10. The operating surface temperature of the ceiling panels is kept about 3°F higher than the space dew-point temperature.

Step 4. Select a panel surface temperature at least 2°F higher than the operating dew temperature, which is 57°F + 2°F = 59°F.

Step 5. From the 59°F panel operating temperature or panel MWT, derive the supply water temperature and water temperatures to and from the panels. Typically, the difference between supply and return water temperature is 8°F, so the panel water supply temperature in this case is 59°F – 4 = 55°F and the panel return water temperature is 59°F + 4 = 63°F.

Step 6. Figure 11 shows the panel output, the flow rate, and the cooling output in comparison with the air temperature MWT.

Step 7. The required sensible cooling output from the panel is 4383 Btu/h; therefore, the required panel area is 4383/30 = 146 ft^2. The office has a ceiling area of 13 × 16.5 ft = 215 ft^2. The ratio of ceiling panel to ceiling is 68%.

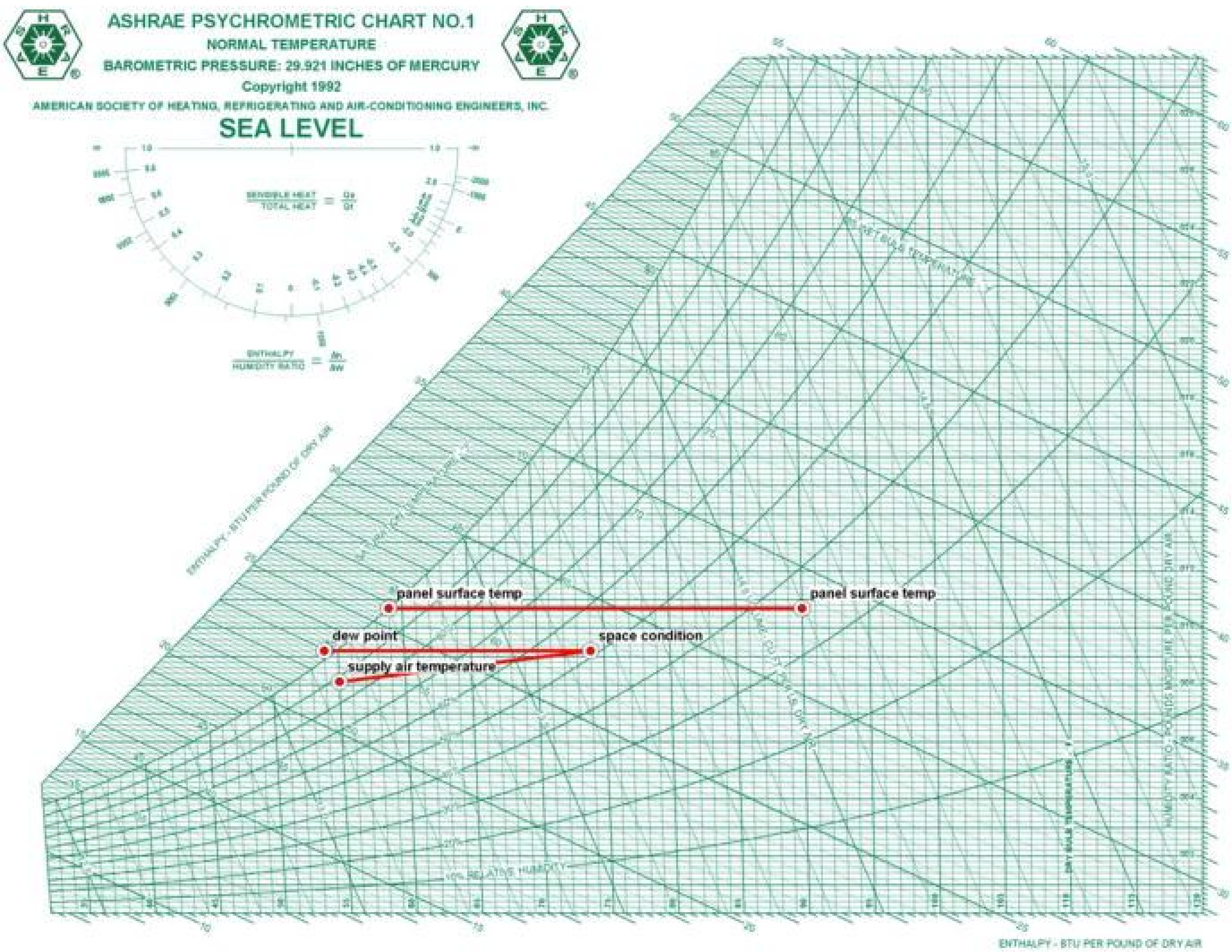

Figure 10. Dew Point of Space Based on Operating Temperatures

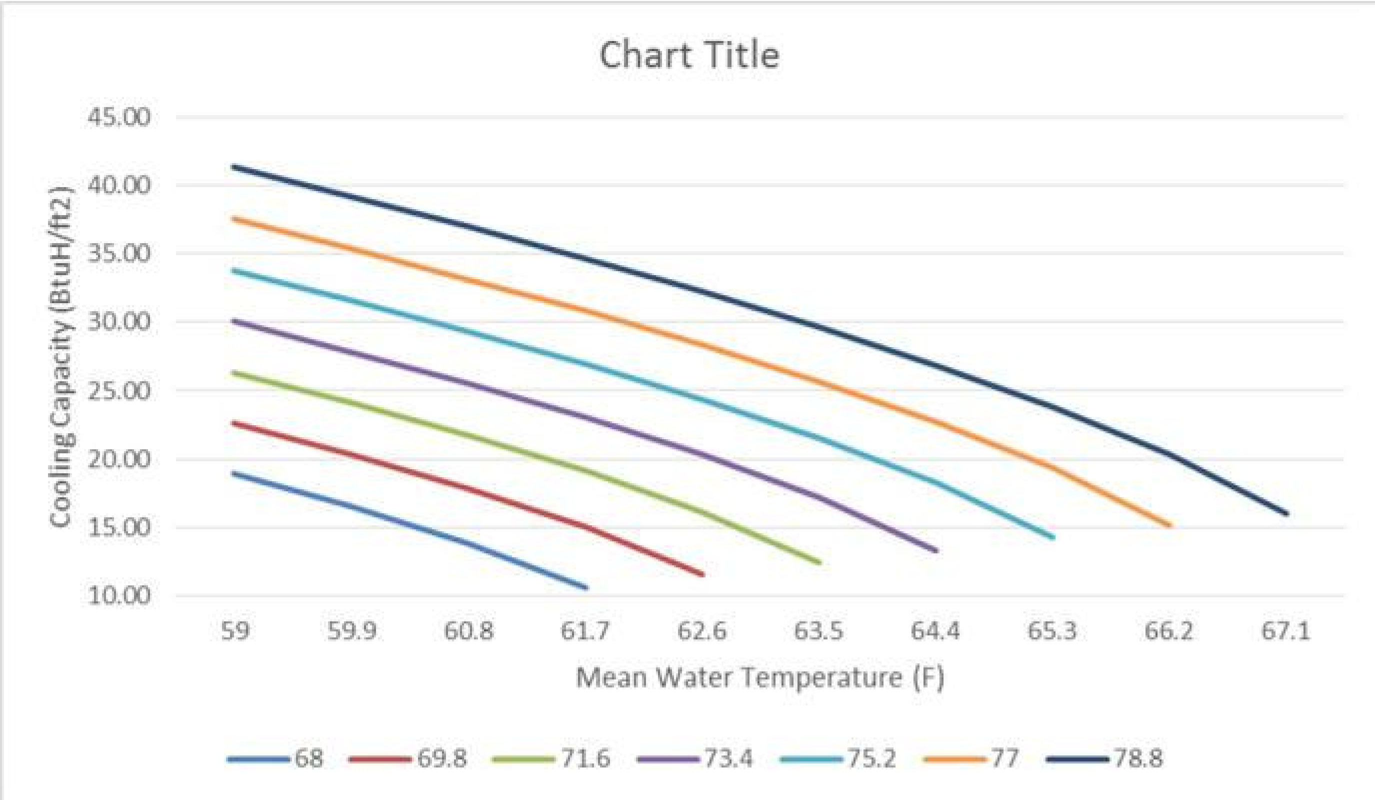

Figure 11. Panel Output for Office Example: 30 Btu/h·ft^2 at Room Temperature of 75.2°F and MWT of 59°F, and 18 Btu/h·ft^2 at Updated MWT of 61°F

8. CONDENSATION CONTROL

When the dew-point temperature of the space air has been determined, the surface temperature of the radiant ceiling can be controlled to be above the dew point and therefore avoiding the risk of condensation. Monitoring the space air temperature and the space humidity levels will provide the space moisture content. In simple terms the supply water temperature to the panels must be controlled to avoid the possibility of condensation.

The only possibility of condensation occurring is when radiant cooled ceilings are used in a space with operable windows. From practice it is known that operable windows can induce up to 6 ACH through a space.

From the previous example the dew point temperature of the room condition is 57°F 13.8°C, so if this condition were to come into contact with a radiant panel with a surface temperature of 56°F 13.3°C and condensation would occur.

A very simple control methodology to avoid condensation is to elevate the panel surface temperature to roughly 2 to 3°F above the space air dew point temperature of 57°F, which would give a panel surface temperature of 60°F.

PRIMARY AIR CONDITIONING

Control of the building humidity level is paramount when panels systems are used. Space ventilation and humidity control are solely provided by the supply air delivered from the air handling unit. The minimum supply airflow rate should be sufficient to provide both functions. The indoor dew point is determined by the latent load to the space and the condition of the supply air. To avoid the risk of condensate formation on the panels, the AHU must condition the primary air to be sufficiently dry so as to absorb the moisture generated in or infiltrating into the building.

Commissioning of the AHU control system must ensure that the supply air conditions are achieved without significant oscillations. A range of air conditioning technologies may be applied provided that the supply dry bulb and dew point temperature are both precisely controlled.

CONDENSATION PREVENTION

The implementation of an adequate chilled-water temperature control system, together with the supply of an adequate amount of correctly conditioned primary air, is sufficient to avoid the occurrence of condensation on the panels.

Water supply to panels should not be activated when space dew point temperatures are above the zone chilled-water supply temperature. It must be ensured that the panels system chilled-water supply is shut off any time the air handler is not in operation and only restored when the space dew-point temperature is safe for non-condensing panels operation. In the transition from spring to summer, or from summer to winter, the outdoor air condition may result in a higher dew point and therefore the system can remain in operation.

In practice, a moisture sensor on the supply pipe work, or a dew point calculation warns of the possibility of condensation. The formation, and subsequent falling, of water droplets on the panel surface lags the onset of conditions that could cause condensation. It is common for panels to operate with chilled-water temperatures below the zone dew point without significant moisture collection on the panels surface with condensation forming first on the uninsulated surface of the chilled-water pipework feeding the panel.

Condensation prevention can be implemented with either reactive or proactive strategies, or a combination of both. In a proactive strategy, the control system acts to avoid or prevent the formation of condensate. In a reactive strategy, the control system acts in response to condensate that has formed.

PROACTIVE STRATEGIES

In a proactive strategy, the dew point can be monitored via contact humidistats by attaching the sensor to an uninsulated portion of the piping, immediately prior to the supply connection to the panels' panel (Figure 12).

The dew point temperature of the space air is determined by continuous sensing of the space air temperature, and relative humidity. The calculated dew point must then be compared to the supply water temperature.

Figure 12. Surface Condensation Sensor

In response to the sensor feedback, chilled water supply temperature set point can be reset above the room dew point, or the supply water to the zone can be halted.

Figure 13 presents a control strategy where the chilled-water supply temperature is varied in accordance with the space dew-point temperature. In this case the panel's chilled-water temperature control system maintains a minimum differential between the space dew-point temperature and its chilled-water supply. To maintain this differential, the dew-point temperature must be monitored or calculated. This can be done by monitoring the humidity either in the return air duct or directly in the zone. This strategy allows the panel within the panels to continue to contribute to the space sensible cooling even during periods of elevated space moisture levels.

Figure 14 shows a strategy whereby the panel's chilled-water flow is interrupted when the measured space dew-point temperature rises above the panel's chilled-water supply temperature.

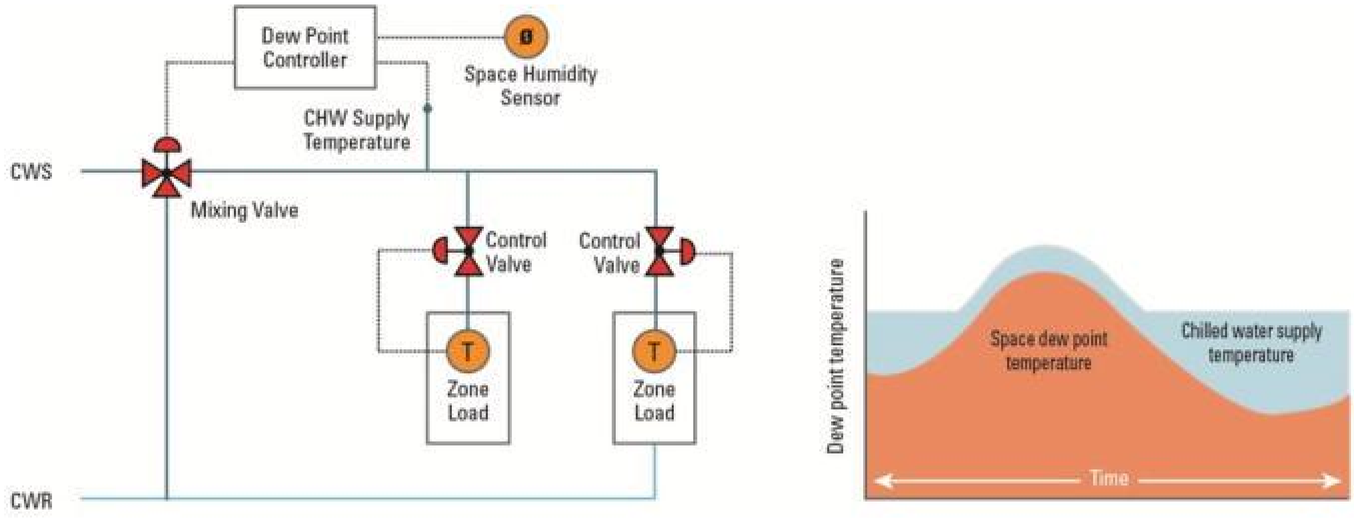

Figure 13. Condensation Prevention Strategy Involving Reset of Panel's Chilled-Water Supply Temperature

REACTIVE STRATEGIES

Condensation sensors such as those shown in Figure 12 used to detect moisture on the chilled-water supply pipe. When an indication of moisture is received from the sensor, the water supply is stopped, or its supply temperature is increased. This is a reactive method that can stand alone or form part of a total strategy.

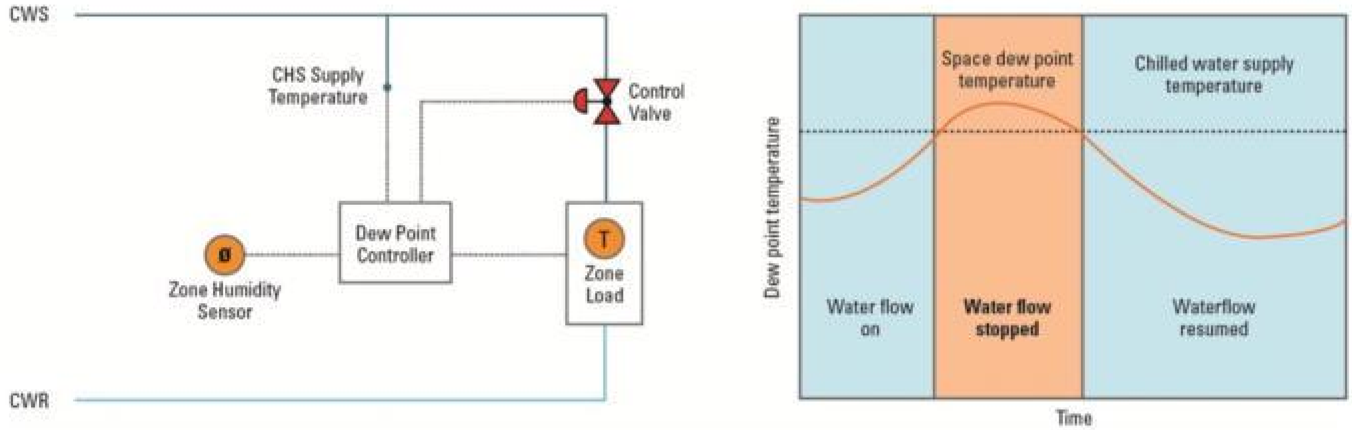

Figure 14. Condensation Prevention Strategy Where Water Flow Is Discontinued When Chilled-Water Temperature Is Below Space Dew-Point Temperature due to Rise in Humidity in Zone or Temperature Drop

Modern condensate sensors are very sensitive to the smallest amount of moisture forming. Condensation detected on the supply pipe work does not mean that the chilled panels will start to condensate and then drip; in fact, condensation can exist on the supply pipe for a very long period and still none present on the chilled panels. The protection provided by these devices must be weighed against the probability that more thermal complaint calls may arise when and if the chilled-water supply is discontinued during periods where an inconsequential amount of moisture is detected by the contact switch.

Figure 15 shows a reactive control strategy that relies on surface moisture sensors affixed to the chilled-water supply pipe. In the event condensation formation has been detected, the sensors override the space temperature sensor and close the chilled-water supply valve until the moisture has evaporated. If the flow of water is halted in response to a moisture sensor, then a control scheme will need to be developed to address when to turn the water system back on since the stoppage of water flow would typically halt the formation of moisture on the chilled-water supply pipe, even if the differential between the zone humidity level and the chilled-water supply temperature has not been restored.

SPACES WITH OPERABLE WINDOWS OR DOORS

For applications in spaces with operable windows or doors, occupants and staff should be educated on the effect opening them can have on their thermal environment. When windows or doors are opened, the supply of chilled water should be halted to avoid risk of condensation and/or prevent loss of cooling energy.

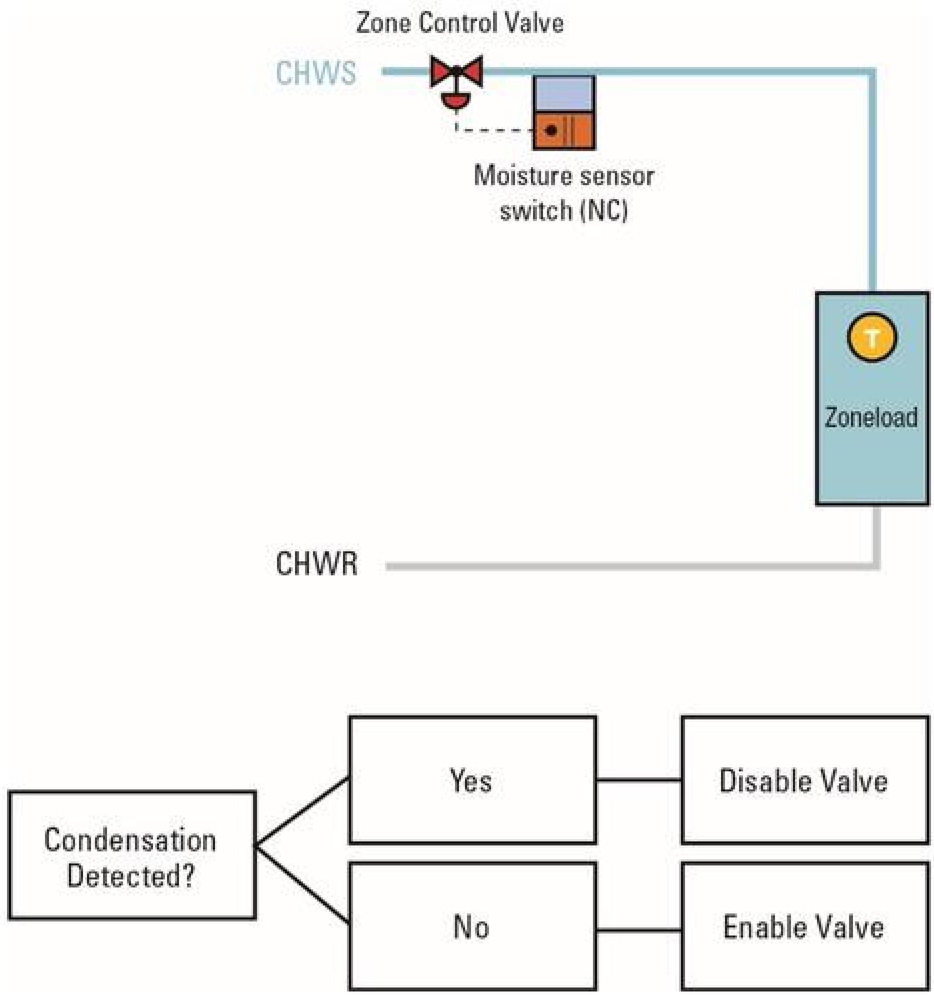

Figure 15. Control Strategy Where Chilled-Water Supply Halts When Moisture Is Detected on CHWS Pipe

Detection of a window being opened can be accomplished by the employment of window contact switches. Alternatively, moisture sensors such as that shown in Figure 12 may be used to discontinue the chilled-water flow during periods of condensation risk. When these are applied, one sensor should be installed on the chilled water supply pipe in each room (with operable windows or doors) the system serves. It must be remembered that the chilled water flow will not be restored until the sensor determines that all moisture has been evaporated from the surface of the pipe.

Figure 16 illustrates a proactive sequence where a sensor detects the opening of a window and interrupts the chilled-water flow to the space. A surface moisture sensor is also used to ensure that the space dew-point temperature is acceptable for restoring the chilled-water flow once the window is closed. Now the control system could boost the airflow to the space to reduce the time to restoring the chilled water.

9. EMBEDDED SYSTEMS

An embedded system is a surface cooling and heating system where water tubes as well as electric cables are integrated within the floor, wall, or ceiling. Radiant systems have been successfully used worldwide for heating and cooling of buildings. A radiant system provides a very good method of discharging high specific heating and cooling loads while maintaining thermal comfort at relatively low operation costs. The most popular use of radiant systems is radiant floor heating (Figure 17); however, ceiling as well as wall heating are also used (Figure 18).

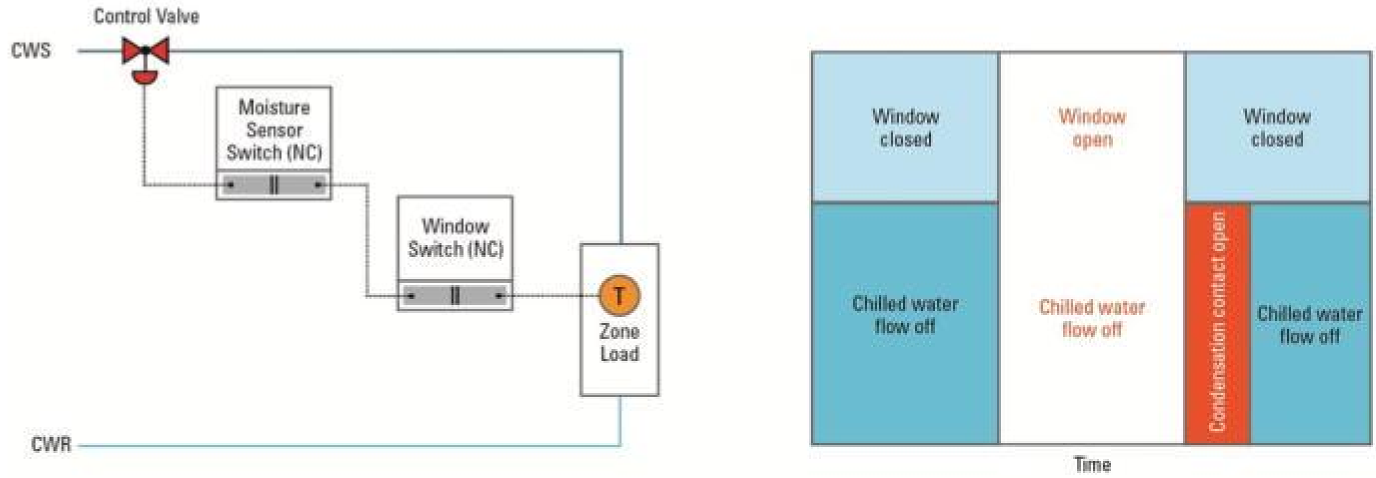

Figure 16. Condensation Prevention Strategy Involving Interruption of Water Flow After Window Opening

Radiant systems are dimensioned in accordance to the ensuing radiant heat exchange in the space. Radiant systems usually designed as a hydronic system therefore the amount of space necessary for the installation is considerably smaller than a conventional air conditioning system. Because of the low plenum height necessary to accommodate the installations, more architectural freedom is provided. The radiant system can be installed so that both individual and zone control can be achieved.

Many hospitals have radiant systems, and commercial buildings are starting to use the potential of radiant floors and ceilings. In Europe, several buildings such as the PGEM in Arnhem and the Groninger Museum have successfully used radiant systems to control the indoor environment. There has also been advancement in simulation programs that allow more detailed analysis of the indoor environment. With these advanced simulation tools and the individual elements necessary for the creation of a comfortable indoor climate using the predicted mean vote (Fanger 1972), radiant heat exchange can be studied. Because each individual surface temperature and its relationship (i.e., position to the other surfaces) can be determined a solution to the comfort balance equation can easily be found.

When incorporating a radiant system and a constant-volume ventilating system, the ventilation system may only be dimensioned to supply outdoor air for each person and to remove the latent as well as the material load if the radiant system is selected to remove the remaining cooling loads. Simmonds (1994) reported on some of these designs and how effective they are in providing an effective means of comfort climate.

Because cooling surfaces make no contribution to air renewal, they may always be operated in conjunction with a ventilating or air-conditioning plant, which also ensures the probably necessary dehumidification. However, also a combination with a natural ventilation system as well as operable windows may also be possible.

Radiant cooled floors have been successfully used on many projects over the past ten years and many ASHRAE and other peer-reviewed publications have been written on these projects. However, many questions still arise regarding the performance of radiant cooled floors when subjected to solar radiation. The convective cooling performance of a cooled floor has been reported by Olesen, Simmonds; Borressen (1994) reported on the solar absorption of a radiant cooled floor, this was further reported by Simmonds et al. (1996). Even with this research, many questions are still being raised regarding the performance of a radiant cooled floor when absorbing solar radiation. This section explains the performance of the floor, including some limitations of the presented calculation method, but also includes information on the controllability.

This section outlines the performance of a radiant cooled floor subjected to different solar radiation intensities and will look further into the controllability of the radiant cooled floor to maintain certain conditions such as space temperatures and floor surface temperatures. The influence of floor coverings on radiant cooled floors will also be discussed.

Previous papers by Simmonds (1994) have shown that radiant cooled floors are capable of removing 11 to 12.5 Btu/h·ft^2 from spaces. Borressen (1994) and Simmonds et al. (1996) have shown that radiant cooled floors are capable of removing up to 27.0 Btu/h·ft^2 of energy from a space 11.1 Btu/h·ft^2 by convection and 16 Btu/h·ft^2 by solar absorption. This section uses a simple steady-state equation to explain the performance of a radiant cooled floor when performing at its maximum capacity of both reducing the space air temperature and absorbing solar radiation that is reaching the floor. There are several dynamic simulation programs that accomplish the dynamic performance of a radiant floor. Many papers have been written on the performance of radiant floor for heating. MacCluer, Athienitis, and Simmonds. Olesen and Meirhans have reported on the performance of active concrete systems.

There has been advancement in simulation programs that have permitted a more detailed analysis of the indoor environment. With these advanced simulation tools, the individual elements necessary for the creation of a comfortable indoor climate using the predicted mean vote (PMV as determined by Fanger 1972), which has been adopted in ASHRAE *Standard* 55-2020 and radiant heat exchange can be studied. Because each individual surface temperature and its relationship (i.e., position to the other surfaces) can be determined a solution to the comfort equation can be found.

The PMV/PPD comfort equation, derived by Fanger and shown in ASHRAE *Standard* 55-2020, can be influenced by the control or balance of the radiant heat exchange in a space. When operating a radiant heated or cooled floor correctly, the surface temperature of the floor can be regulated. Absorbing a major portion of the solar radiation entering the space also prevents the floor from emitting thermal energy, in the form of long-wave radiation, back into the space and onto other surfaces.

Because cooling surfaces make no contribution to air renewal the surfaces usually operated in conjunction with a ventilating or airconditioning plant, which also ensures the necessary dehumidification is provided. The combination of a radiant cooled system with a natural ventilation system and operable windows may also be possible in certain climate conditions.

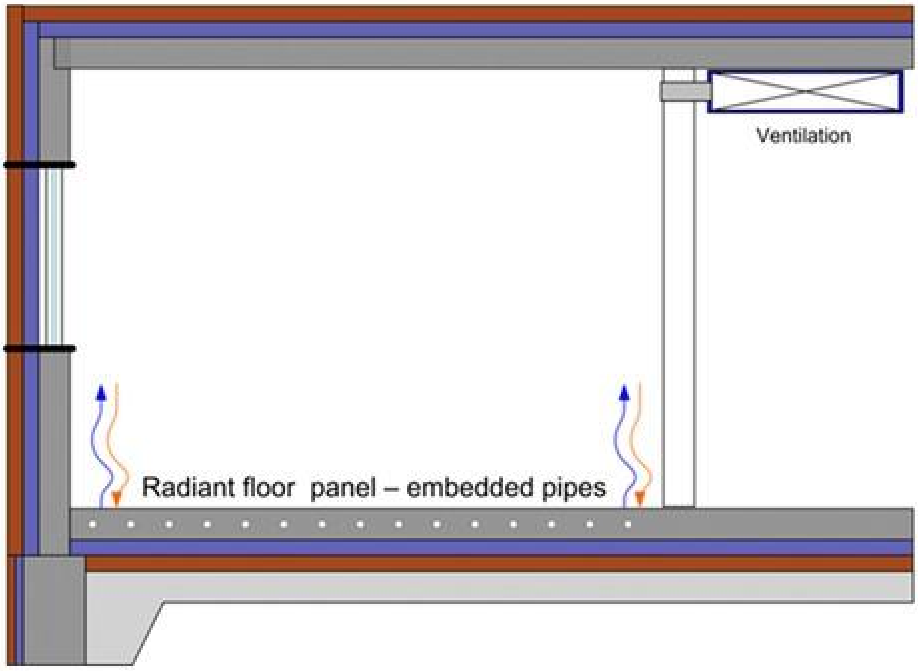

Figure 17. Radiant Floor Heating

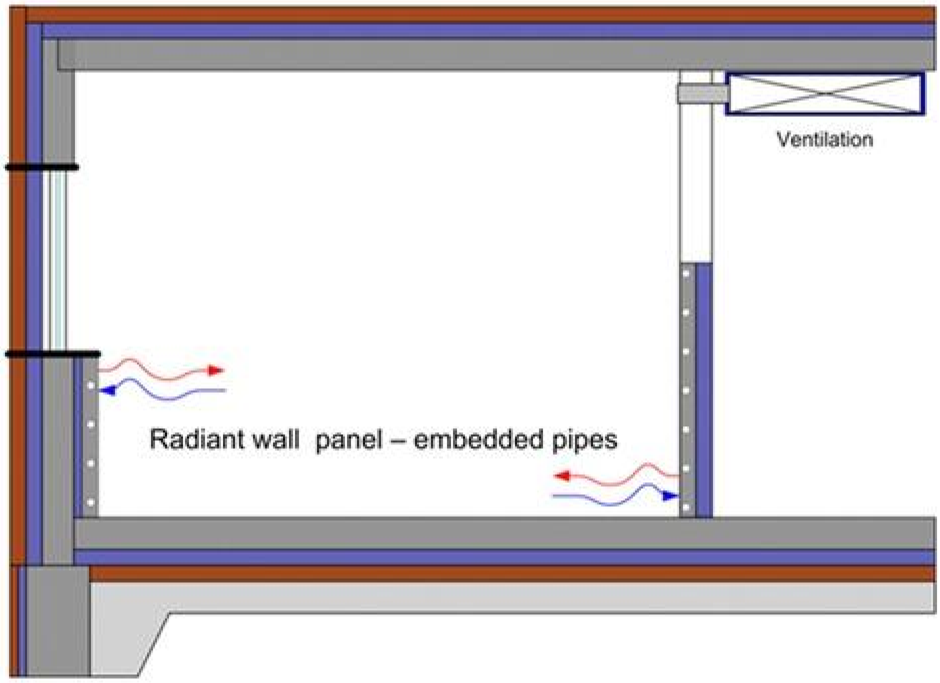

Figure 18. Radiant Wall Heating

10. FUNDAMENTALS

A radiant floor system exchanges thermal energy with the space by means of convection, short-wave radiation, and long-wave radiation. Because the understanding of the three different kinds of heat transfer is very important to calculate and optimize a radiant floor system, this section contains a detailed narrative and mathematical description. Figure 19 shows a radiant floor with edge and back insulation.

The three different types of heat transfer of the radiant floor are:

- **Convection:** heat transfer between the floor surface and the conduction air of the space.
- **Long-wave radiation:** heat flux between the floor surface and the room surfaces; its quantity and wavelength are temperature dependent
- **Short-wave radiation:** sources include high-temperature surfaces such as the sun and electric lights. The transfer of short-wave radiation within a room does not depend on the temperature of surfaces. Short-wave radiation on the floor will be either absorbed or reflected, but the radiant floor itself is not a source of short-wave radiation. The main source of short-radiation in a space is from the direct sunlight entering the space and from electrical lighting.

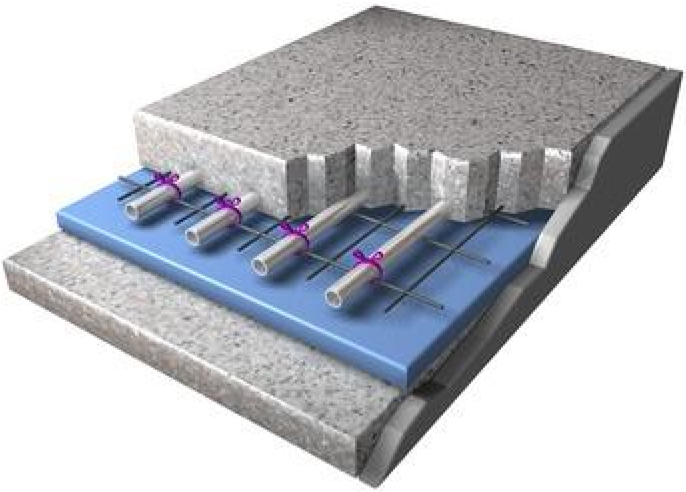

Figure 19. Typical Radiant Floor with Edge and Back Insulation

11. METHOD TO DETERMINE HEATING AND COOLING CAPACITY

A given type of surface (floor, wall, and ceiling) delivers, at a given average surface temperature and indoor temperature (operative temperature θ_i), the same heat flow intensity (specific thermal output) in any space independent of the type of embedded system. It is therefore possible to establish a basic formula or characteristic curve for cooling and a basic formula or characteristic curve for heating, for each of the type of surfaces (floor, wall, and ceiling), independent of the type of embedded system, which is applicable to all heating and cooling surfaces.

Based on the calculated average surface temperature at given combinations of medium (water) temperature and space temperature, it is possible to determine the steady state heating and cooling capacity.

HEAT EXCHANGE COEFFICIENT BETWEEN SURFACE AND SPACE

The relationship between heat flow density and mean differential surface temperature (see Figure 20 and Equations [1] to [4]) depends on the type of surface (floor, wall, ceiling) and whether the temperature of the surface is lower (cooling) or higher (heating) than the space temperature.

For floor heating and ceiling cooling in Figure 20, the heat flow density q is given by

$$q = 8.92(\theta_{S,m} - \theta_i)1.1 \quad \textbf{(1)}$$

where $\theta_{S,m}$ is the average surface temperature and θ_i is the nominal indoor operative temperature.

For other types of surface heating and cooling systems, the heat flow intensity q is given by

$$\text{Wall heating and wall cooling: } q = 8(|\theta_{S,m} - \theta i\,|) \quad \textbf{(2)}$$

$$\text{Ceiling heating: } q = 6(|\theta_{S,m} - \theta_i|) \quad \textbf{(3)}$$

$$\text{Floor cooling: } q = 7(|\theta_{S,m} - \theta_i\,|) \quad \textbf{(4)}$$

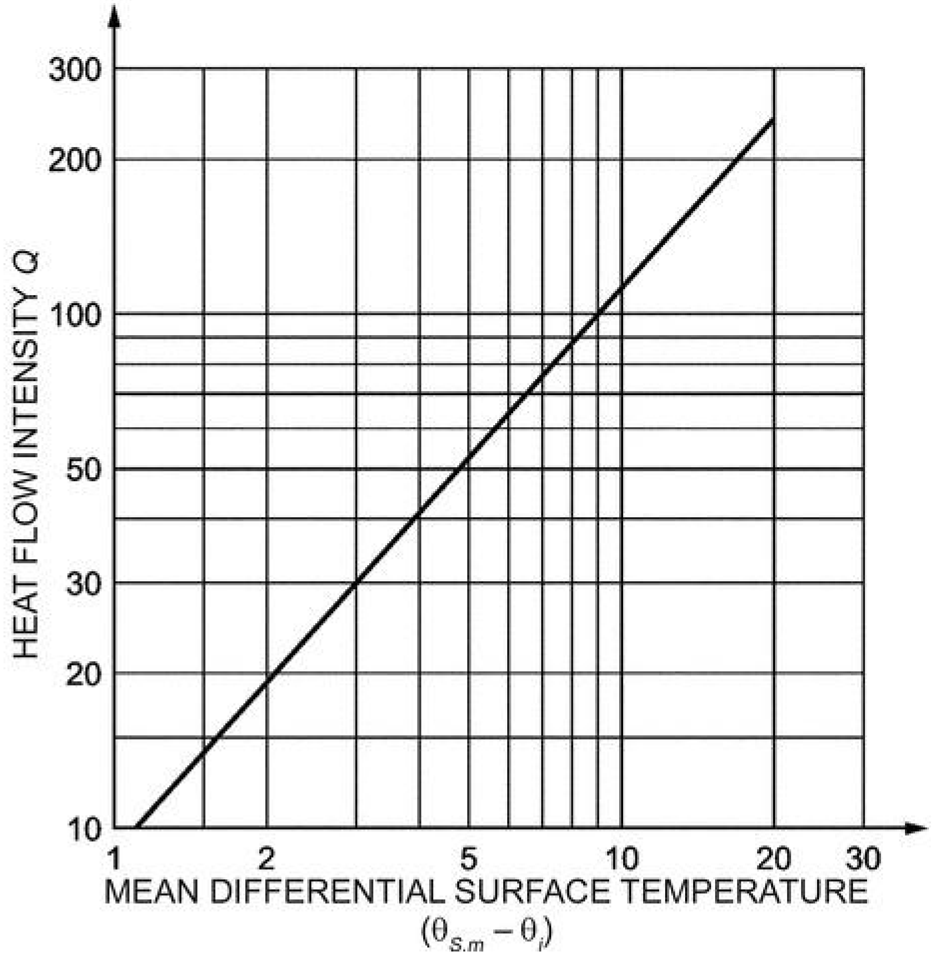

Figure 20. Basic Characteristic Curve for Floor Heating and Ceiling Cooling

12. THERMOACTIVE BUILDING SYSTEMS (TABS)

Thermoactive building systems exploit the high thermal inertia of the slab in order to perform the peak-shaving. The peak-shaving consists in reducing the peak in the required cooling power (Figure 21), so that it is possible to cool the structures of the building during a period in which the occupants are absent (during nighttime, in office premises). This way the energy consumption can be reduced, and lower night time electricity rate can be used. At the same time a reduction in the size of heating/cooling system components (including the chiller) is possible.

TABS may be used both with natural and mechanical ventilation (depending on weather conditions). Mechanical ventilation with dehumidifying may be required depending on external climate and indoor humidity production. In the example in Figure 22, the required peak cooling power needed for dehumidifying the air during daytime is sufficient to cool the slab during night time.

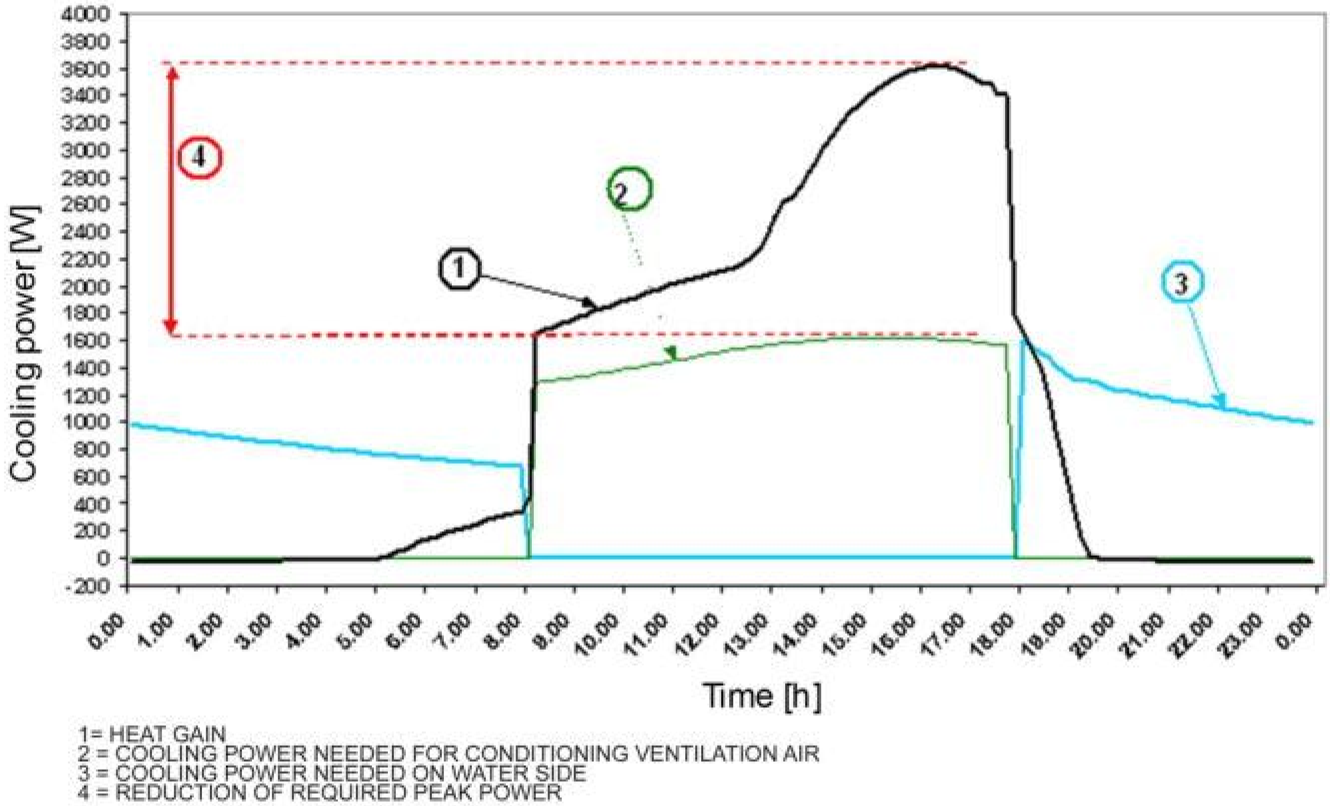

Figure 21. Example of Peak-Shaving Effect

As regards the design of TABS, the planner needs to know if the capacity at a given water temperature is sufficient to keep the room temperature within a given comfort range. Moreover, the planner needs also to know the heat flow on the water side to be able to dimension the heat distribution system and the chiller/boiler. The present document provides methods for both purposes.

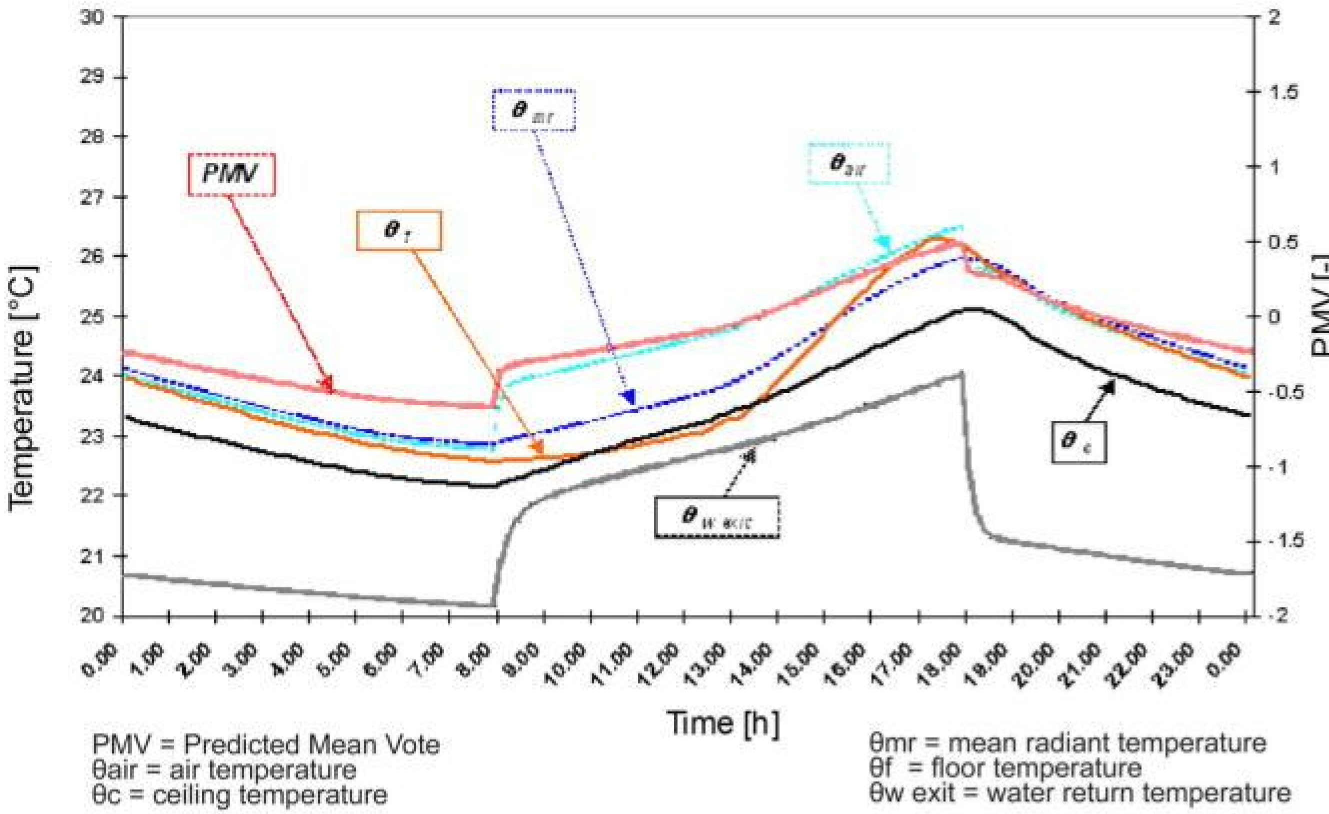

Figure 22. Example of Temperature Profiles and PMV Values Versus Time

When using TABS (Figure 23), the indoor temperature changes moderately during the day and the aim of a good design of TABS is to maintain internal conditions within the range of comfort, i.e. –0.5 < PMV < 0.5, during the day, according to ISO *Standard* 7730 and ASHRAE *Standard* 55-2020.

Some detailed building-system calculation models have been developed, as for the determination of the heat exchanges under unsteady state conditions in a single room, determination of thermal and hygrometric balance of the room air, prediction of comfort conditions, check of condensation on surfaces, availability of control strategies and calculation of the incoming solar radiation. The use of such detailed calculation models is, however, limited due to the high amount of time needed for the simulations. The development of a more user-friendly tool is required.

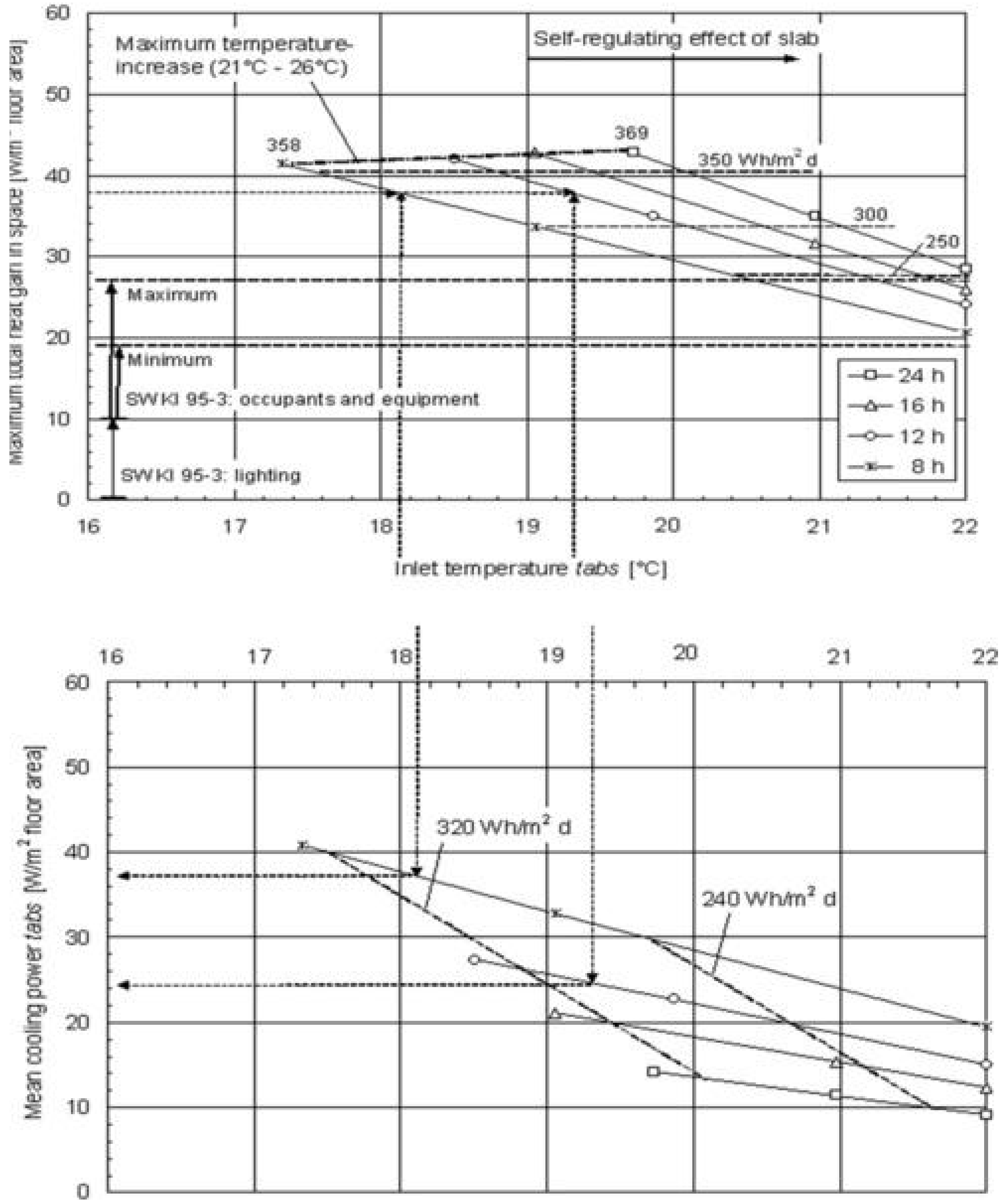

Figure 23. Working Principle of TABS

The diagrams in Figure 22 show an example of the relation between internal heat gains, water supply temperature, heat transfer on the room side, hours of operation and heat transfer on the water side. The diagrams refer to a concrete slab with raised floor (R = 2.5 ft^2·h·°F/Btu) and an allowed room temperature range of 69.8 to 78.8°F.

The upper diagram shows on the *y* axis the maximum permissible total heat gain in space (internal heat gains plus solar gains), and on the *x* axis the required water supply temperature. The lines in the diagram correspond to different operation periods (8, 12, 16, and 24 h) and different maximum amounts of energy supplied per day.

The lower diagram shows the cooling power required on the water side (to dimension the chiller) for thermoactive slabs as a function of supply water temperature and operation time. Further, the amount of energy rejected per day is indicated.

The example shows that, for a maximum internal heat gain of 12 Btu/h·ft^2 and 8 h operation, a supply water temperature of 64.76°F is required. If, instead, the system is in operation for 12 h, a supply water temperature of 66.74°F is required. In total, the amount of energy rejected from the room is approximately 106 Btu/h·ft^2 per day. In the same conditions, the required cooling power on the water side is 11.8 Btu/h·ft^2 (for 8 h operation) and 7.9 Btu/h·ft^2 (for 12 h operation) respectively. Thus, by 12 h operation, the chiller can be much smaller.

13. EMBEDDED SYSTEMS CONTROLS

Over the years there have been many different control solutions provided for embedded systems; most of these have been for heating systems. With the advent of cooling systems different approaches have been taken; not all of these have been successful. The control systems described in this section are some of the most successful used to control an embedded system for both heating and cooling. The control system will be capable of varying heating or cooling outputs as well as maintaining predetermined surface temperatures.

Control of the heating and cooling system shall enable the specified designed indoor temperatures to be achieved under the specified variation on internal loads and external climate and, if specified, protect buildings and equipment against frost and moisture damage where necessary (when normal comfort temperature level is not required).

The design of control system shall take into account the building, its intend use and the effective functioning of the heating system, efficient use of energy and avoiding heating the building to full design conditions when not required. This includes keeping distribution heat losses as low as possible (e.g., reducing flow temperature when normal comfort temperature level is not required). Control and operation of the system help us to handle the conditioning systems with savings of operational costs and enable the maintenance of required indoor environmental conditions.

To maintain the stable thermal environment, the control system is required to keep the balance between supplied heat from the system and the losses/gains of building environment under transient conditions. Slowly varying energy flows in form of heat losses through the envelope are determined by indoor and outdoor temperature, and direction and speed of wind. The envelope properties of building materials and windows, ventilation systems, building tightness and orientation play the key role in terms of heat flow quantity. The thermal capacity of building envelope (when active) can remove quickly varying uncontrolled internal heat gains (occupants, artificial lighting, equipment) and window shading elements eliminate quickly varying periodic and predictable external heat gains from sun radiation. These uncontrolled gains play a significant role in the fall and the spring. Principally, the control strategy depends on the design characteristics, such as building envelope, thermal inertia, the system response times and others. The heating control modes are based on three system levels:

- Local (individual) control, where the heat supplied to heated space is controlled
- Zone control heat supplied to a zone normally consisting of several spaces (rooms) is controlled
- Central control heat supplied to the whole building is controlled by a central system

The control system classification is based on performance level:

- Manual: heat supply to the heated space is only controlled by a manually operated device
- Automatic: a suitable system or device automatically controls heat to the heated spaces
- Timing function: heat supplied to heated space is shut off or reduced during scheduled periods (e.g., night setback)
- Advanced timing function

Heat supply to the heated space is shut off or reduced during scheduled periods (e.g., daytime with more expensive electricity tariff). Restarting the heat supply is optimized based on various considerations, including reduction of energy use.

Several years ago, most of the controls were manual (i.e., the user could regulate a water temperature or a water flow rate by manually adjusting a valve, or the system could be turned on or shut off). Today, automatic controls are used everywhere and have in the last decade developed significantly (fuzzy logic, wireless data transmission, introduction of protocols for data communication, etc.).

For a floor heating and cooling system, the control is normally split up in a central control and an individual room control. The central control will accord the outdoor climate (based on the heating curve, which is influenced by building mass, heat loss, and differences in heat required by the individual rooms) to control the supply water temperature to the floor system. The room control will then control the water flow rate or water temperature individually for each room, according to the point selected by the user.

CENTRAL CONTROL (HEATING ONLY)

Instead of controlling the supply water temperature, it is recommended to control the average water temperature (mean value of supply and return water temperature) according to outdoor and/or indoor temperatures. This is more directly related to the heat flux into the space. If, during the heating period, for example, the internal load in the space increases, the heat output of the floor system will decrease and the return temperature will increase. If the central control is controlling the average water temperature, the supply temperature will automatically decrease due to the increase in return temperature. This will result in a faster and more accurate control of the heat in put to the space and will give about 15% better energy performance than controlling the supply water temperature. If the heating system is operated intermittently (night and/or weekend set-back) the central control is also important for providing high enough water temperatures (boost effect) during the preheat period in the morning (additional 10 to 15%; the absolute heat requirement compared to no night setback will be lower). The energy savings by night setback in residential buildings are, however, relatively low due to the high thermal insulation standard in new houses.

INDIVIDUAL CONTROL

Installation of individual room temperature controls is recommended to improve comfort and the possible energy savings. Besides the energy benefits it is essential for the thermal comfort of the occupants, that they have a possibility for individual adjustment of the room temperature set point from room to room.

The influence of the individual room control strategy for floor heating and radiators have shown a 15 to 30% energy saving by using an individual room control compared to central control only. Also the effect of night set back and boost by reheating in the morning was studied, the advantage of a boost heating (i.e., the water temperature is increased above the temperature corresponding to the heating curve during the beginning of the preheat period in the morning). This reduces the preheat time and a longer setback period is possible. Boost reheating improves the energy performance by around 8%.

ROOM THERMOSTATS/SENSORS

In case of a floor heating system, the valve on the manifold is controlled by and connected to a room sensor by wiring.

In terms of comfort, it is preferable to control the room temperature as a function of the operative temperature in the area occupied by the person. Besides the position, it is important to consider the shape, size, and color (important for short-wave radiation, sunlight) of the sensor in order to express convective and radiant heat exchange between sensor and space similarly as for the person.

The positioning of the room temperature sensor in the area occupied also save energy in comparison to positioning on the wall. By positioning in the occupied area, the variations in the room temperature are smaller.

TIME DELAY, TIME RESPONSE

Floor mass is naturally closely linked to system time response. Unpredictable thermal gains/losses leading to reduced/increased heating load with characteristic times shorter than time constant cannot be compensated by the control system. An obvious consequence of the response time of a conventional floor structures is that the instant control of the heating power is not necessary. Hence, regulating the energy supply to the floor system within a time interval correlated to the response time is not different from a continuous power control.

SELF-REGULATING EFFECT

Due to the high impact the quickly varying heat gains (sunshine through windows) may have on the room temperature, it is necessary that the heating system can control for that (i.e., reduce or increase the heat output). For a low-temperature heating system like floor heating, a significant effect is so-called self control (Figure 24), which depends partly on the temperature difference between room and floor surface and partly on the difference between room and the average temperature in the layer, where the tubes are embedded. It means that fast change of operative temperature will equally change of heat exchange and result in great influence of total heat exchange.

14. RADIANT COOLING SYSTEM CONTROL

The cooling output of radiant heat exchange may be limited to avoid condensation on the surface and in the building structure.

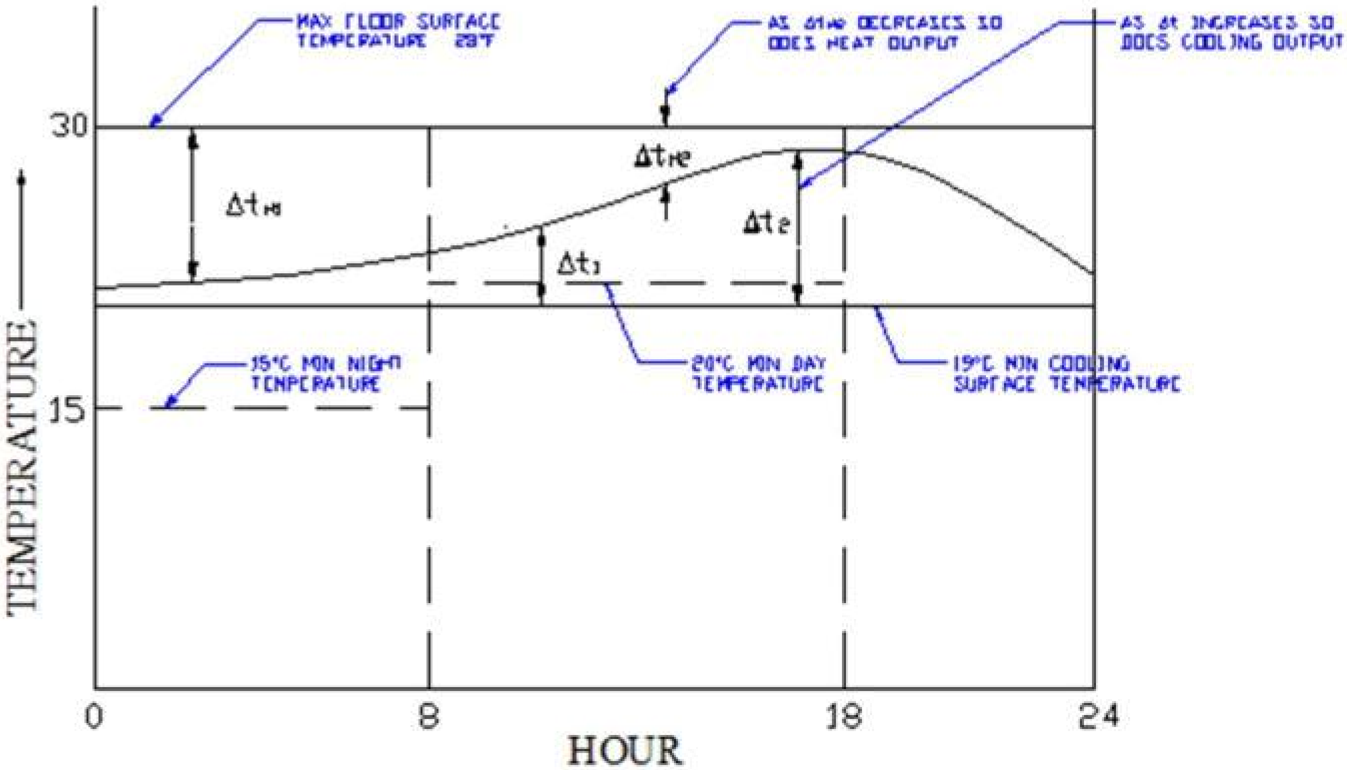

Figure 24. Self-Regulating Effect from Radiant Floor: As Temperature Differential Between Floor Surface and Space Dry-Bulb Temperature Increases, so Does Cooling Output from Floor for both Heating and Cooling

A central control is also essential when using floor cooling. Due to the limitation of the cooling capacity a floor system will not always be able to control the room temperature at a fixed level. Basically, the control should provide the maximum cooling power taking into account comfort (floor temperature, room temperature) and the risk for condensation (dew point temperature). The central control for floor cooling must then take into account the dew point in the building/space, when controlling the supply water temperature. This is done by adding a humidity sensor in the building/space connected to the central control unit.

CONTROL OF TABS

In this case, individual room control is not reasonable, but a zone control (south to north), where the supply water temperature, average water temperature, or flow rate may differ from zone to zone, is available. The zoning should consider the external and/or internal heat loads. Relative small temperature differences between the heated or cooled surface and the space are typical for TABS systems. This matter results in a significant degree of self control in specific cases by well-designed systems with low heating/cooling load, a concrete slab can be controlled at a constant core (water) temperature year round. If, for example, the core is kept at 71.6°F, the system will heat at room temperature below 71.6°F, and cool above the room temperature of 71.6°F.

To avoid condensation (on or under surface), the water temperature or the surface temperature and the absolute humidity have to be controlled. One possibility is to set a lower limit for the supply

If the supply water temperature is limited, so it will never be below the dew point, then all temperatures (water, floor surface) after the mixing valve will be higher than the dew point, and there is no risk for condensation on the pipes, in the floor construction or on the floor.

CONTROL SYSTEM COMPONENTS

Figure 24 shows the control system components which comprise a closed circuit complete with circulating pumps. This circuit is connected to the supply and return header. The circuit is also provided with all the necessary isolating and balancing valves. There are two mixing pipes installed in the circuit. One is for heating and the other is for cooling. The heating mixing pipe is connected to a heating supply which can have a variable temperature together with a return. The cooling mixing pipe is connected to a cooling supply which can have a variable temperature together with a return.

This section will outline the performance of a radiant cooled floor subjected to different solar radiation intensities and will look further into the controllability of the radiant cooled floor to maintain certain conditions such as space temperatures and floor surface temperatures. Cooled floors with and without coverings will also be analyzed to assess the influence of floor coverings.

TEMPERATURE DIFFERENCES AND FLOW RATES

Figure 25 shows an example layout for dimensioning the flow through the embedded system. The flow rate is generally determined for cooling as this is more critical. Typical supply and return temperatures through the floor are 55°F and 63°F, so to provide a certain cooling output the flow rate can be determined. The flow rate through the embedded system is constant volume. Variable-volume systems can also be used and these will also be described in this section.

For example, assume the embedded system will have a cooling capacity of 1000 W (1 kW).

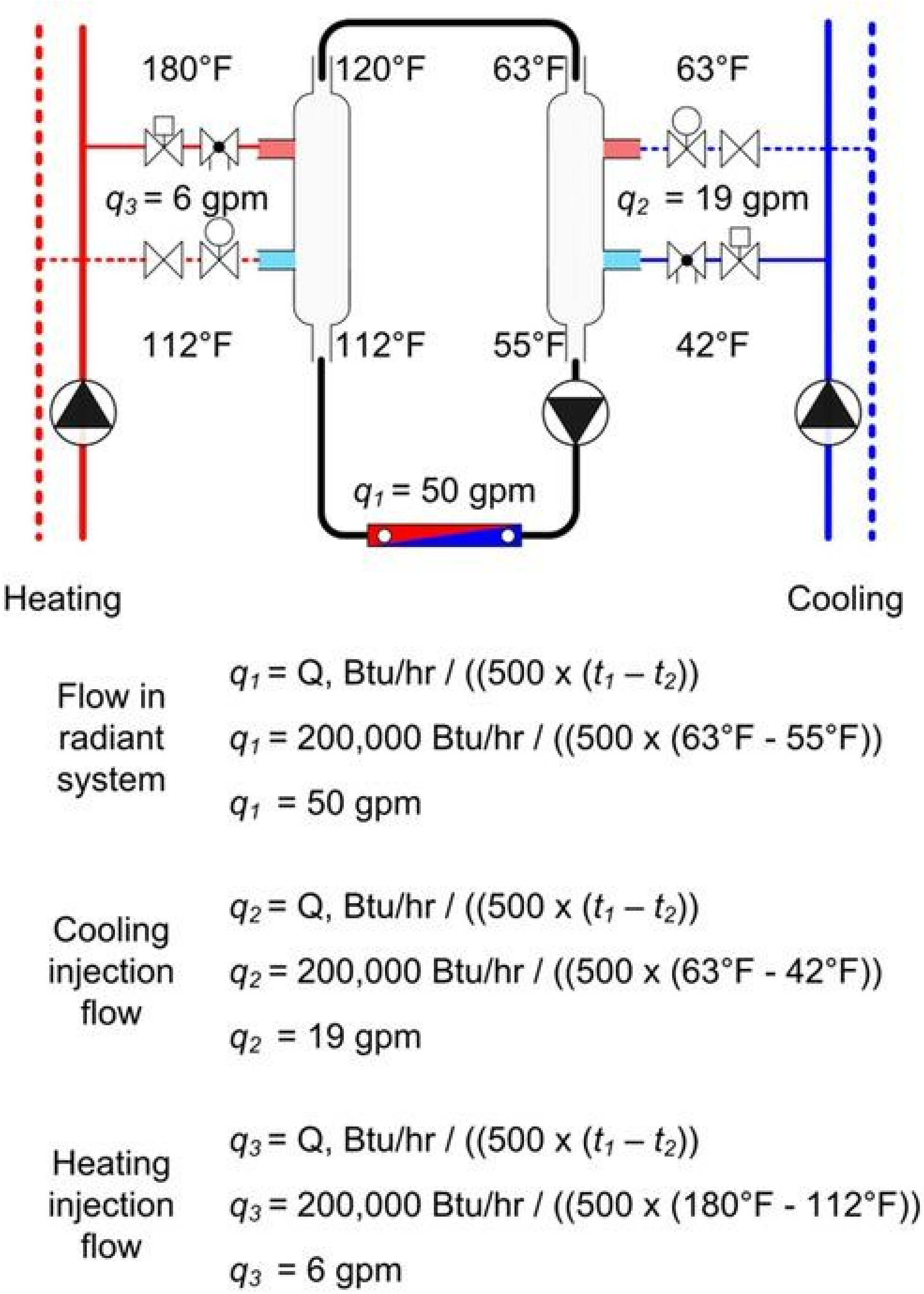

Figure 25. Heating and Cooling Connections to Radiant Floor Loop

Assuming a supply water temperature of 55°F and a return water temperature of 63°F, the flow rate through the embedded system will be 7.9 lb/min.

This is the constant-volume flow through the embedded system. As shown in Figure 25, there are both heating and cooling connections to the control loop.

Assuming that the floor is in full design cooling capacity, then the return water from the embedded system will be 63°F, and this will have to be cooled to 55°F to provide the required cooling from the embedded system. The mixing pipe is connected to a chilled-water supply and return. Again, we can assume that the chilled-water supply temperature to the mixing pipe is 42°F. The amount of cooling required from the primary chilled water is the same as the cooling output from the embedded system (1 kW). The temperature differential through the mixing pipe is 63°F return from the loop and a 42°F supply, so the $\Delta t = 63 - 42 = 21$°F.

The flow rate through the primary chilled-water connection to the mixing pipe is 2.6 lb/min.

By keeping the primary flow rate lower than the circuit flow rate, the primary connections are also smaller, which enables the two-way control valve in the chilled water connection to have an improved K_v value and authority.

Another reason for this design option is that the primary chilled-water flow rate is much lower than the circuit loop flow rate and this limits the amount of 42°F water that could be circulated through the floor.

For heating, it is a very similar situation. The flow rate through the loop remains the same constant flow rate. The maximum surface temperature of an embedded system for heating is 82.4°F. Typical heating supply and return temperatures in the circuit are 140 to 125.6°F, but the maximum heating water primary supply to the mixing pipe could be 194°F. The flow rate to the mixing pipe would then be 1 lb/m.

This is about 1/10 of the loop flow rate. Again, this keeps the primary connections smaller than the circulating loop, which keeps the pipe connection smaller, which in turn keeps the control valve to a small size which improves the K_v and the valve authorities.

For **variable-flow, constant-temperature control** (Figure 26), the water supply temperature to the floor is kept at a constant 54°F, and the floor surface temperature is maintained at 68°F. As the cooling output to the space increases to maintain space set-point temperature, the water flow through the floor is increased. For this control option, the water flow rate through the floor is proportional to the cooling output from the floor. As the water flow rate increases to provide the cooling output from the floor the leaving water temperature from the floor decreases to a minimum of 63°F.

For **constant-flow, variable-temperature control** (Figure 27), the water supply temperature to the floor is varied, the floor surface temperature is maintained at 68°F. As the cooling output to the space increases to maintain space set-point temperature, the water flow through the floor is constant. For this control option, the supply water temperature to the floor is varied from 61 to 52°F, which is proportional to the cooling output from the floor. As the supply water temperature decreases to provide the cooling output from the floor the leaving water temperature from the floor decreases to a minimum of 63°F.

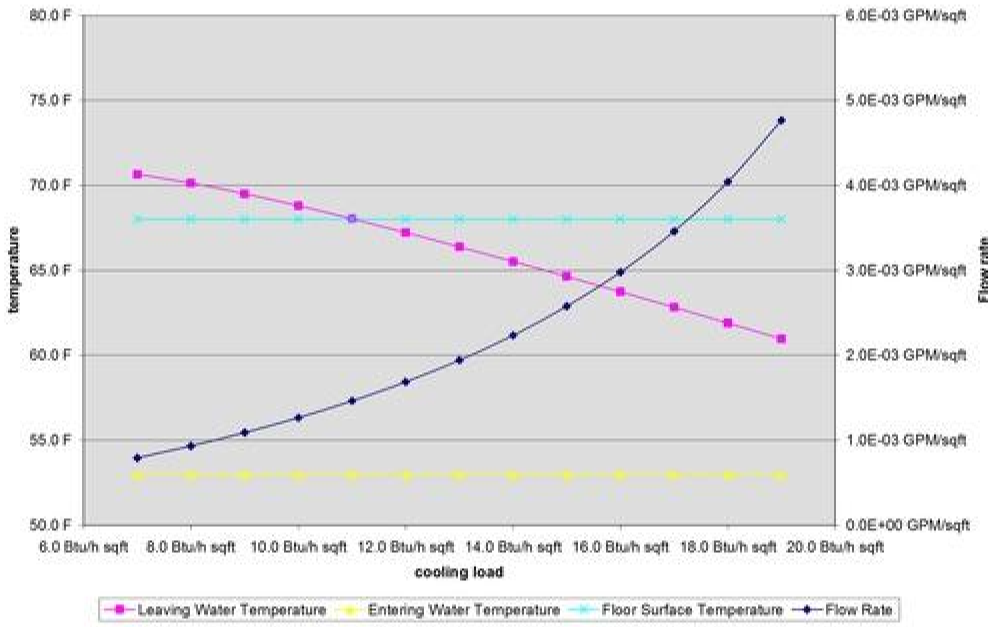

Figure 26. Characteristics of Variable-Flow Constant-Temperature Control

For **constant-flow, constant-temperature control** (Figure 28), the water supply temperature to the floor is kept at a constant 55°F, The floor surface temperature is varied from 62 to 71°F. As the cooling output to the space increases to maintain space set-point temperature, the water flow through the floor is constant. For this control option, the leaving water temperature from the floor is proportional to the cooling output from the floor. As the cooling output increases to provide the cooling output from the floor surface temperature is also increased.

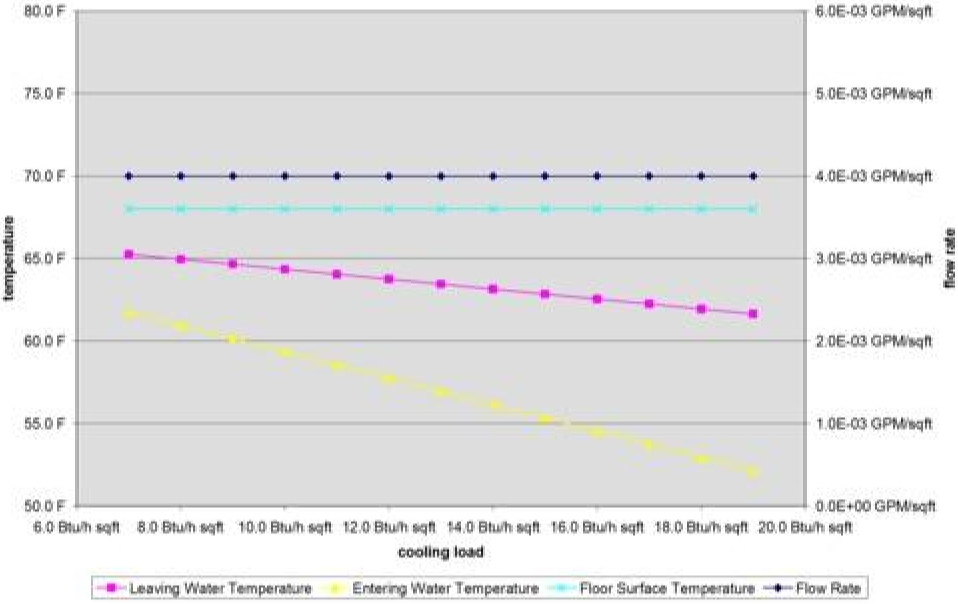

Figure 27. Constant-Flow, Variable-Temperature Control

From these three alternatives, constant flow, variable temperature provides the smoothest control.

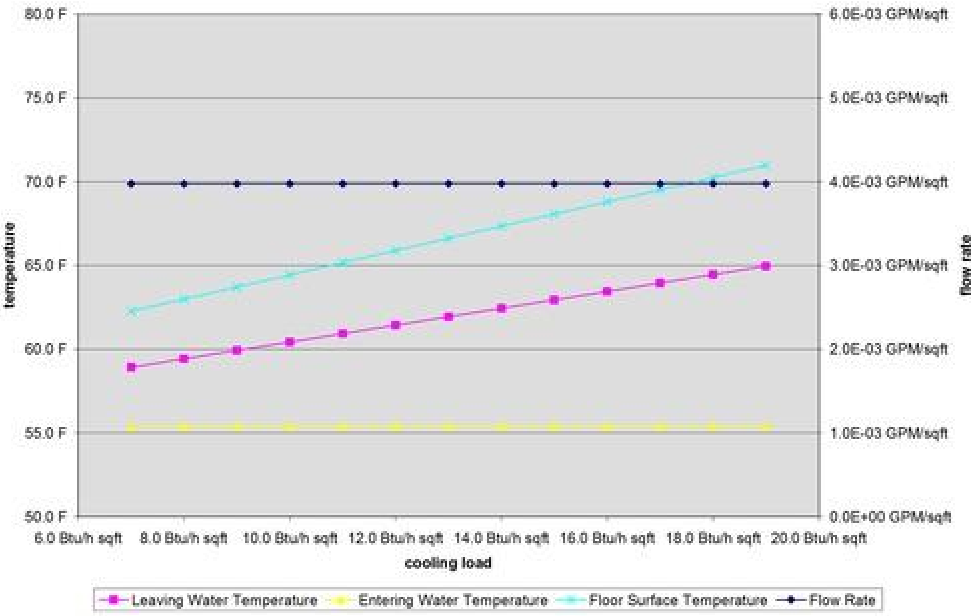

Figure 28. Constant-Flow, Constant -Temperature Control

DEW-POINT

The floor surface temperature of the radiant floor is lower than the air temperature in the space. Therefore, if the surface temperature of the floor falls down below the space air dew point at any area, water falls out and the floor gets wet. The floor surface temperature is dependent on the water temperature at the specific area, the cooling load of this area and the distance between the tubes within the floor construction. Because of the distance between the tubes, the floor surface temperature is not constant. It is recommended to select a minimal EWT equal to the supposed maximal dew point temperature to avoid a condensation on the floor surface and to guarantee the selected cooling capacity of the radiant system. Additional, the EWT should be controlled by a humidity control sensor.

If on any reason an EWT below the supposed dew point is selected, the minimal possible surface temperature of the floor in areas with reduced loads must be calculated carefully.

For example, if the indoor air is controlled of a dry bulb temperature of 74°F and a relative humidity of 50%, the suggested dew point is approximately 55°F. Therefore, a minimal EWT of 55°F may be selected. The combination of the radiant floor cooling system with a natural ventilation concept requires an accurate calculation of the maximal suggested dew points within the space, considering the outdoor air conditions as well as the latent loads of the space.

ROOM CONTROL

In some buildings, it may be necessary to control individual rooms separately from other rooms without adjusting the weather-compensation control. One way to accomplish this is with valve actuators attached to individual manifold loops that service the rooms where control is desired.

Figure 29 shows the principles that may be applied for a space with mainly outdoor cooling and heating loads, with a combined radiant heating and cooling floor. Spaces with a high amount of interior cooling loads are quite more difficult to control if the radiant floor is used both for heating and cooling.

CONTROL STRATEGY FOR OFFICE BUILDINGS

The control strategy for office buildings with its high number of internal loads, which are independent of outdoor conditions (weather), requires more considerations.

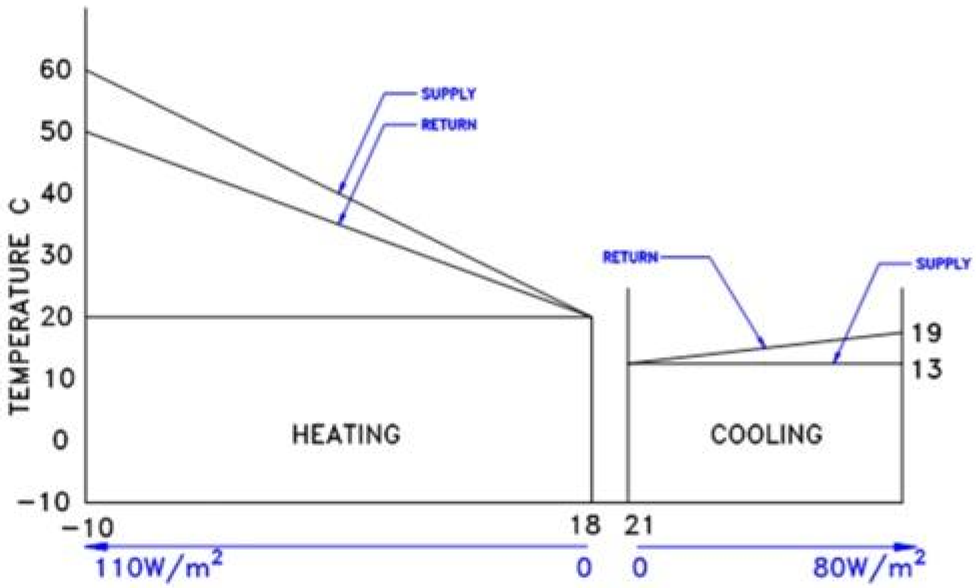

Figure 29. Control Strategy for Combined Radiant Heating and Cooling Floor

Interior Zones. The interior zones may have only cooling loads. There, the radiant floor may provide only cooling to the zone. The EWT may be kept constant and the water flow (on or off) is controlled by a space thermostat. It is important to avoid an under cooling of the space during the night or during unoccupied periods. The water flow should switch off, if the space temperature decreases under the allowed level. It is important to consider, that there is a defined amount of cooling power stored within the construction slab, so the switch-off level should be higher than the allowed minimal temperature level.

Perimeter Zones. Perimeter zones may be equipped with a radiant floor system that provides both heating and cooling to the space. If the radiant floor is the only heating system, the radiant floor must be selected for the heating loads. The EWT may be controlled by a weather compensation control combined with a zone control system. In cooling mode, the EWT should be kept on a constant level. In most cases, the radiant floor cooling system for a perimeter zone is combined with either a mechanical ventilation or a natural ventilation system to provide the remaining cooling capacity and the necessary fresh air and to reduce the latent loads. It is important to avoid the cooling of a heated floor slab or the reverse situation during normal weather situations. The following strategy may be used.

During periods where heating and cooling is necessary during the same day, the radiant floor should provide only heating to the space. The space cooling should be provided by the additional system (may be through operable windows). This can be designed by using a delay element of at least 1 day to switch the radiant floor system between heating and cooling mode. Outside temperatures are usually not as high during periods that require heating and cooling during the same day. So the additional air ventilation system may be used in economizer mode.

During periods where usually no heating is necessary, the floor is used only for cooling as the first cooling system in a same way as in the interior zones.

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