uponor

Radiant Cooling Design Manual For Use in Humid Climates

June 2016 Edition

www.uponor.hk

Table of Contents

Introduction

As we discussed in our previous eBook, compared to conventional All-Air systems, radiant cooling in tandem with dedicated outdoor air systems (DOAS) as an 'Air-and-Water' system has many significant and compelling advantages for use in South East Asia. Increased energy efficiency, superior comfort, greater architectural freedom to reduced operating and maintenance costs make radiant cooling the obvious choice- whatever the overriding constraints defined by the project owner.

However, as a basic introduction and comparison with All-Air systems we left out many of the technical details regarding design and implementation of radiant cooling systems. The aim of the design manual is to provide comprehensive and tangible methods and means for designing radiant cooling systems in South East Asia. It is designed for engineers and architects alike, aiming to enable the synergy and common understanding between the two fields that is required for the successful implementation of any ambitious building project.

Our first port of call is to review the fundamental principles and calculations on which radiant solutions are built, followed by an introduction to zoning, controls and piping, finally to Air-and-Water system design.

1. Fundamental Calculations Behind Air-and-Water Systems

Mean Radiant Temperature

ASHRAE Standard 55-2010 defines six factors that affect thermal human comfort: air temperature, radiant temperature, humidity, air velocity, clothing and metabolism. Since the latter two factors are occupant-dependent, only the first four can be monitored and controlled by an All-Air system.

Traditional All-Air air conditioning systems typically only monitor and control three of these space conditions, ignoring radiant temperature. Radiant heating and cooling systems address mean radiant temperature (MRT), which is a key factor in thermal comfort. MRT is defined as the theoretical uniform surface temperature of an enclosure in which an occupant would exchange the same amount of radiant heat as in the actual non-uniform enclosure. Unlike in an All-Air system, the MRT in a radiant conditioned space recognizes the intimate relationship occupants have with the surroundings via radiant heat transfer.

This relationship is a key component in thermal comfort when integrated with air temperature to "operative temperature" indices as referenced in thermal comfort standards.

Operative Temperature

The operative temperature is numerically the average of the air temperature ta and mean radiant temperature tr, weighted by their respective heat transfer coefficients. Most requirements for comfort are based on the operative temperature in a space.

The operative temperature is calculated as:

$$\theta_{f,i} = \frac{(h_c \cdot t_a) + (h_r \cdot t_r)}{h_c + h_r}$$

Where

t_a = air temperature in reference point, ºF (ºC)

t_r = mean radiant temperature in reference point, ºF (ºC)

h_c = convective heat transfer coefficient for the human body, Btu/h . ft^2 . ºF (W/m^{2}K)

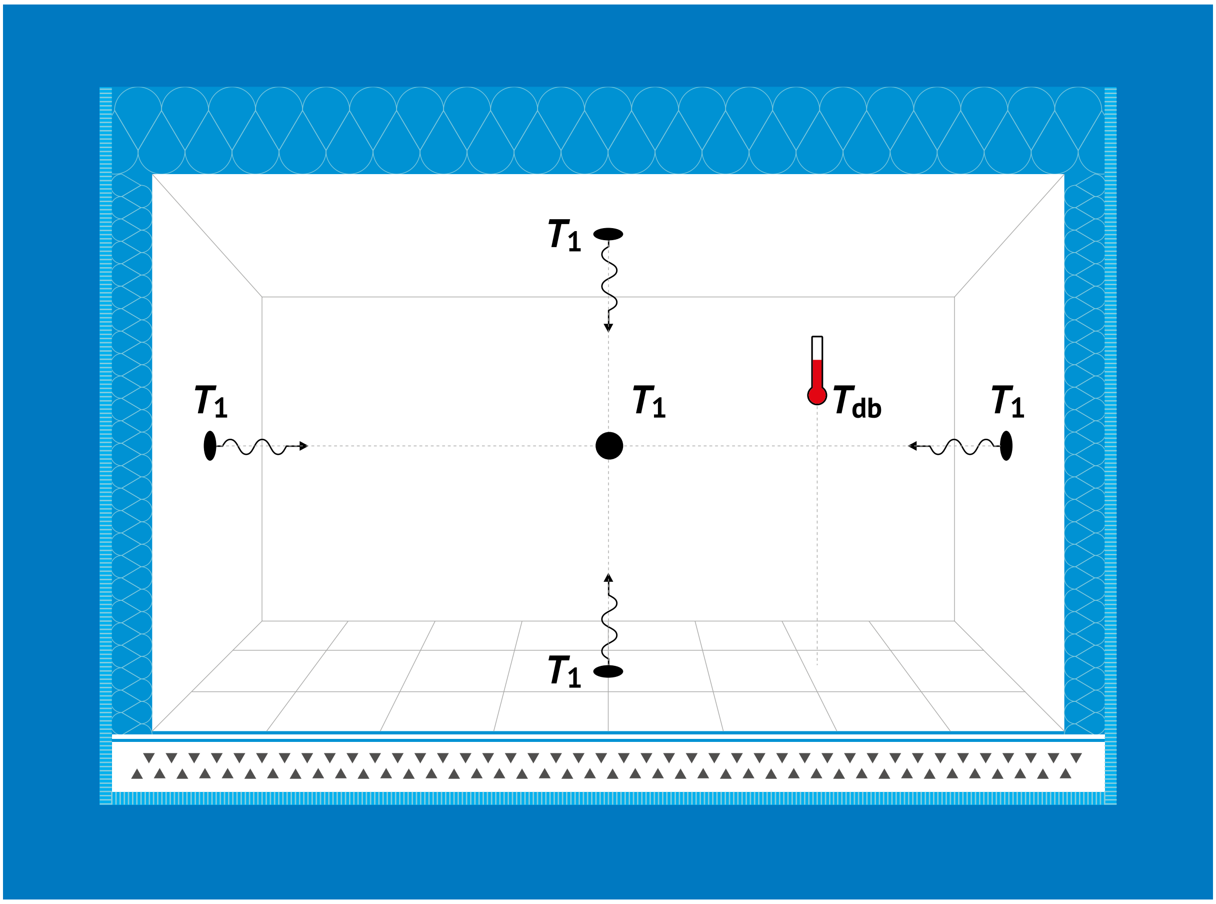
h_r = radiant heat transfer coefficient for the human body, ºF (ºC) Btu/h . ft^2 . ºF (W/m^{2}K)

In most practical cases where the relative air velocity is small at < 40 fpm (0.2 m/s) or where the difference between mean radiant and air temperature is small at < 4°C, the operative temperature can be calculated with sufficient approximation as the average of air and mean radiant temperature.
(Source: ANSI/ASHRAE Standard 55-2010, Thermal Environmental Conditions for Human Occupancy.)

$$\theta = \frac{AirTemperature + MRT}{2}$$

However, if the mean radiant temperature is significantly lower or higher than the air temperature, the convective and long-wave radiant heat flux should be calculated separately.

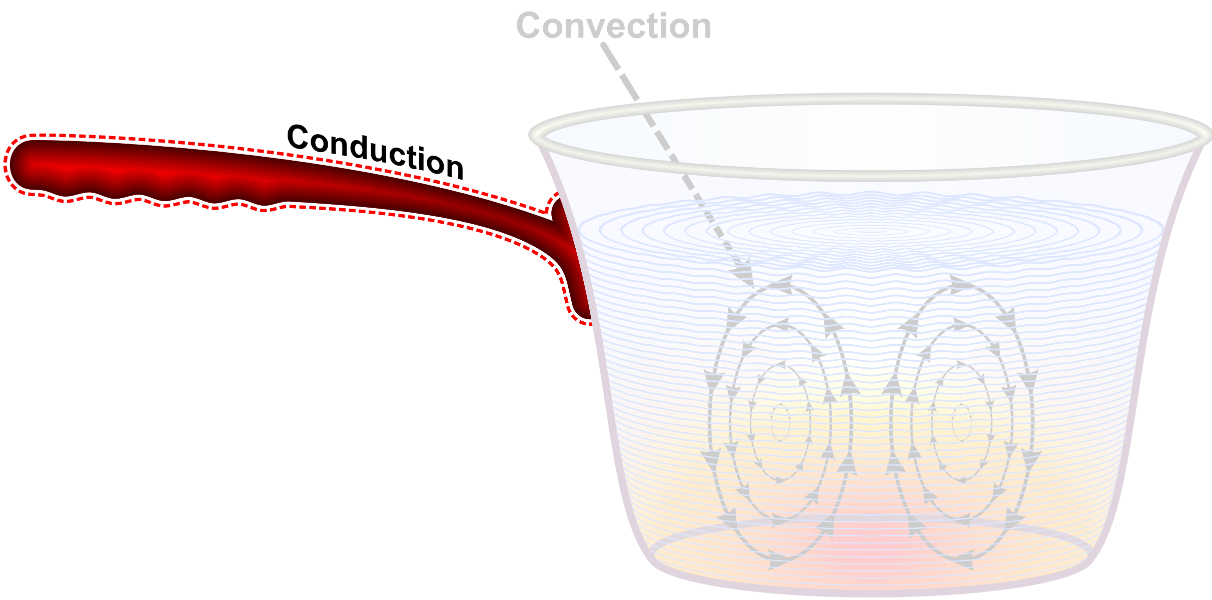
MRT calculation diagram



Heat Transfer Basics

Heat transfer occurs whenever there is a temperature difference between two objects, and it continues until both objects are in thermal equilibrium. According to a formulation of the Second Law of Thermodynamics known as the Clausius statement, heat cannot naturally flow from a colder temperature to a hotter temperature. In other words, heat will always naturally flow from hot to cold. Heat is transferred in three ways: **conduction, convection and radiation.** A radiant cooling system uses all three modes of heat transfer.

Conduction



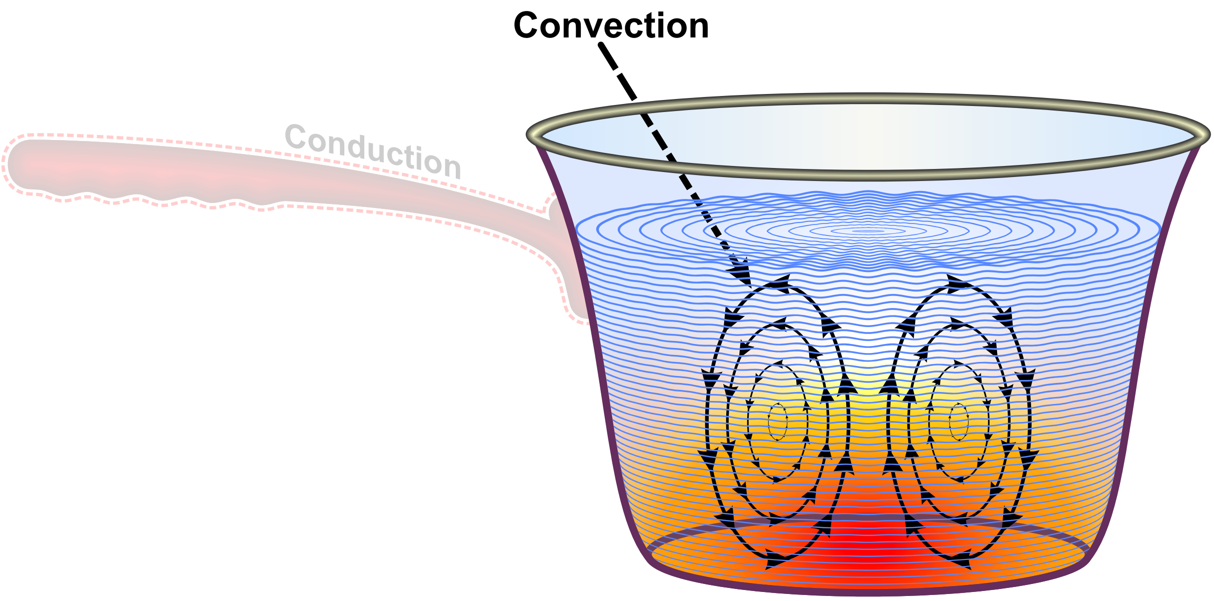
Conduction is heat transfer between two solids that are in direct contact with each other. In radiant cooling systems, conduction occurs between the PEX tubing (essentially the pipes used to transmit the water comprised of medium to high-density polyethylene containing crosslinked bonds introduced into the polymer structure, changing the thermoplastic into an elastomer) and the concrete slab.

The heat transfer rate is based on the conductivity of materials, the tubing surface, and the temperature difference between the tubing and the slab. Conduction also occurs between the cooled slab and the objects in the space that are in contact with the slab, including air film, furnishings and occupants of the building.

If a person is standing on a cooled slab, then a quantity of body heat will naturally flow via conduction to the slab. The heat transfer rate is based on the cumulative R-values from footwear, the floor conductivity, and temperature difference between the occupant and the floor surface.

To prevent discomfort due to temperature differentials, ASHRAE Standard 55-2010 recommends that floor slab temperatures be above 18.9°C for occupants wearing normal footwear in occupied spaces. It should be noted that in temperature ranges typical of radiant floor cooling systems, and in consideration of footwear R-values, the amount of conductive heat transfer from foot to slab is relatively low and, therefore, typically considered negligible.

Convection



Convection is heat transferred through a moving fluid or gas. In the case of Air-and-Water systems, natural or "free" air convection occurs due to differences in air densities influenced through contact with warmed or cooled surfaces.

Natural convection is a design consideration with radiant-cooled ceilings as the layer of air in contact with the cool ceiling will drop due to its higher density, increasing air movement, and thus heat transfer, in the space.

Forced convection occurs in the air handler, where fans are used to force the cooled air into the space. Because convection deals with heat transfer through the movement of air, the air temperature is directly affected.

Radiation

Not surprisingly, the sensible heat transfer in a radiant cooling system is through radiation. Radiation is heat transfer through electromagnetic waves travelling through space. When the incident waves from a warmer surface come into contact with a cooler surface, the energy is absorbed, re-radiated, reflected or transmitted.

An example of radiation is sunlight, which travels through the vacuum of space as short-wave radiation to warm the Earth's surface. The heat-transfer rate is influenced by a number of factors, including the absorptivity, reflectivity and emissivity of the surfaces; wavelength; temperature and the spatial relationship between the cooled surface and the occupant (defined as the view and angle factors). In radiant cooling, the electromagnetic waves from the occupant are drawn toward the cooled surface, resulting in the occupant experiencing a cooling effect.

Long-wave Radiation: Long-wave radiation is the heat flux that occurs between the conditioned surface and the unconditioned room surfaces; its quantity and wavelength are temperature-dependent.

Short-wave Radiation: The transfer of short-wave radiation upon room surfaces from solar gains or high intensity lighting is not dependant on the temperature of the absorbing surface. Energy at this intensity upon a surface at room conditions will be absorbed, reflected and/or transmitted based upon the color and optical characteristics (reflectivity, absorptivity, transmissivity) of the receptor surfaces.

The total heat flux of the radiant floor system can be written as the sum of the three types of heat transfer:

$$q_{tot} = q_{con} + q_{l,\ rad} + q_{s,\ rad}$$

Where:

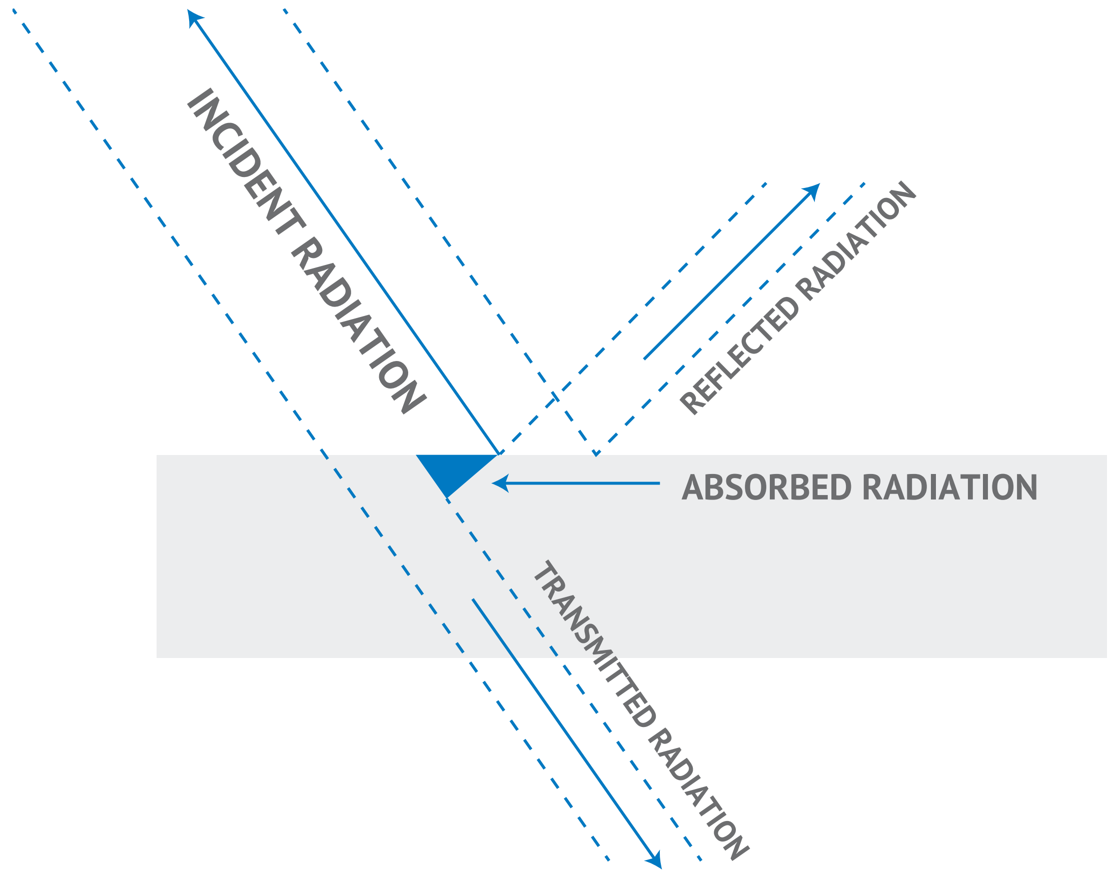
q_{tot} = total energy transfer, Btu/ft², (W/m²)
q_{con} = convective energy transfer, Btu/ft², (W/m²)
$q_{l,\ rad}$ = long-wave radiant energy transfer, Btu/ft², (W/m²)
$q_{s,\ rad}$ = short-wave absorption, Btu/ft², (W/m²)

The sum of convective q_{con} and long-wave radiant heat transfer expression $q_{l,rad}$ is defined as the space energy transfer qs, Btu/ft2, (W/m2), and can be written as:

$$q_s = q_{con} + q_{l,rad}$$
$$q_{con} = h_{con} \cdot (t_f - t_{air})$$
$$q_{l,rad} = h_{s,rad} \cdot (t_f - t_{MRT})$$

Where:
h_{con} = convective energy exchange coefficient floor to space, Btu/ft² • °F, (W/m²K)
$h_{l,rad}$ = long-wave radiant energy exchange coefficient floor to space, Btu/ft² • tF, (W/m²K)
t_{air} = space air temperature, °F (°C)
t_{MRT} = surrounding surface temperature, °F (°C)
t_f = floor surface temperature, °F (°C)

As previously discussed, the radiation within a space is usually separated into two groups: longwave and shortwave. The long-wave radiation is that which occurs between room surfaces. The short-wave radiation upon a cooled floor should be considered; its incident energy will be absorbed, reflected and/or transmitted based upon the color and optical characteristics of the receptor surfaces.



Where:

The first law of thermodynamics:

$$\alpha + \tau + \rho = 1$$

α = fraction of incident radiation absorbed (absorptance).
τ = fraction of incident radiation transmitted (transmittance).
ρ = fraction of incident radiation reflected (reflectance).

The floor surface is opaque, so the transmittance of the floor surface $\tau = 0$. For a perfectly black surface where $\alpha = 1$, $\rho = 0$, $\tau = 0$, all short-wave radiation reaching the surface will be absorbed by the black surface. In reality, for most surfaces, absorptance for short-wave radiation (high-temperature radiation) is different than emittance for long-wave radiation (low-temperature radiation).

Surfaces	Absorbtance α for solar Radiation
Carpet - dark-colored	0.80 - 0.90
Carpet - light-colored	0.50 - 0.60
Tile or plaster, white or cream	0.30 - 0.50
Red tile, stone or concrete, dark paints (red, brown, green, etc.)	0.65 - 0.80
White-painted surfaces	0.23 - 0.49

Source: ASHRAE Fundamentals

Solar absorptance can also vary with the size of windows. Absorptance can range from 0.90 for dark-colored spaces with small windows to 0.60 or less for light-colored spaces with large windows. Thus, when considering radiant cooling as part of a holistic Air-and-Water cooling solution window size can be an important and useful architectural consideration.

When using textile-based floor coverings, the slab temperature required to draw down the floor surface temperature must be evaluated to ensure it does not approach the dew point temperature. Similarly, this is also an important architectural and engineering consideration when designing radiant systems.

	Surface Temperature	
	Maximum Heating	Minimum Cooling
Floor occupied zone	66°F (18.9°C)	66°F (18.9°C)
Floor perimeter zone	95°F (35°C)	66°F (18.9°C)
Wall	104°F (40°C)	62°F (16.7°C)
Ceiling	80°F (26.7°C)	62°F (16.7°C)

Direct Solar Loads

In lobbies, atria or other areas with high direct solar loads, radiant floor cooling can be especially effective. Without floor cooling, maintaining comfort in these areas is a significant challenge. Given the trend for lobbies with elevated ceilings and roomy atria throughout skyscrapers in regions such as Hong Kong and Singapore; clearly an opportunity to reduce operational costs and increase energy efficiencies by exploiting an existing architectural trend.

Energy from the sun is transmitted through the glazing as short-wave radiation. As it hits the floor surface, a portion of this energy is absorbed by the slab, while the remaining portion is reflected into the space as longwave radiation, contributing to the space heat gain.

Flux, Radiant Floor Cooling Including Short-wave Radiation

$$q_c = h_{c,tot} \cdot (t_o - t_s) + q_{s,rad}$$

Where:

q_c = specific heat flux between floor surface and space, Btu/ft²(W/m²)
$h_{c,tot}$ = heat exchange coefficient, Btu/hr . ft² . °F, (W/m²K)
t_o = operative temperature, °F (°C)
t_s = floor surface temperature, °F (°C)
$q_{s,rad}$ = absorbed short-wave radiation, Btu/ft²(W/m²)

In floor areas with absorption of shortwave radiation, the total floor cooling capacity can be as high as approximately 25 to 32 Btu/h/ft2 (80 to 100 W/m2). The capacity is dependent on the efficiency of the floor viewed as a heat exchanger. If the exchanger shows a high thermal resistance, then the space cooling capacity is suppressed.

If short-wave absorption through the floor is higher than the steady cooling capacity of the floor, energy not absorbed will raise the temperature of the floor surface and could eventually cause the surface to emit long- wave radiation back into the space.

Long-wave Radiant Energy Exchange Coefficient

It is assumed that all surfaces are radiantly gray, so the radiant heat flux between floor surface and the other surfaces (Fanger 1982) can be written as:

$$q_{rad} = \varepsilon_f \cdot \sigma \cdot T_i^4 - \sum_{i=1}^{N} \varepsilon_i \cdot \sigma \cdot T_i^4 \cdot F_{Af-Ai}{}^4$$

Where:

q_{rad} = radiant heat flux between floor surface and the other surfaces, Btu/h ft² (W/m²)
ε_f = emittance of floor surface
ε_i = emittance of the other surfaces
σ = Stefan-Boltzmann constant
T_f = absolute temperature of floor surface, K
T_i = absolute temperatur of surfaces, K
$F_{Af\text{-}Ai}$ = view factor between floor surface $_{Af}$ and $_{Ai}$ dimensionless

Because of the emittance for gray surfaces, as the case of internal wall surfaces in a space are nearly equal (0.9 to 0.95), the equation above can be linearized:

$$q_{rad} = \varepsilon \cdot \sigma \sum_{i=1}^{N} \theta_{f,i} \cdot (T_f - T_{air}) \cdot F_{Af-Ai}$$

Where:

$$\theta_{f,i} = \frac{T_f^4 - T_i^4}{T_f - T_i}, inK^3$$

The values of $\theta_{f,i}$ vary only slightly with the temperature level in normal spaces by using floor cooling and heating.

Now the equation can be written as:

$$q_{rad} = \varepsilon_f \cdot \sigma \cdot \theta \sum_{i=1}^{N} f,i \cdot (T_f - T_{air}) \cdot F_{A_f} - A_i$$

Where the constant value of 1.05 x 108 K3 is now used for $\theta_{f,i}$.

The product ε_f σ $\theta_{f,i}$ is the radiant heat exchange coefficient between floor surface and space, written as:

hrad = $\varepsilon_f \cdot \sigma \cdot \theta$

Radiant heat flux can thus be calculated by this equation:

$$q_{rad} = h_{rad} \cdot \sum_{i=1}^{N} f,i \cdot (T_f - T_{air}) \cdot F_{A_f} - A_i$$

Of the Stefan-Boltzman constant σ, two different values are given in the literature:

- IP Units, 0.1714×10^{-8} Btu/h · ft² · °R⁴ or 0.1744×10^{-8} Btu/h · ft² · °R⁴
- SI Units, 5.67 x 10-8 W/m²K⁴, (ASHRAE 1996) or 5.77 x 10-8 W/m²K⁴ (Fanger 1982).

Calculating hrad for different combinations of ε and θ shows that, with only little error, a constant value of h_{rad} = 5.5 W/m² K can be used.

	σ = 5.67×10^{-8} W/m² K⁴	σ = 5.77×10^{-8} W/m² K⁴	σ = 0.1714 x 10-8 Btu/h · ft² · °R⁴
ε=0.90	h_{rad}=5.35 W/m² K	h_{rad}=5.45 W/m² K	h_{rad}=.94 Btu/h · ft² · °F
ε=0.95	h_{rad}=5.67 W/m² K	h_{rad}=5.76 W/m² K	h_{rad}=1.00 W/h · ft² · °F

Variant Calculation of hrad Using Equation Above, θ_f = 1.05 x 108 K3

Solar Radiation

The amount of shortwave solar radiation entering a room depends upon the orientation of window(s), the properties of the glazing, the shading devices, the month and the time of day.

The calculation of the short-wave, direct-sun transmission through the window can be easily be determined with software. The amount of shortwave radiation entering the floor surface may be estimated, based on the percentage of the floor surface covered by furniture or equipment.

The specific sun radiation to the floor surface is the direct sun transmission through the window on the floor per radiated floor area, written as:

$$q_{s,rad} = \alpha \cdot \frac{Q_{directsun}}{A_f}$$

Where:

$q_{s,rad}$ = specific short-wave sun radiationper ft² (m²) upon the cooled floor, Btu/ft² (W/m²)
$Q_{directsun}$ = sun radiation through the window on the floor, Btu/h (W)
A_f = radiated floor area ft² (W/m²)
α = absorptance of the floor surface (dimensionless)

Floor Cooling

Calculating the convective heat transfer coefficient for floor cooling [Δt between 5.4°F (3 K) to 12.6 °F (7 K)] gives a range of hcon between 0.19 to 0.25 Btu/h·ft2 °F (1.1 to 1.4 W/m2 K) (ASHRAE Fundamentals) and 0.14 to 0.18 Btu/h · ft^2 ·°F (0.8 to 1.0 W/m2 K) (Recknagel/Sprenger).

The formula from Recknagel/Sprenger gives only the natural convective heat exchange factor, without consideration of the space air currents. Experimental tests have been made to measure and calculate the natural convective heat exchange coefficient (Olesen, Michel, Bonnefoi and DeCarli, 1998) with the result that a value of 0.18 Btu/h·ft^2·°F (1.0 W/m2 K) can be used. This result corresponds nearly with the variant calculation by using the literature formulas.

Approach: h_{con} = 0.18 Btu/h · ft² ·°F (1.0 W/m² K)

The space heat exchange coefficients above are a summation of hcon and h l,rad (1.0W/m² K + 5.5W/m² K = 6.5W/m² K).

The radiation heat exchange coefficient hl,rad remains fairly consistent at 0.97 Btu/h · ft² ·°F (5.5 W/m² K) when surface temperature is in the range of 59°F to 95°F (15°C to 35°C). The convective heat exchange coefficient will vary, not only due to the space and air- temperature difference, but also due to air velocity (REHVA). Hence, in floor cooling, the space heat exchange coefficient can be taken with reasonable accuracy as 0.97 + 0.26 = 1.23 Btu/h · ft² ·°F.

This simplification is physically correct if the air temperature of the space and the surrounding surface temperature are the same. For common applications like office spaces, the surface temperatures of the exterior elements (wall, window and roof) in summer are above the air temperature, while the temperature of the other surfaces are similar to the air temperature. This means for the perimeter zones, a physical higher space heat flux can be achieved. For the interior zones, the assumption of similar temperatures of air and surrounding surfaces is nearly correct.

Space Heat Flux Coefficient — Floor Systems

The idealized method of calculating the heat flux between floor surface and space requires that convective and the long-wave radiant heat flux be calculated separately. The convective heat flux is calculated using the temperature difference between floor surface and air temperature, and the long-wave radiant heat exchange is calculated using the temperature difference between floor surface and the surrounding surface temperatures. For specific calculations as well as for optimization calculation, this method should be used.

The generalized method described in this manual lends itself to the simplified use of a combined heat exchange coefficient referred to as the space heat exchange coefficient (hs). Since most comfort requirements and standards are based on the operative temperature as are some heat load calculation procedures, the operative temperature will used as the reference temperature (Olesen, Michel, Bonnefoi and DeCarli, 1998, ASHRAE 55-1992, DIN 4701).

It can be written:

$$q_s = h_s \cdot (t_f - t_o)$$

h_t = space heat exchange coefficient, Btu/h · ft² ·°F (W/m² K)
t_r = floor surface temperature, °F (°C)
t_o = operative temperature, °F (°C)

Understanding the approximation, the space heat exchange coefficient for common applications such as radiant floors in office buildings with normal ceiling height, residential buildings, and similar buildings and spaces are as follows:

Values Reflect Both Radiant and Convective Transfer	Floor		Wall		Ceiling	
	Heating	Cooling	Heating	Cooling	Heating	Cooling
Btu/h ·ft² ·°F	1.94	1.23	1.40	1.40	1.06	1.94
W/m² ·K	11	7	8	8	6	11

Source: REHVA
Nominal Surface Heat Transfer Coefficients, htot , from Various Surfaces

Flux, Radiant Floor Cooling

For example, using the recommended minimum floor surface temperature (tf-min) of 66°F (18.9°C) and a summer indoor design temperature (to) of 78°F (25.6°C), with hs equal to 1.23 Btu/h·ft² ·°F (11 W/m² · K) excluding short-wave radiation, the sensible space cooling capacity of the radiant floor in IP units becomes:

$$q_s = h_s \cdot (t_0 - t_{f-min})$$

$$q_s = 1.23 \cdot (78 - 66)$$

$$q_s = \frac{14.7\ Btu}{h\ ft^2}$$

In SI units,

$$q_s = \frac{46.3\ W}{m^2}$$

The maximum space cooling capacity of a radiant floor cooling system can be stated as 14.7 Btu/h/ft² (46.3 W/m²) when using an operative design temperature of 78°F (25.6°C).

	Surface Temperature	
	Maximum Heating	Minimum Cooling
Floor occupied zone	84°F (28.9°C)	66°F (18.9°C)
Floor perimeter zone	95°F (35°C)	66°F (18.9°C)
Wall	104°F (40°C)	62°F (16.7°C)
Ceiling	80°F (26.7°C)	62°F (16.7°C)

Maximum Heating/Minimum Cooling Temperatures

Sensible Capacity Calculation

Calculate total sensible cooling capacity of a radiant-cooled floor surface with hs equal to 1.23 Btu/h · ft² ·°F (11 W/m² ·K) and given the following room conditions:

Room Area (A): 2,000 ft²
Surface Temperature (t_f): 66°F (18.9°C)
Operative Temperature (t_o): 76°F (24.4°C)

$q = h_s \cdot A(t_o - t_f)$
$q = 1.23 \cdot 2{,}000\ (76\text{-}66)$
$q = 24{,}600$ Btu/h, in SI units, $q = 7.21$ kW

2. Designing zones, Controls & Piping

Zoning the System

As with any type of All-and-Water system, there are a number of factors to consider when zoning a radiant heating and cooling system. Variances in envelope load, internal load, occupancy and schedule are important considerations. The level of control desired also needs to be assessed against the cost of the associated controls.

With radiant heating and cooling systems, the control system is used to manage the slab temperature by controlling both the temperature and the flow of the water circulating in the loops. A system can have single or multiple water temperature zones, where water temperature is controlled. Each water temperature zone can have single or multiple local zones, where water flow is controlled. This system is analogous to a variable air volume (VAV) system. The chilled water control valve for the air handler is modulated to maintain a setpoint discharge air temperature; with local room thermostats controlling the VAV boxes to hold target space temperatures.

Water Temperature Zones

For each water temperature zone, a series of sensors (space temperature, operative temperature, relative humidity and slab temperature) is used to evaluate space conditions to determine the optimum target supply water temperature for the zone. A set of control valves (two- way or three-way) and pumps then modulate to maintain that ideal water temperature in the loop.

Base and Peak (Trim) Loads

In some applications for multiple occupancies and uses, it serves to use a single fluid temperature for a floor, parts of the building or the entire building; and to designate the fluid flow to the radiant system for the base sensible load. The ventilation system can then be fitted with the necessary primary coils and secondary zone coils to condition the air for control of latent loads and secondary (peak) sensible loads.

Local Zones

A building zone served by a target fluid temperature can be subdivided into multiple space zones, where room thermostats can provide additional occupancy and room use control. This is accomplished either by controlling the flow through stand-alone control valves or the flow through one or more loops connected to a zoned distribution manifold

Determining the Load Requirement

As with any system design, determining the heating and cooling load requirements of the building is one of the first steps to correctly size the system. There are a variety of tools available to determine heating and cooling loads. Regardless of the method used, it is important to identify the following components that comprise the total load:

- Total heating load
- Sensible cooling load, including direct solar load
- Latent cooling load

Sensible Cooling Load

As we discussed in our eBook, radiant cooling systems can manage all or part of the sensible cooling load. Average capacities are between 12 and 14 Btu/h/ft2 for bare concrete installations. Loads exceeding this base capacity will require supplemental systems such as chilled ceilings to handle the trim loads.

Latent Cooling Load

Since the radiant cooling is tasked exclusively to sensible loads, parallel schemes are required for latent loads and humidity control. These methods are typically part of the ventilation strategy such as found in the use of dedicated outdoor air systems (DOAS). Working together, these comprise the hybrid Air-and-Water system that is so effective in humid climates.

Direct Solar Load

In areas with high direct solar load, a radiant cooling system capacity can increase up to 25 to 30 Btu/h/ft^2 (78.9 to 94.6 W/m^2).

By understanding each of these components in the Air-and-Water system, one can properly design the radiant system to maximize its effectiveness with base building loads, while sizing the secondary system to handle latent, ventilation and trim loads.

Multiple Water Temperature Zones

Q: "If a radiant cooling system is used for the base load, while the secondary system is used to take care of the balance of load, when would multiple water temperature zones ever be necessary?"

A: "In many cases, a single water temperature zone is adequate. However, there are cases where having multiple water temperature zones is beneficial. For example, if a multistory office building has different tenants, it may make sense to have each tenant (or each floor) zoned as separate systems, each with their own water temperature.

This way, tenants (or floors) can be isolated if needed. Also, if one zone has a much higher indoor relative humidity (due to process loads or occupancy), the water temperature may be limited to prevent surface condensation. If the system had a single water temperature zone, then the entire building would be limited due to this isolated condition."

Load Analysis Example 1

Estimate the radiant heating and cooling capacities of a system, given the following parameters:

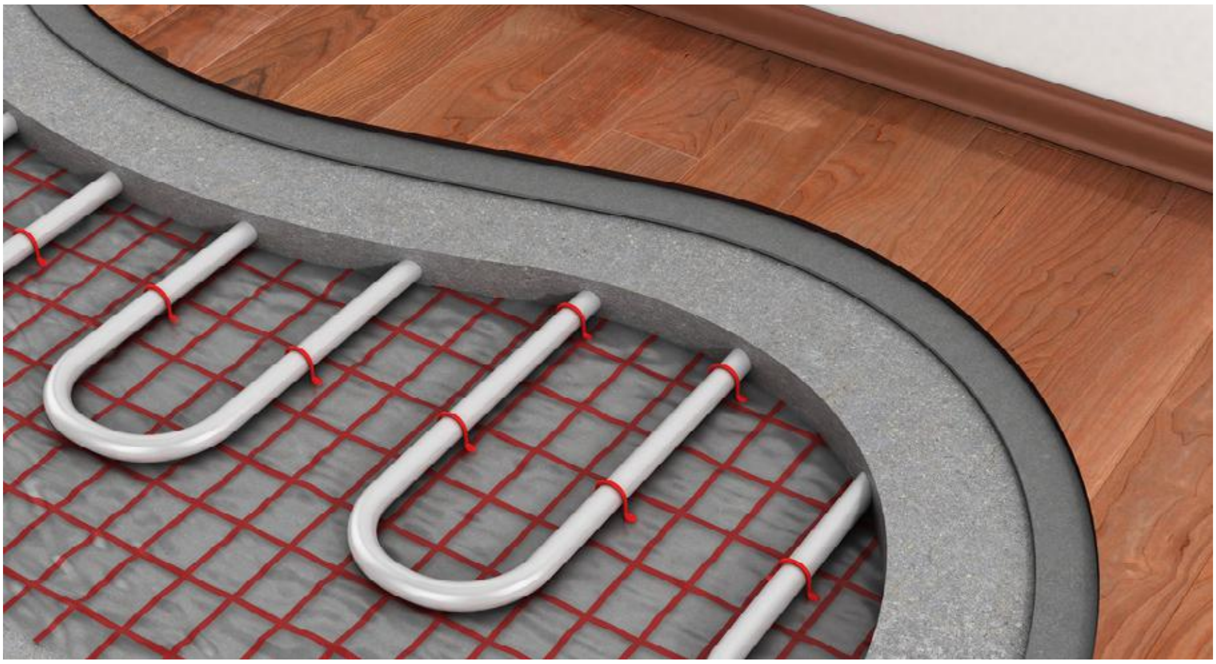
Room Conditions

Room Area: 5,000 ft^2 (464.5 m^2) (bare concrete floor)
Total Cooling Load: 141,175 Btu/h (44.8 kW)
Sensible Cooling Load: 120,000 Btu/h (38.0 kW)
Latent Cooling Load: 21, 175 Btu/h (6.7 kW)
Total Heating Load: 150,000 Btu/h 47.5 (kW)
Note: Direct solar load does not make up a significant portion of the total sensible cooling load.

Depending on entering air conditions, this space would require an air-based system with a nominal capacity of 12.5 tons.

Slab Conditions

Active Floor Area: 5,000 ft^2 (464.5 m2)
Cooling Operative Temperature: 77°F (25°C)
Cooling Slab Temperature: 66°F (18.9°C)
Heating Operative Temperature 68°F (20°C)
Heating Slab Temperature 84°F (28.9°C)



Based on These Parameters

Sensible Cooling Capacity (IP Units)

$q_c = h_s \cdot A(t_o - t_f)$
$q_c = 1.23\ Btu/h \cdot ft^2 \cdot °F \times 5{,}000\ ft^2\ (77°F - 66°F)$
$q_c = 67{,}500\ Btu/h$, in SI units 19.82 kW

Heating Capacity (IP Units)

$\mathbf{q_h} = h_s \cdot A(t_f - t_o)$
$\mathbf{q_h} = 1.94\ \text{Btu/h} \cdot \text{ft}^2 \cdot {}^\circ\text{F} \times 5{,}000\ \text{ft}^2\ (84^\circ\text{F} - 68^\circ\text{F})$
$\mathbf{q_h} = 155{,}200$ Btu/h, in SI units 45.49 kW

Note: while the radiant floor can adequately manage the total heating load, a supplemental system will be required to handle the balance of the cooling loads.

Sensible Load: 120,000 Btu/h (38.0 kW)
Radiant Floor Sensible Capacity: 67,650 Btu/h (21.4 kW)
Suplemental System Capacity: 120,000 - 67,650 Btu/h = 52,530 Btu/h (16.6 kW)

Therefore, the supplemetal system will be required to provide
Total Cooling Load: 73,525 Btu/h (23.3 kW)
Sensible Cooling Load: 52,350 Btu/h (16.6 kW)
Latent Cooling Load: 21,175 Btu/h (6.7 kW)
Total Heating Load: 0 Btu/h (0 kW)

Depending on entering air conditions, this space would require a supplemental air-based system with a nominal capacity of 7.5 tons.

Control System

The control system is a critical component of any high-efficiency building system. The radiant control system continuously monitors indoor temperature and relative humidity to determine the optimal target supply water temperature for maximizing the system's performance while ensuring that condensation never forms on the slab.

Over the years there have been many different control solutions for embedded systems; most have been for heating. The advent of cooling systems has brought different approaches. The control systems described in this section can be used for both heating and cooling.

The control system should be designed to make efficient use of energy while avoiding overheating or overcooling the building. This should include keeping distribution losses as low as possible, e.g. reducing flow temperature when normal comfort temperature level is not required.

To maintain the stable thermal environment, the control system is required to manage the heat balance between enclosures losses/gain and the All-and-Water system under transient occupant and outdoor climate conditions.

Without a doubt, the performance of the enclosure plays a major role in stabilizing the inherent indoor climate oscillations due to transitory conditions. When the building can solve a significant portion of the heating and cooling load it, enables the Air-and-Water system to work within a considerably more stable indoor environment. Cooled surfaces further stabilise the indoor climate by absorbing uncontrolled and less predictable shortwave and longwave energy while shading and enclosure strategies can manage periodic and predictable solar radiation.

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:

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

The control system classification is based on performance level:

- Manual – the supply to the cooled space is only controlled by a manually operated device
- Automatic – a suitable system or device automatically controls cooled water to the cooled spaces
- Timing function – cooled water supplied to cooled space is shut off or reduced during scheduled periods, e.g. night setback
- Advanced timing function – cooled water supplied to the cooled space is shut off or reduced during scheduled periods, e.g. daytime with more expensive electricity tariff. Re-starting of the supply is optimised based on various considerations, including reduction of energy use.

Decades 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, even simpler, the system could be turned on or shut off. Today, automatic controls are used everywhere and have in the last decade developed significantly (e.g., fuzzy logic, wireless data transmission, introduction of protocols for data communication, etc.).

For a floor cooling system, the control is normally split up in a central control and an individual room control. The central control will evaluate the outside climate (based on the heating/cooling curve, which is influenced by building mass, heat loss, and differences in cooling required by the individual rooms) and 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 setpoint selected by the user.

Control Strategies

Central Control

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 outside and/or indoor temperatures. This is more directly related to the heat flux into the space. If during the cooling period, for example, the internal space temperature rises due to occupant, lighting and equipment loads, the cooling output from the floor system will increase and the return temperature will decrease.

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 input to the space and will provide better energy performance than controlling the supply water temperature. If the cooling system is operated intermittently (night and/or weekend set-back) the central control is also important for providing adequately cool water temperatures during the precooling period in the morning.

Zone or Local Control

The installation of individual room temperature controls is recommended to improve comfort and potential energy savings. It is also essential for the thermal comfort of the occupants that they have individual control adjustment of the room temperature setpoint from room to room.

The influence of the individual room control strategy for floor cooling has shown a 15 to 30% energy savings by using an individual room control compared to central control only.

Control of Thermo Active Building Systems (TABS)

Thermo Active Building Systems (TABS) engage the entire concrete mass as a thermal battery using chilled or heated water to charge the system. Specifically for TABS, individual room control using the floor is not practical. However, a zoning strategy such as northside/southside or compass quadrants is suggested for cases where supply water temperature, average water temperature or the flow rate may differ from zone to zone.

Relatively small temperature differences between the cooled surface and the space are typical for TABS. This matter results in a significant degree of self control. In specific cases with low cooling loads, a concrete slab can be controlled at a constant core (water) temperature year round. If, for example, the core is kept at 72°F (22.2°C), then the slab will be space heating when room temperatures are below 72°F (22.2°C) and space cooling when room temperatures are above 72°F (22.2°C).

Condensation Control

An effective Air-and-Water system in any building in South East Asia must provide effective humidity management, including dew point control for microbial control over pathogens and allergens. This promotes respiratory and thermal comfort for occupants as well as dimensional stability in hygroscopic materials like wood.

When humidity is managed to enhance the indoor climate for health of the environment and, by association, health of the occupants, and for the dimensional stability of architectural materials, it enables radiant cooling systems to operate at their peak capacity within acceptable thermal comfort parameters.

Cooling control systems should set lower limits for supply fluid temperature and limit surface temperatures for comfort to 62°F (16.7°C) and 66°F (18.9°C) for walls or ceilings and floors, respectively (ASHRAE Standard 55-2010).

The indoor operative temperature should also be controlled so the standard required temperature for Class A systems (as per ISO 7730) corresponding to 10% PPD is maintained within the range specified by ASHRAE Standard 55-2010. This can be achieved with control I/O logic using if/then statements on feedback from space conditions; including instruction to terminate fluid flow to a cooled surface until space humidity and surface temperatures return to acceptable conditions.

Dew Point Control

The surface temperature at any point on the floor is location dependant on the fin efficiency of the slab. Fin efficiency being the thermal characteristics due to the log mean temperature differential (LMTD), tube diameter, spacing and depth, and conductivities of the materials in the slab. The dew point calculation and location for surface sensor placement should be where the coldest temperature ought to exist for acceptable control.

For radiant cooled, naturally ventilated spaces, bin-based climate data must be used for a thorough assessment of potential space conditions imposed on the conditioned surface.

Interior Zones

It is likely the interior zones of larger spaces may only need cooling. In this case, the EWT can be fixed and the flow varied through control of valves or circulators based on room temperature set-points.

During unoccupied times, there is no benefit in pulling the slab temperature down lower than what is necessary to compensate for the next occupied load. Control strategies should consider terminating flow to the slab when the core temperature of the slab has been reached.

Perimeter Zones

Perimeter zones may be equipped with a radiant floor system that provides both heating and cooling to the space. The EWT may be controlled by a weather-compensation control combined with individual occupant controlled zones. In cooling mode, the EWT should be kept constant. 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.

Controlling the Water Temperature Using Indoor Adaptive Reset

A combination radiant heating (if necessary in the winter) and cooling system can be effectively controlled using an indoor adaptive reset strategy. This strategy determines the ideal target water temperature by assessing the space conditions (temperature, operative temperature and relative humidity), the water temperature (supply and return) and the slab temperature. The control system then continually adjusts the target water temperature based on the rate at which the space temperature changes to maximize the effectiveness of the slab while ensuring that the surface temperature never reaches dew point or gets too cold or too hot.

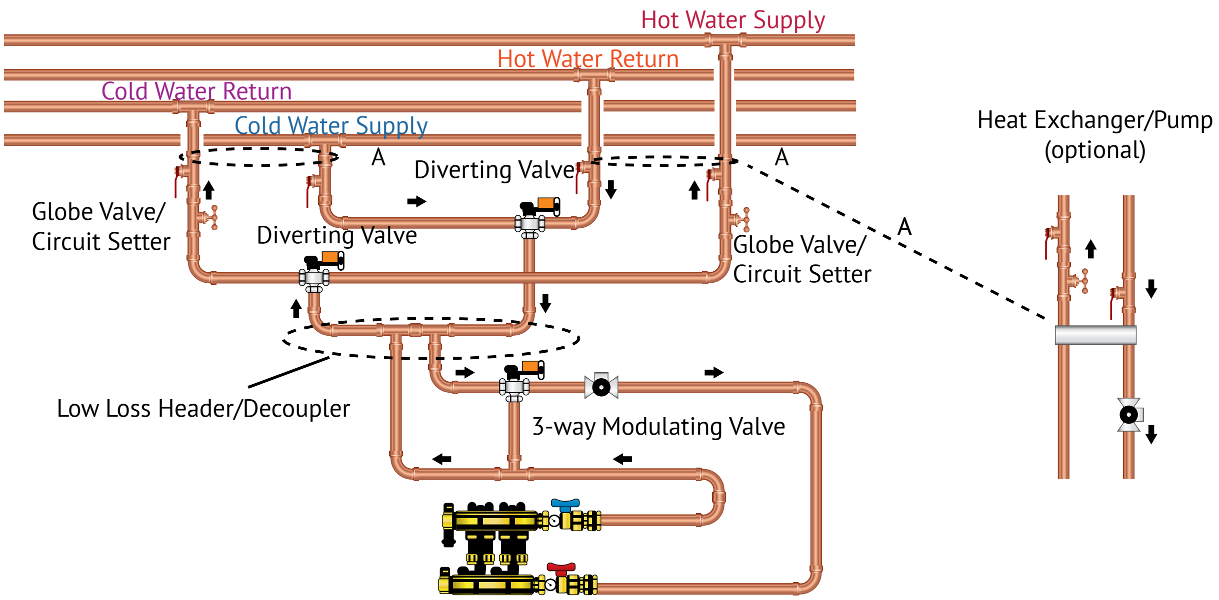
How Air-and-Water Systems Control Integration with the Air-side System

The radiant cooling slab is able to effectively handle all or a portion of the building's sensible load. A supplemental system — such as an air handling unit or dedicated outside air system (DOAS) — is assigned to handle the balance of the sensible load, the latent load and the ventilation load. It is important to control these systems together so that they do not function in opposite modes of operation. A common strategy is to operate the radiant system as an offset to room setpoint, using the radiant slab to handle the base sensible load and relying on the air-side system to manage the trim loads.

Using this two-stage approach, the radiant system setpoint could be set at 76 °F (24.4°C), while the air-side system setpoint is set at 78 °F (25.6°C). If the space temperature exceeds 76 °F (24.4°C), the radiant loop starts flowing chilled water through the embedded tubing to control the slab and space temperatures. As the load increases, the flow rate and supply water temperature would adjust accordingly until the system is operating at maximum capacity. If the space temperature continues to increase and exceeds 78 °F (25.6°C), the air-side system comes on to handle the rest of the load.

Piping Strategy for Mixed Water Temperature Control

For each water temperature zone, there are a variety of piping strategies that can be used to deliver the target supply water temperature for combination radiant floor cooling and heating applications. It is important to note that with each of these strategies, the assemblies shown in the diagrams below are necessary for each mixed water temperature, not necessarily for each local zone or manifold. The manifold shown in the diagram can represent a single or multiple manifolds in the system.



Cooling/Heating Switchover with Diverting Valves

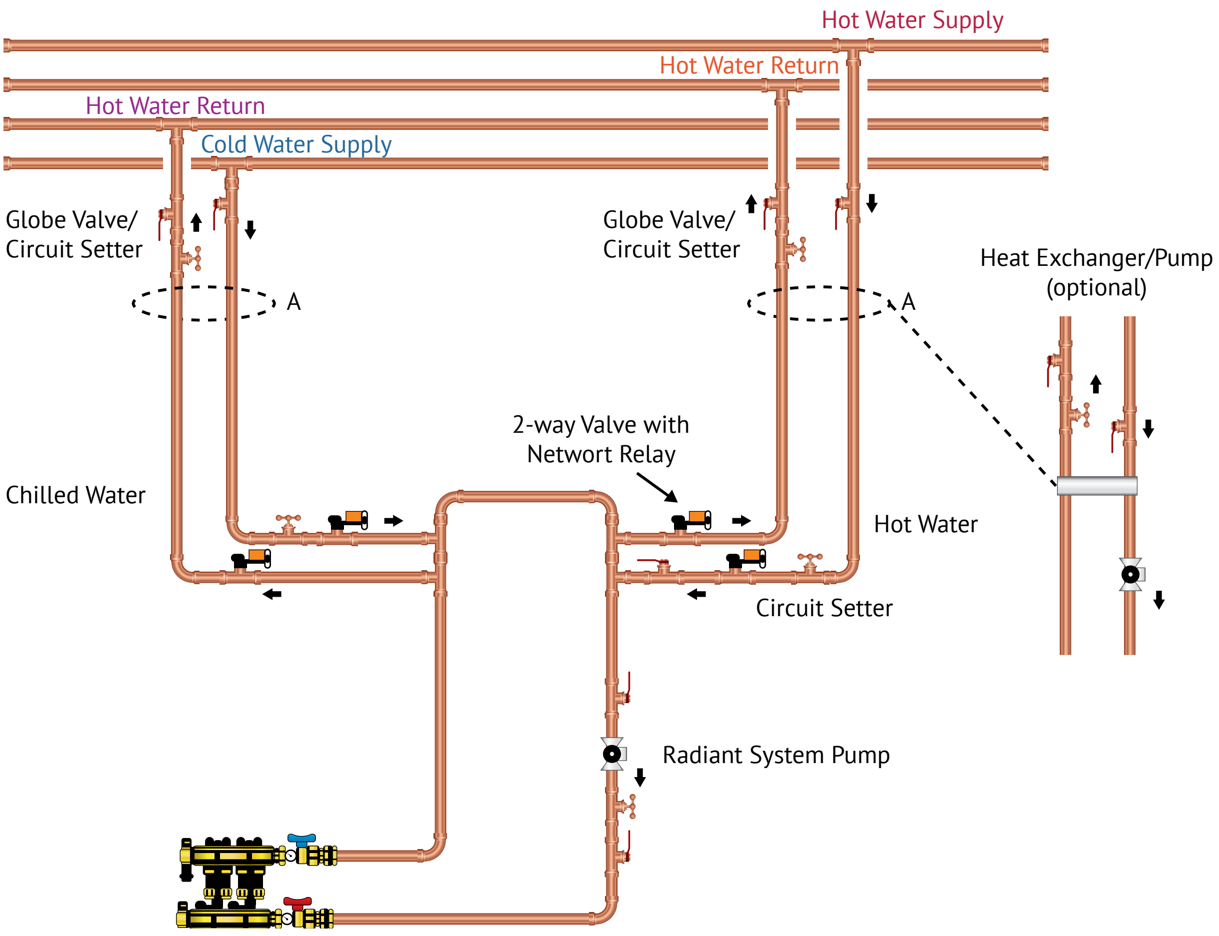
This arrangement requires the following components:

1. **Two (2) three-way diverting valves**
2. **One (1) three-way modulating mixing valve**
3. **One (1) radiant system circulator**

Heat exchangers can be installed if the system pressure is high or to ensure that the heating hot water and chilled water sources do not mix.

With this arrangement, the supply water temperature is controlled as follows:

1. Upon a call for cooling, the supply and return diverting valves open to the chilled water ports.
2. The radiant circulator starts.
3. The control system determines the optimum target water temperature based on space temperature, indoor relative humidity, calculated operative temperature and slab temperature.
4. The control system controls the three-way modulating mixing valve to the target loop temperature.
5. Upon a call for heating, the cold diverting valves are closed. A timer is engaged to delay the changeover. Once the delay period has ended, or the slab temperature has drifted up to a predetermined base temperature, the supply and return diverting valves are opened to the heating hot water ports. Target water temperature is based on space temperature, outdoor temperature, calculated operative temperature and slab temperature.



Four-pipe Injection System

This arrangement requires the following components:

1. Two (2) two-way modulating control valves [select characteristics (linear or logarithmic) based on good control valve practices]
2. Two (2) two-way solenoid valves or on/off zone valves with <45 second closure
3. One (1) radiant system circulator

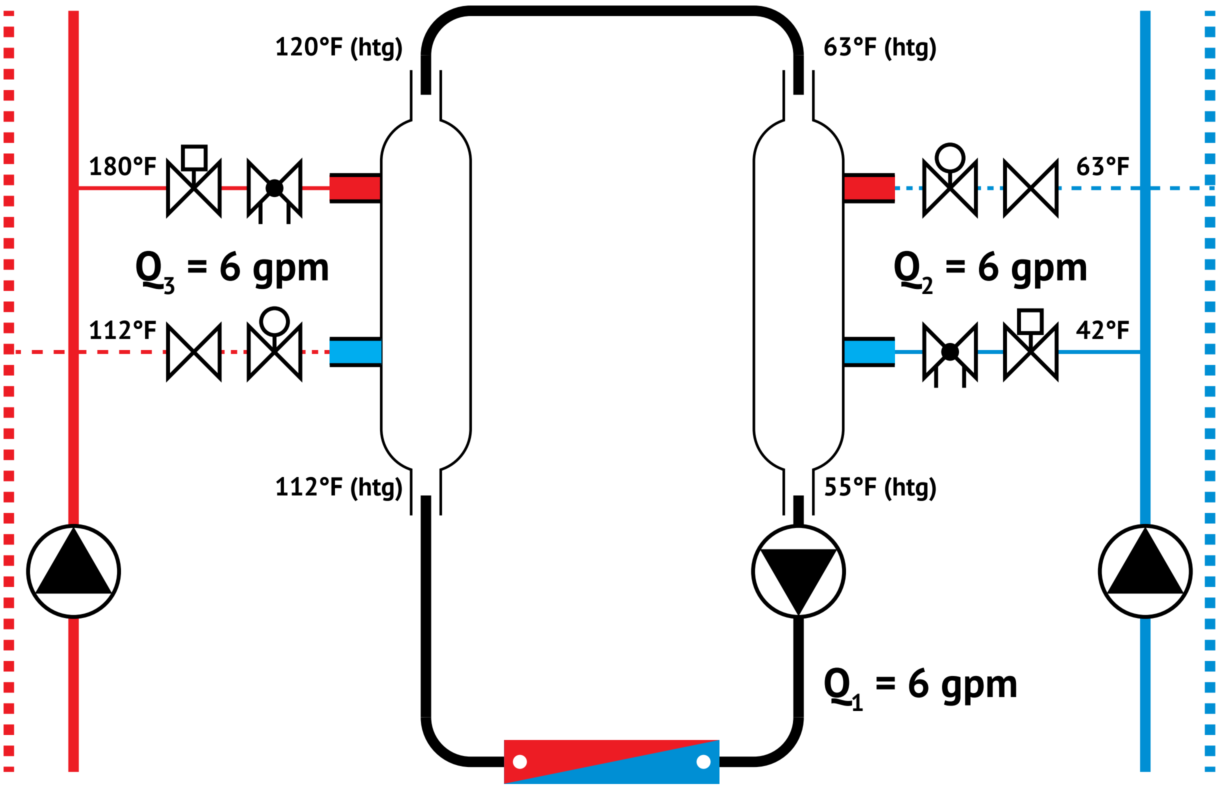
Heat exchangers can be installed if the system pressure is high or to ensure that the heating hot water and chilled water sources do not mix.

With this arrangement, the supply water temperature is controlled as follows:

1. Upon a call for cooling, the control system determines the optimum target water temperature based on space temperature, indoor relative humidity, calculated operative temperature and slab temperature.
2. The radiant circulator is then ramped up or turned on.
3. The chilled water solenoid valve fully opens while the modulating control valve is regulated to inject sufficient chilled water into the radiant system to maintain target water temperatures. Upon a call for heating, the cooling solenoid valve and associated modulating control valve are allowed to close. A timer is engaged to delay the changeover. Once the delay period has ended, or the slab temperature has drifted down to the base temperature, the heating water solenoid valve fully opens while its associated modulating control valve is regulated to inject sufficient heated water into the radiant system to maintain target water temperature.

Temperature Differences and Flow Rates Temperature Differences and Flow Rates

Using the layout shown in Figure 3-3, we can first start by dimensioning the flow through the embedded system. The radiant circulator flow is generally determined for cooling as this is more critical. Typical supply and return temperatures through the floor are 55°F (12.8°C) and 63°F (17.2°C) so from these values and the load, the flow rate can be determined. The flow rate through the embedded system is constant volume. Variable volume systems can also be used.



Flow in radiant system

Q_1 = q, Btu/hr / ((500 x (t_1 - t_2))

Q_1 = 200,000 Btu/hr / ((500 x (63°F - 55°F))

Q_1 = 500 gpm

Cooling injection flow

Q_2 = q, Btu/hr / ((500 x (t_1 - t_2))

Q_2 = 200,000 Btu/hr / ((500 x (63°F - 42°F))

Q_2 = 19 gpm

Heating injection flow

Q_2 = q, Btu/hr / ((500 x (t_1 - t_2))

Q_2 = 200,000 Btu/hr / ((500 x (180°F - 112°F))

Q_2 = 6 gpm

To illustrate this further we will assume the embedded system will have a cooling capacity of 200,000 Btu/hr (63.4 kW).

Assuming a supply water temperature of 55°F (12.8°C) and a return water temperature of 63°F (17.2°C), the flow rate through the embedded system will be 50 gpm (3.15 L/s).

This is the constant volume flow through the embedded system. There are both heating and cooling connections to the control loop.

Assuming the floor is in full design cooling capacity, then the return water from the embedded system will be 63°F (17.2°C) and this will have to be cooled to 55°F (12.8°C) 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 (5.6°C). The amount of cooling required from the primary chilled water is the same as the cooling output from the embedded system. The temperature differential through the mixing pipe is a 63°F (17.2°C) return from the loop and a 42°F (5.6°C) supply so the Delta T is 21°F (63°F - 42°F). The flow rate through the primary chilled water connection to the mixing pipe is 19 gpm (1.2 L/s).

By keeping the primary flow rate lower than the radiant flow rate, the primary connections are also smaller. This enables the two-way control valve in the chilled water connection to have an improved Cvs (kvs) value and authority.

Another reason for this design option is that the primary chilled water flow rate is much lower than the radiant flow rate. This limits the amount of 42°F (5.6°C) water required for the floor.

In practice, the supply temperature (ts) and return temperature (tr) are established from the average temperature (tavg). This comes from the design of the radiant panel based on its fin efficiency which includes tube diameter, spacing, depth, and conductivities of the slab and variuos boundatry layers.

$t_s = t_{avg} + \Delta t/2$ and $t_r = t_{avg} - \Delta t/2$

Where, the differential temperature (Δt) is a selected value based on good practice. Normally for reversible floors, less than a nominal 7°F (4°C) is used. For heating-only floors, this number can be considerably higher [greater than 15°F (8°C)]. The Δt, as well as the loop depth and pattern, has an influence on the surface temperature efficacy and should be seleceted carefully for floors requiring high-quality surface temperatures.

Constant Flow, Variable Temperature

For illustrating this control concept, the water supply temperature to the floor varies, and the floor surface temperature is maintained at 68°F (20°C). As the cooling output to the space increases to maintain space setpoint temperature, the water flow through the floor is constant. For this control option, the supply water temperature to the floor varies from 61°F (16.1°C) to a minimum of 52°F (11.1°C) 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 (17.2°C).

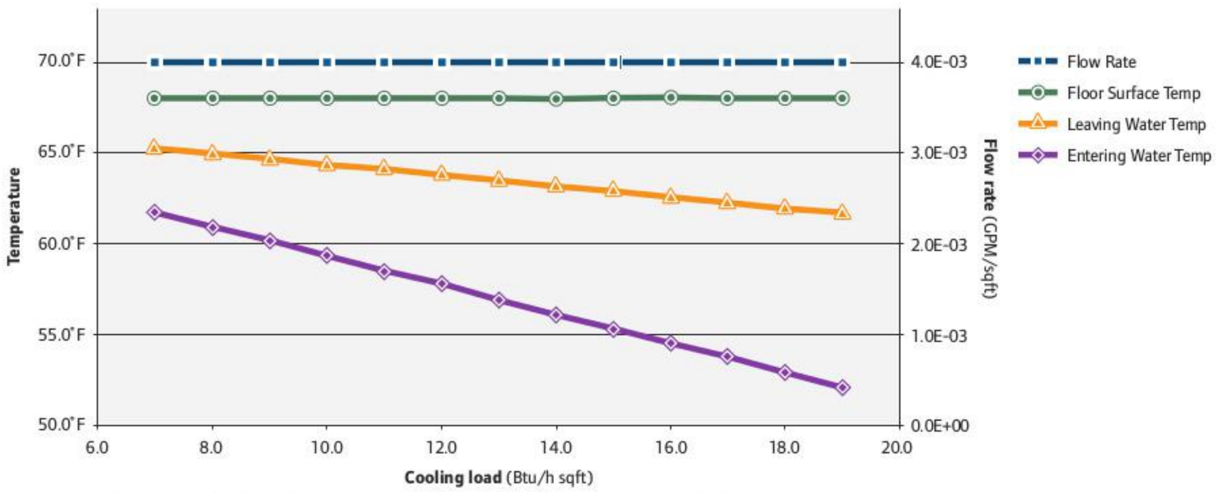


Figure 3-4: Characteristics of a Constant Flow, Variable Temperature Control

Variable Flow, Constant Temperature

For illustrating this control concept, the entering water supply temperature to the floor is kept at a constant 54°F (12.2°C), and the floor surface temperature is maintained at 68°F (20°C). As the cooling output to the space increases to maintain space setpoint 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 61°F (16.1°C)

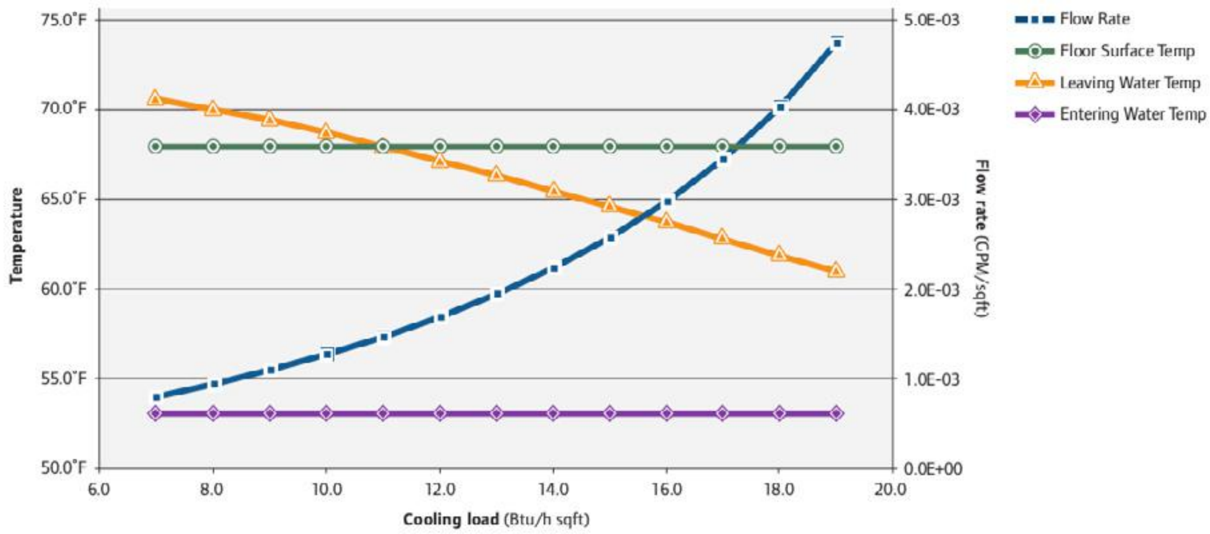
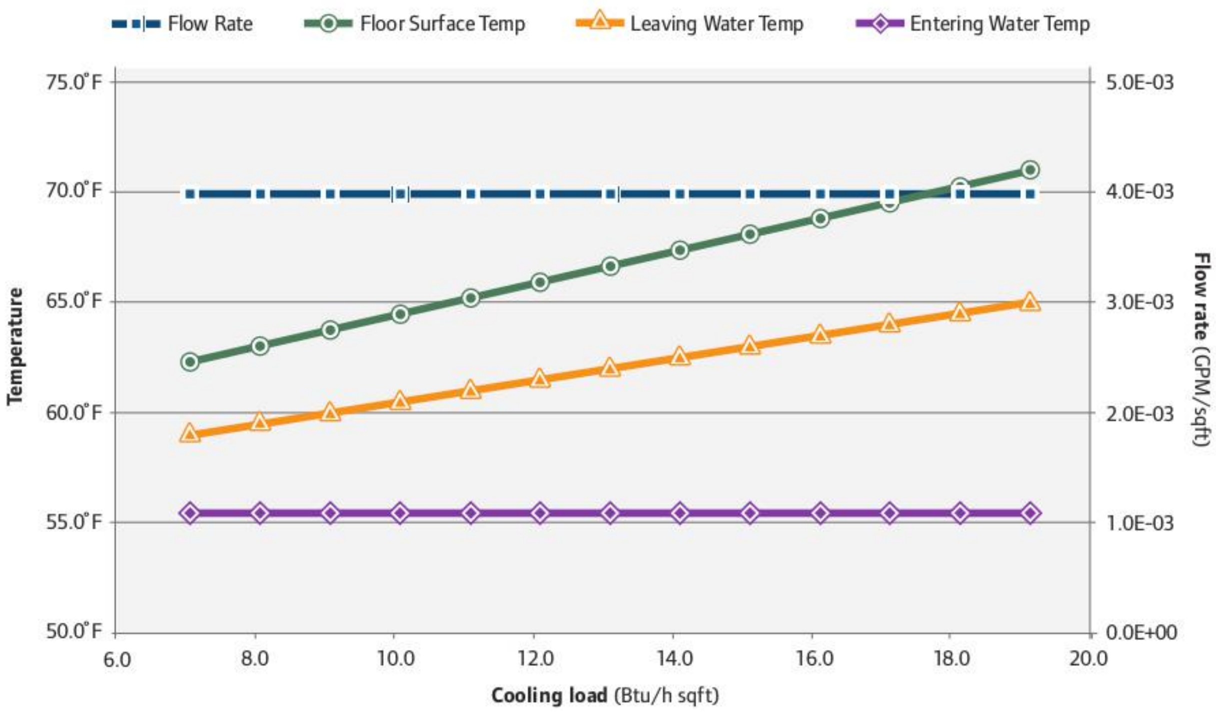


Figure 3-5: Characteristics of a Variable Flow, Constant Temperature Control

Constant Flow, Constant Temperature

For this control concept, the water supply temperature to the floor is kept at a constant 55°F (12.8°C), and the floor surface temperature varies from 62°F (16.7°C) to 71°F (21.7°C). As the cooling output to the space increases to maintain space setpoint 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.



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

Local Zone Control

As described previously, it is possible to have local zone control within each mixed water temperature zone. While the below piping arrangements will dictate target water temperature, zone valves can add local control by controlling water flow.

Depending on the size of the local zone, zone valves can be added to control a group of manifolds or a single manifold. Individual loops can be controlled with thermal actuators mounted directly on the manifold. The valves and actuators are controlled by space temperature sensors. In special cases where humidity may be a concern, local relative humidity sensors may also be used.

Uponor Motorized Valve Actuator and Zone Valve

Uponor Thermal Valve Actuator

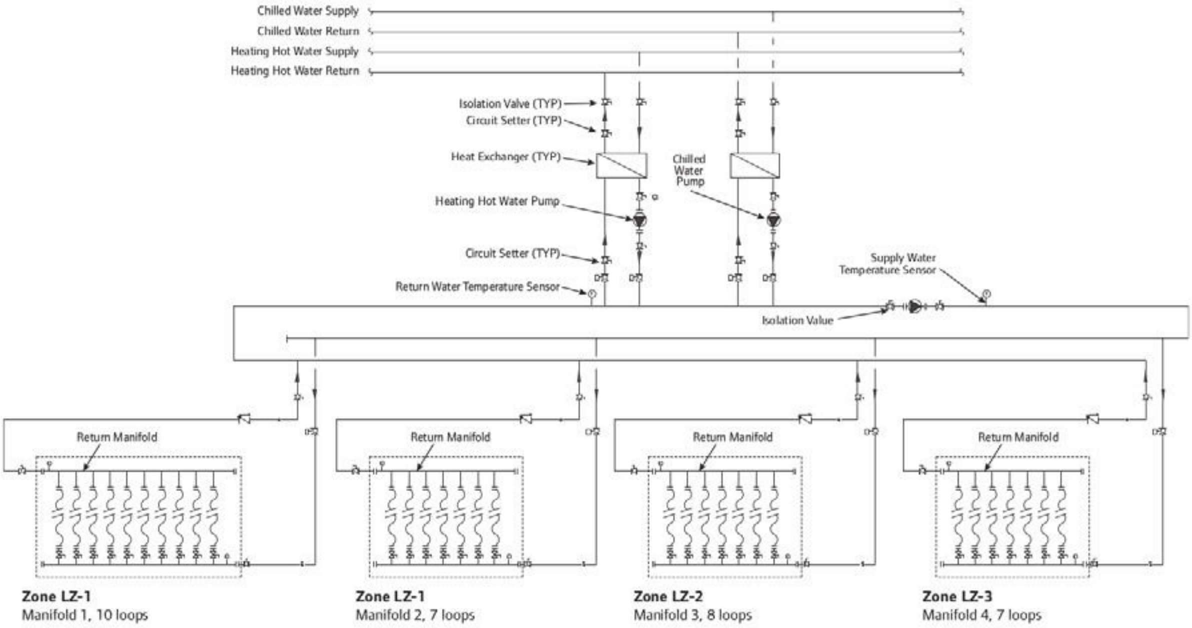


Figure 3-9: Piping Diagram Showing Four Manifolds with Zone Valves in One Water Temperature Zone

Mixing or Injection Valves

Depending on the specific design requirements, a radiant cooling system can have one or more operating temperatures or "mixed water temperatures." For each mixed water temperature, valves will be needed to reach the target supply water temperature using the main chilled water and heating hot water supply temperatures.

Temperature and Humidity Sensors

The following are temperature and humidity sensors which are required to effectively manage the space and radiant surface conditions:

- Indoor air temperature sensor
- Outside air temperature sensor
- Supply and return water temperature
- Slab temperature sensors
- Indoor relative humidity sensor

Room Thermostats/Sensors

Wireless and wired thermostats/ sensors are available as standard offerings. Regardless of the choice of temperature and humidity sensing for feedback or feedforward, the receiving controller will process "if/then" logic to control the flow entering the occupied zone via the water temperature zone by regulating valves and circulators.

The sensing should always take place to best represent the controlled element without interference from other aggravating influences. For example, outdoor sensors for cooling controls should best represent the outdoor dry bulb temperature without interference from solar gains which would incorrectly feed forward the wrong signal to the controller. Floor surface sensors should be within the near surface of the floor to represent what the occupants sense. Sensing for dew point with textile flooring should be done under the floor covering instead of on top.

Additionally, space temperature control is about providing a reading to the Air-and-Water control system which best represents what the occupants are experiencing. Therefore, the sensing should take place in the occupied area without influences from other heat sources.

Follow good practice, such as keeping wall-mounted sensors off exterior walls. Instead, place them on interior walls at a representative height for seated or standing work, and away from high-intensity heat sources, such as lamps and office equipment.

For wireless operative controls with specially designed radiant sensors, locate the control in such a place which can take advantage of both the air and surface temperature influences as well as direct solar gains. Ideally the best placement would be near the occupant and in such a position to sense what the occupant is thermally sensing.

Peak Shaving with ABS

Incorporating TABS can exploit the high thermal inertia of the slab to perform peak shaving. Peak shaving can reduce the peak in the required cooling power, making it possible to efficiently cool the building during unoccupied periods, reducing energy consumption and using lower off-peak electricity rates. This can also reduce the size of cooling system components, including the chiller.

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 3-10, the required peak cooling power needed for dehumidifying the air during day time is sufficient to cool the slab during night time.

When designing TABS, the engineer needs to determine whether or not the system capacity at a given operating water temperature is sufficient to maintain the room temperature within a given comfort range. The indoor temperature will change moderately during the day. The goal is to maintain internal conditions within the range of comfort, i.e. $-0.5 < PMV < 0.5$, according to ISO 7730 and ASHRAE Standard 55-2010. The engineer also needs to determine the heat flow on the water side to properly size the heat distribution system and the chiller/boiler.

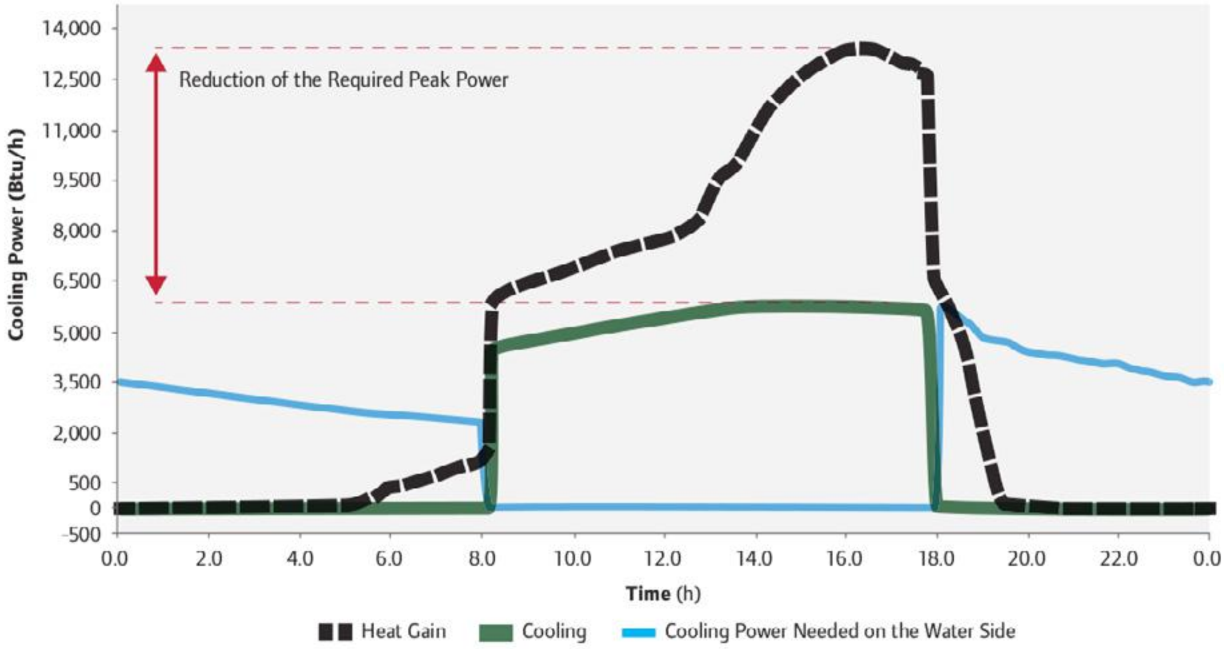


Figure 3-10: Example of Peak Shaving Effect

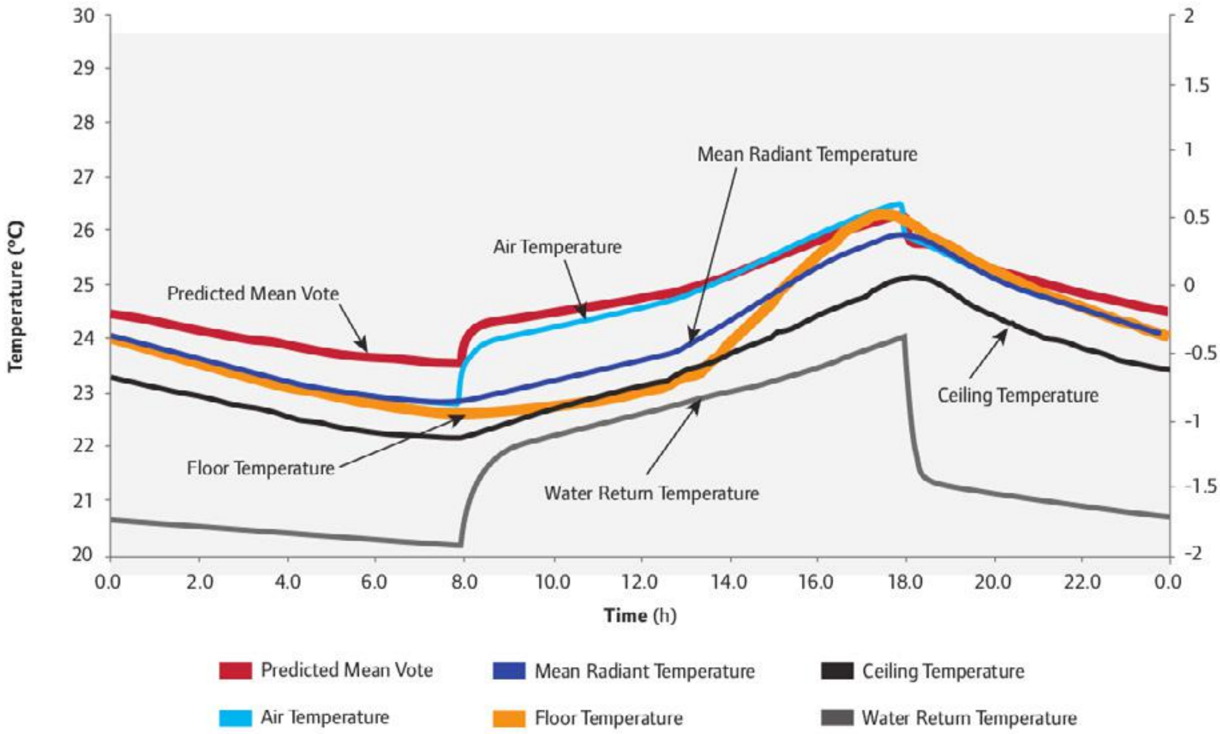


Figure 3-11: Example of Temperature Profiles and PMV Values vs. Time

Detailed building system calculation models have been developed which evaluate the heat exchanges under unsteady state conditions in a single room, the thermal and hygrometric balance of the room air, prediction of comfort conditions, condensation on surfaces, availability of control strategies and the 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. A simplified approach, as shown here, can be used to simulate thermo active building systems.

The diagrams in Figures 3-12 and 3-13 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 = 2.6, RSI = 0.46) and an allowed room temperature range of 70°F (21.1°C) to 78°F (25.6°C).

The upper diagram shows on the Y-axis the maximum permissible total heat gain in space (internal heat gains plus solar gains) [Btu/h/ft2], and on the X-axis the required water supply temperature. The lines in the diagram correspond to different operation periods (8 hours, 12 hours, 16 hours and 24 hours) and different maximum amounts of energy supplied per day [Btu/(ft2 · d)].

The lower diagram shows the cooling power [Btu/h/ft2] required on the water side (to dimension the chiller) for thermo active slabs as a function of supply water temperature and operation time. Further, the diagram shows the amount of energy rejected per day [Btu/(ft2 ·d)].

The example shows that a system with a maximum internal heat gain of 12 Btu/h/ft2 (37.9 W/m2) and 8-hour operation requires a supply water temperature of 64.8°F (18.2°C). If, instead, the system is in operation for 12 hours, the system requires a supply water temperature of 66.7°F (19.3°C). In total, the amount of energy rejected from the room is approximately 106.2 Btu/ft2 (1.2 MJ/m2) per day. In the same conditions, the required cooling power on the water side is 11.7 Btu/h/ft2 (36.9 W/m2) (for 8-hour operation) and 7.9 Btu/h/ft2 (24.9 W/m2) (for 12-hour operation), respectively. Thus, by 12-hour operation, the chiller can be much smaller.

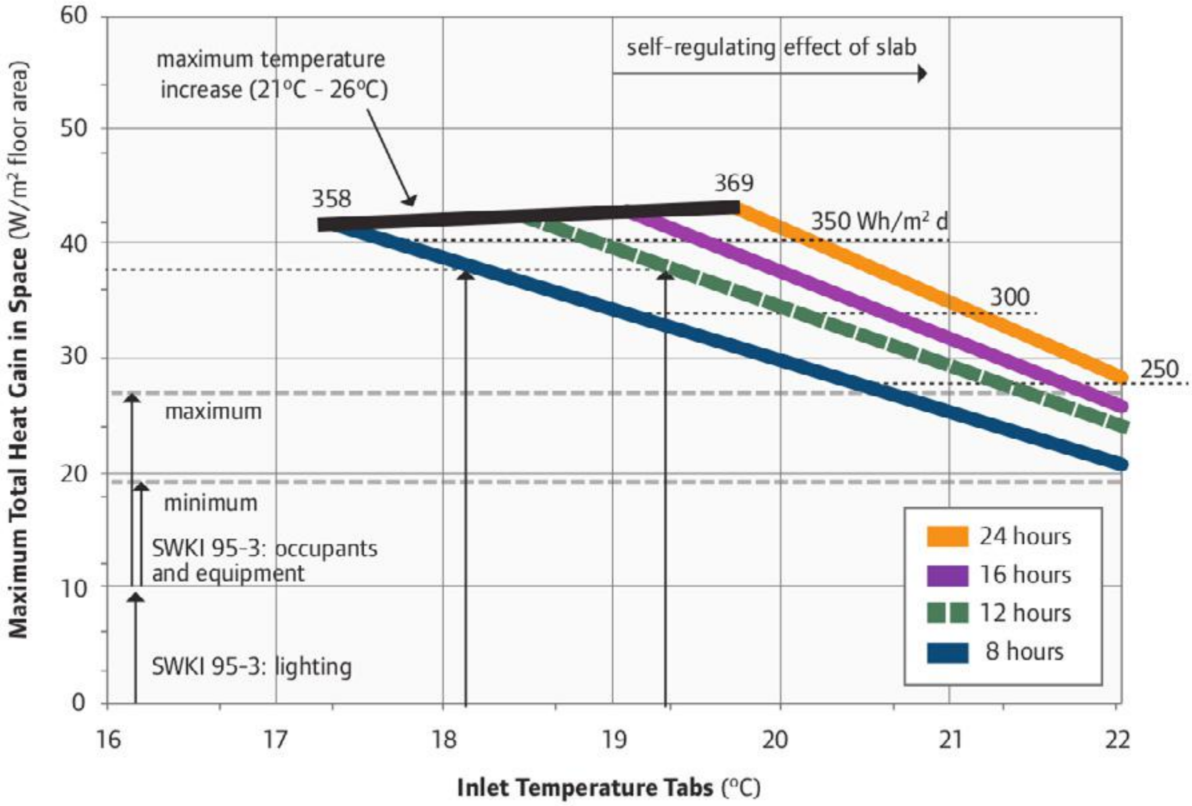


Figure 3-12: Maximum Total Heat Gain in Space

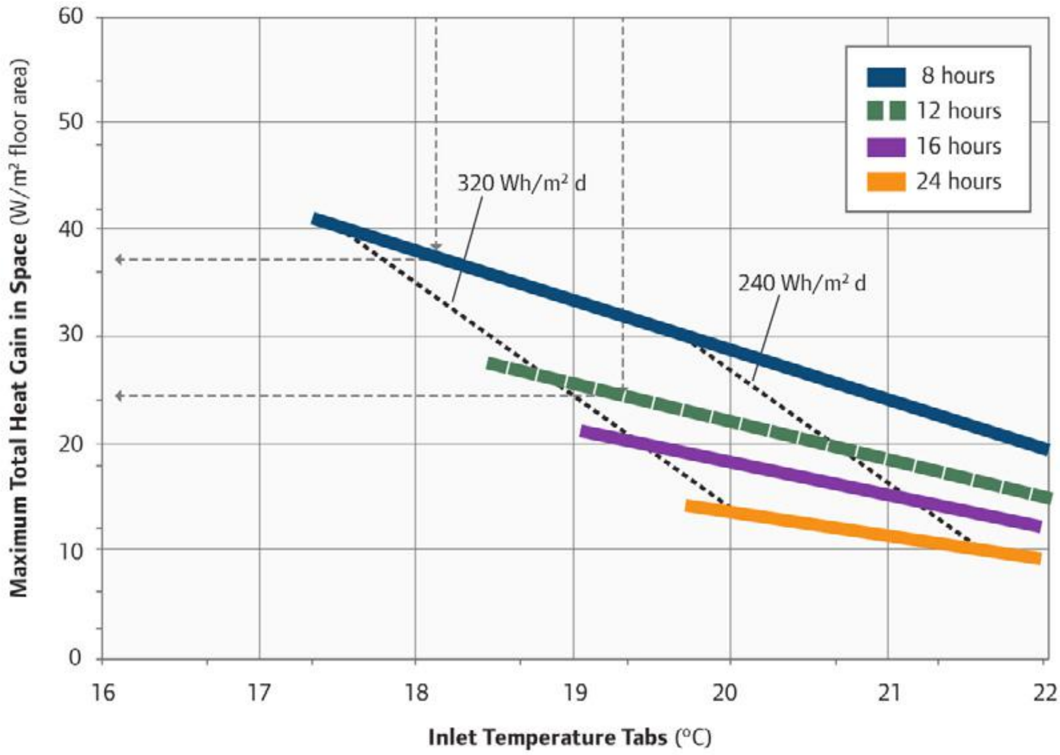


Figure 3-13: Required Cooling Power on the Water Side

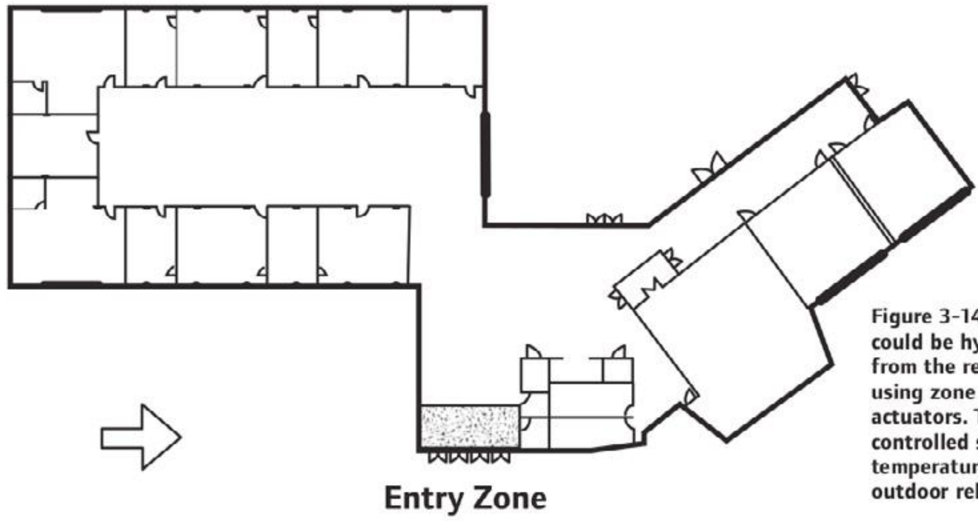


Figure 3-14: An entry area could be hydronically isolated from the rest of the system by using zone valves or thermal actuators. This zone could be controlled so the supply water temperature is reset based on outdoor relative humidity.

Peak Shaving with ABS

The control strategy described above lays the basic framework for effectively controlling the radiant heating and cooling slab. It also points out the flexibility in addressing atypical situations.

High Humidity Isolation

In addition to the ever present humidity in South East Asia, it may be necessary on some projects in some climates to isolate atypical spaces from the base load cooling system due to an inherent localized rise in humidity. Consider high-volume vestibules, water parks, kitchens, laundry or other rooms that may experience an extraordinary control condition. In these cases, it is prudent to place these types of zones on a separate space control which would limit the fluid temperature and/or flow based on feedback or feedforward sensing of temperature and humidity.

Rapid Swings in Internal Loads

The optimum conditions for an occupied building are where the enclosure has been designed around the stable and self-regulating characteristics of the radiant cooling system. Through the radiative and convective principles of heat transfer to/ from a conditioned slab, a rise in space temperature will suppress heat transfer; likewise a drop in space temperature will promote heat transfer.

The greater the performance of the enclosure, the more stable this phenomena occurs. However, in spaces that may see rapid changes to a controlled condition, it is best to respond with fast-acting systems and use the radiant system for the base load. Take for example a lecture hall, museum or hotel. In such cases, occupancy can move from vacant to fully occupied in a very short period of time.

Through the use of hybrid Air-and-Water system through packaged terminal air conditioners (PTAC) or dedicated outdoor air systems (DOAS), aggressive fluctuations in latent and sensible loads can be effectively managed while allowing the efficient and stable radiant system to operate in the background.

Condensation Control

As noted previously, moisture management is necessary for a number of reasons. Surfaces used for cooling should be kept 2°F to 3°F (1°C to 2°C) above the occupied dew point. This can be accomplished with control I/Os based on if/then statements. The feedback to the control can be based on the entering water temperature (EWT) for operational control and feedback from the floor surface for safety limit control. EWT sensing should be done at the manifold cabinet which best represents the fluid temperature before it enters the floor. Control output will be to regulate fluid temperature and flow to the cooled surface.

Bangkok International Airport, Thailand

Project data

- 150,000 m2 Uponor underfloor cooling (the approximate surface area of 20 football pitches)
- Architect: Helmut Jahn, Chicago, USA
- Completion: 2006

The largest hybrid radiant cooling installation in the world is at the Bangkok International Airport in Thailand, a city where the average relative humidity is between 70-85% year round.

Gardens by the Bay, Singapore

Project data

- 12,000 m2 Uponor radiant floor cooling for the conservatories and the concrete walkways for the visitors
- Thermal simulation by Uponor for the combination of radiant floor cooling with the A/C system
- Developer: National Parks Board, Singapore
- Completion: 2011

Want Deeper Insights Into How Air-and-Water Systems Can be Used for Commercial Building Cooling in South East Asia?

Uponor's support brochure completes the case for radiant cooling solutions in South East Asia by presenting you with:

- A summary of Air-and-Water systems
- A review of the key areas of radiant cooling use in non-residential buildings
- A closer look at the compelling data behind Uponor's Malaysia case study
- A comprehensive look at Uponor's radiant cooling systems, their benefits and applications
- More information on how Uponor supports your project

Uponor - The Radiant Cooling Experts

Uponor is a world leading provider of radiant cooling systems. With thousands of projects in service worldwide, we are the experts at designing an effective, energy-efficient solution for any application.

Experienced Team of Commercial Cooling Specialists

Uponor has an experienced team of dedicated professionals in Asia to assist the engineering and architecture community from concept to commissioning:

- Commercial sales representatives provide onsite training and education
- Design engineers provide concept and design support
- Project managers provide project coordination from concept to commissioning
- Inside technical support provides CAD drawings, specifications and submittals

How Can an Uponor Air-and-Water Cooling Solution Meet The Cooling Needs of Your Building Project in Asia?

Our specialists are on hand to walk you through just exactly how our radiant cooling solutions can meet the cooling needs of your current or future project.

In our call we will:

- Discuss your project at length
- Identify and analyze possible structural & design challenges
- Answer every question pertaining to our radiant cooling solutions
- Identify the most suitable radiant cooling system for your project