

CHAPTER 6. RADIANT HEATING AND COOLING

PANEL heating and cooling systems use temperature-controlled indoor surfaces on the floor, walls, or ceiling; temperature is maintained by circulating water, air, or electric current through a circuit embedded in or attached to the panel. A temperature-controlled surface is called a **radiant panel** if 50% or more of the design heat transfer on the temperature-controlled surface takes place by thermal radiation. Panel systems are characterized by controlled surface temperatures below 300°F. Panel systems may be combined either with a central forced-air system of one-zone, constant-temperature, constant-volume design, or with dual-duct, reheat, multizone or variable-volume systems, decentralized convective systems, or in-space fan-coil units. In **decoupled** systems, the air system provides ventilation air and meets dehumidification needs. In **hybrid** (or **load-sharing**) systems, the air system may provide significant additional capacity (e.g., where cooling loads exceed the capabilities of a radiant slab system).

This chapter covers temperature-controlled surfaces that are the primary source of sensible heating and cooling in the conditioned space. For snow-melting and freeze-protection applications, see [Chapter 51 of the 2015 ASHRAE Handbook—HVAC Applications](#). [Chapter 16](#) of this volume covers high-temperature panels over 300°F, which may be energized by gas or electricity.

1. PRINCIPLES OF RADIANT SYSTEMS

Thermal radiation (1) is transmitted at the speed of light, (2) travels in straight lines and can be reflected, (3) elevates the temperature of solid objects by absorption but does not noticeably heat the air through which it travels, and (4) is exchanged continuously between all bodies in a building environment. The rate at which thermal radiation occurs depends on the following factors:

- Temperature of the emitting surface and receiver
- Emissance of the radiating surface
- Reflectance, absorptance, and transmittance of the receiver
- View factor between the emitting and receiver surfaces (viewing angle of the occupant to the thermal radiation source)

One example of heating by thermal radiation is the feeling of warmth when standing in the sun's rays on a cool, sunny day. Some of the rays come directly from the sun and include the entire electromagnetic spectrum. Other rays are absorbed by or reflected from surrounding objects. This generates secondary rays that are a combination of the wavelength produced by the temperature of the objects and the wavelength of the reflected rays. If a cloud passes in front of the sun, there is an instant sensation of cold. This sensation is caused by the decrease in the amount of heat received from solar radiation, although there is little, if any, change in the ambient air temperature.

In buildings, thermal radiation occurs mostly in the infrared (longwave) region because the temperature of the bodies in a building environment is low. An example is the radiation heat exchange between human bodies and a cold window on an exterior wall. Another major type of radiation is shortwave radiation; the source is mainly the sun, although lighting fixtures also emit energy in the shortwave region. In this chapter, discussions about radiation thermal properties and their effects are limited to the longwave region, unless specifically indicated otherwise.

ASHRAE research project RP-876 (Lundstrom et al. 1998) concluded that surface roughness and texture have insignificant effects on thermal convection and thermal radiation, respectively. Surface longwave emissance (the ratio of the radiant heat flux emitted by a body to that emitted by a blackbody under the same conditions) for typical indoor surfaces, such as carpets, vinyl texture paint, and plastic, remained between 0.9 and 1.0 for panel surface temperatures of 86 to 131°F. For shortwave (solar) absorptance, other factors, such as color and surface roughness, play a significant role.

Thermal comfort, as defined in ASHRAE *Standard 55-2013*, is "that condition of mind which expresses satisfaction with the thermal environment." Radiant heating and cooling systems can be used to provide unique approaches to dealing with several factors affecting human thermal comfort. Maintaining correct conditions for human thermal comfort by thermal radiation is possible for even the most severe climatic conditions (Buckley 1989).

Panel heating and cooling systems modulate the thermal environment by directly controlling surface temperatures as well as indoor air temperature in an occupied space. With a properly designed system, occupants should not be aware that the environment is being heated or cooled. To provide an acceptable thermal environment to the occupants, the requirements for general and local thermal comfort must be taken into account. [Chapter 9 of the 2013 ASHRAE Handbook—Fundamentals](#) and ASHRAE *Standard 55* have more information on these requirements.

Both the air temperature and the **mean radiant temperature (MRT)** of a space should be taken into account when assessing occupant thermal comfort. The combined influence of these two temperatures is expressed as the **operative temperature**. Mean radiant temperature has a strong influence on human thermal comfort. The magnitude of the effect is slightly greater than that of air temperature at the low air velocities typical to most indoor spaces. When the temperature of surfaces comprising the building (particularly outdoor exposed walls with extensive fenestration) deviates excessively from the air temperature, convective systems sometimes have difficulty counteracting the discomfort caused by cold or hot surfaces. This is because the air temperature used for controlling the HVAC system is significantly different from the MRT, and both affect thermal comfort.

Most building materials have relatively high surface emissance and, therefore, absorb and reradiate heat from active panels. This yields an MRT that is close to the air temperature under most conditions, and thus heating and cooling panels directly lead to a more controlled, acceptable thermal comfort condition in the space.

1.1 HEAT TRANSFER

Sensible heating or cooling panels transfer heat through temperature-controlled (active) surface(s) to or from an indoor space and its enclosure surfaces by thermal radiation and natural convection.

Heat Transfer by Thermal Radiation

The basic equation for a multisurface enclosure with gray, diffuse isothermal surfaces is derived by radiosity formulation methods (see [Chapter 4 of the 2013 ASHRAE Handbook—Fundamentals](#)). This equation may be written as

$$q_r = J_p - \sum_{j=1}^n F_{pj} J_j \tag{1}$$

where

- q_r = net heat flux because of thermal radiation on active (heated or cooled) panel surface, Btu/h · ft²
- J_p = total radiosity leaving or reaching panel surface, Btu/h · ft²
- J_j = total radiosity leaving or reaching panel surface, Btu/h · ft²
- F_{pj} = radiation angle factor between panel surface and another surface in room (dimensionless)
- n = number of surfaces in room other than panel(s)

[Equation \(1\)](#) can be applied to simple and complex enclosures with varying surface temperatures and emissances. The net heat flux by thermal radiation at the panel surfaces can be determined by solving the unknown J if the number of surfaces is small. More complex enclosures require computer calculations.

Radiation angle factors can be evaluated using [Figure 15 in Chapter 4 of the 2013 ASHRAE Handbook—Fundamentals](#). Fanger (1972) shows room-related angle factors; they may also be developed from algorithms in ASHRAE's *Energy Calculations I* (1976). Several methods have been developed to simplify [Equation \(1\)](#) by reducing a multisurface enclosure to a two-surface approximation. In the **MRT method**, the thermal radiation interchange in an indoor space is modeled by assuming that the surfaces radiate to a fictitious, finite surface that has an emissance and surface temperature that gives about the same heat flux as the real multisurface case (Walton 1980). In addition, angle factors do not need to be determined in evaluating a two-surface enclosure. The MRT equation may be written as

$$q_r = \sigma F_r \left[T_p^4 - T_r^4 \right] \tag{2}$$

where

- σ = Stefan-Boltzmann constant = 0.1712 × 10⁻⁸ Btu/h · ft² · °R⁴
 - F_r = radiation exchange factor (dimensionless)
 - T_p = effective temperature of heating (cooling) panel surface, °R
 - T_r = temperature of fictitious surface (unheated or uncooled), °R
- The temperature of the fictitious surface is given by an area emissance weighted average of all surfaces other than the panel(s):

$$T_r = \frac{\sum_{j=p}^n A_j \epsilon_j T_j}{\sum_{j=p}^n A_j \epsilon_j} \tag{3}$$

where

- A_j = area of surfaces other than panels, ft²
- ϵ_j = thermal emissance of surfaces other than panel(s) (dimensionless)

When the surface emissances of an enclosure are nearly equal, and surfaces directly exposed to the panel are marginally unheated (uncooled), then [Equation \(3\)](#) becomes the area-weighted average unheated (uncooled) temperature (AUST) of such surfaces exposed to the panels. Therefore, any unheated (uncooled) surface in the same plane with the panel is not accounted for by AUST. For example, if only part of the floor is heated, the remainder of the floor is not included in the calculation of AUST, unless it is observed by other panels in the ceiling or wall.

The radiation interchange factor for two-surface radiation heat exchange is given by the Hottel equation:

$$F_r = \frac{1}{\frac{1}{F_{p-r}} + \left(\frac{1}{\epsilon_p} - 1 \right) + \frac{A_p}{A_r} \left(\frac{1}{\epsilon_r} - 1 \right)} \tag{4}$$

where

- F_{p-r} = radiation angle factor from panel to fictitious surface (1.0 for flat panel)
- A_p, A_r = area of panel surface and fictitious surface, respectively
- ϵ_p, ϵ_r = thermal emissance of panel surface and fictitious surface, respectively (dimensionless)

In practice, the thermal emissance ϵ_p of nonmetallic or painted metal nonreflecting surfaces is about 0.9. When this emissance is used in [Equation \(4\)](#), the radiation exchange factor F_r is about 0.87 for most indoor spaces. Substituting this value in [Equation \(2\)](#), σF_r becomes 0.15 × 10⁻⁸. Min et al. (1956) showed that this constant was 0.152 × 10⁻⁸ in their test room. Then the equation for heat flux from thermal radiation for panel heating and cooling becomes approximately

$$q_r = 0.15 \times 10^{-8} [(t_p + 459.67)^4 - (\text{AUST} + 459.67)^4] \tag{5}$$

where

- t_p = effective panel surface temperature, °F
- AUST = area-weighted temperature of all indoor surfaces of walls, ceiling, floor, windows, doors, etc. (excluding active panel surfaces), °F

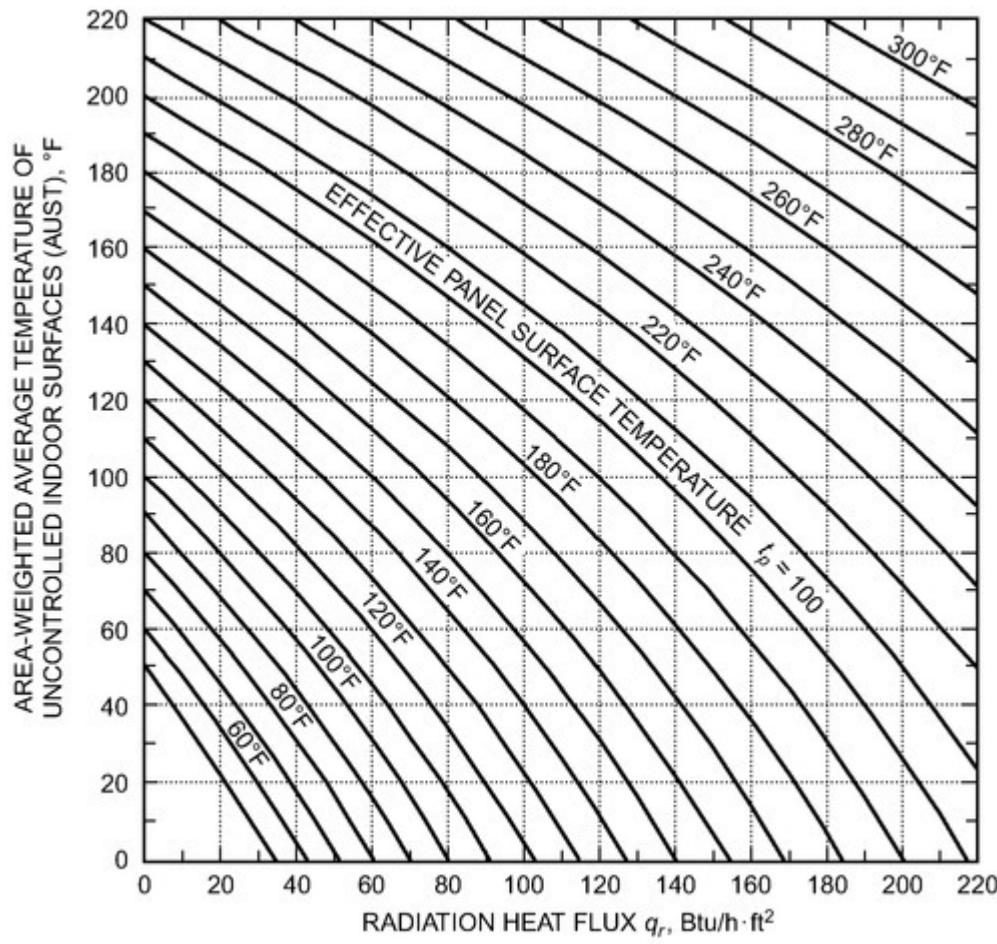


Figure 1. Radiation Heat Flux at Heated Ceiling, Floor, or Wall Panel Surfaces

[Equation \(5\)](#) establishes the general sign convention for this chapter, which states that heating by the panel is positive and cooling by the panel is negative.

Radiation exchange calculated from [Equation \(5\)](#) is given in [Figure 1](#). The values apply to ceiling, floor, or wall panel output. Radiation removed by a cooling panel for a range of normally encountered temperatures is given in [Figure 2](#).

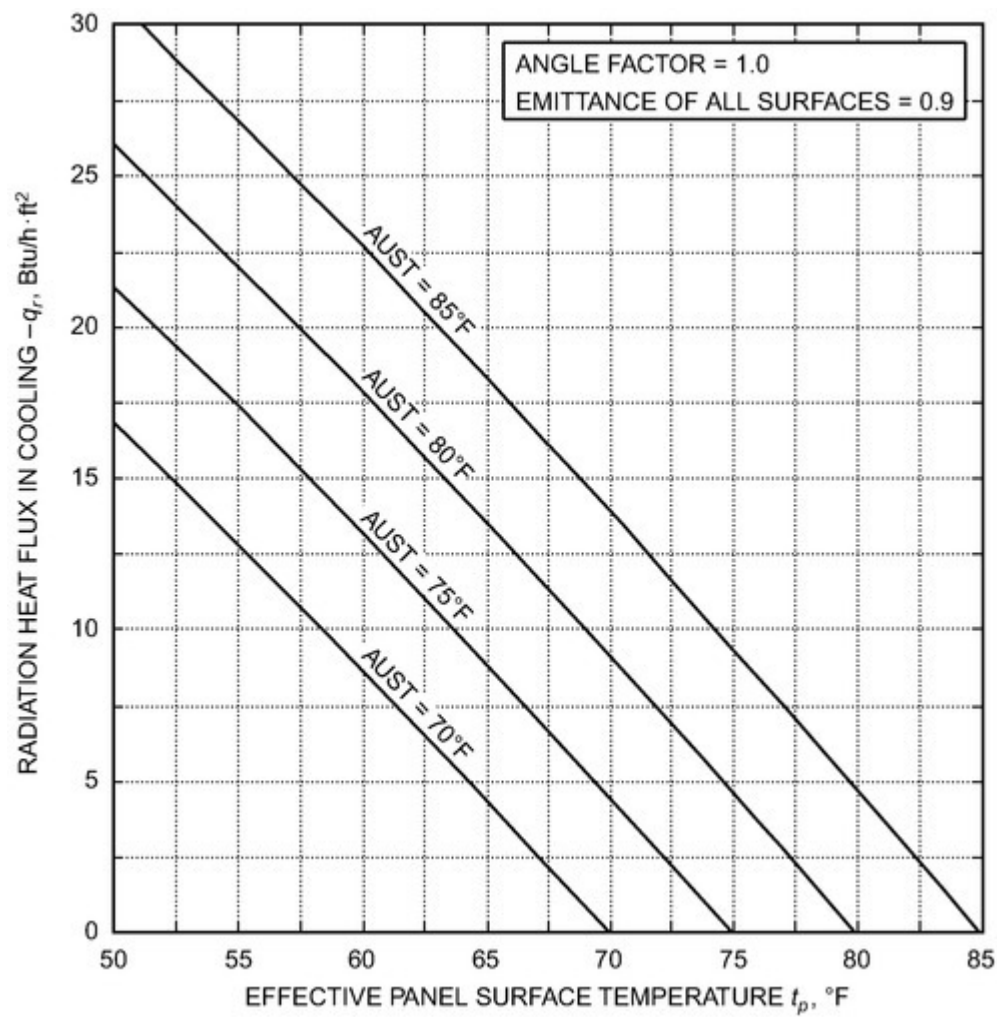


Figure 2. Heat Removed by Radiation at Cooled Ceiling or Wall Panel Surface

In many specific instances where normal multistory commercial construction and fluorescent lighting are used, the indoor air temperature at the 5 ft level closely approaches the AUST. In structures where the main heat gain is through the walls or where incandescent lighting is used, wall surface temperatures tend to rise considerably above the indoor air temperature. Note that [Equation \(5\)](#) is simplified such that it only characterizes longwave radiation heat transfer between the radiant surface and its enclosure surfaces. It does not include the effects of incident solar (shortwave and longwave) radiation and other internal heat gains on the active surfaces. For cases when solar radiation dominates, special considerations must be taken (Feng et al. 2013a; Liu et al. 2015; Simmonds 2006). Dynamic simulation software that implements the heat balance (HB) method (discussed in [Chapter 18 of the 2013 ASHRAE Handbook—Fundamentals](#)) is the most appropriate tool to characterize the performance of radiant surfaces in these cases.

Heat Transfer by Natural Convection

Heat flux from natural convection q_c occurs between the indoor air and the temperature-controlled panel surface. Thermal convection coefficients are not easily established. In natural convection, warming or cooling the boundary layer of air at the panel surface generates air motion. In practice, however, many factors, such as the indoor space configuration, interfere with or affect natural convection. Infiltration/exfiltration, occupants' movement, and mechanical ventilating systems can introduce some forced convection that disturbs the natural convection.

Parmelee and Huescher (1947) included the effect of forced convection on heat transfer occurring on heating or cooling panel surfaces as an increment to be added to the natural-convection coefficient. However, increased heat transfer from forced convection should not be factored into calculations because the increments are unpredictable in pattern and performance, and forced convection does not significantly increase the total heat flux on the active panel surface. Natural-convection heat flux in a panel system is a function of the effective panel surface temperature and the temperature of the air layer directly contacting the panel. The most consistent measurements are obtained when the dry-bulb air layer temperature is measured close to the region where the fully developed boundary layer begins, usually 2 to 2.5 in. from the panel surfaces.

Min et al. (1956) determined natural-convection coefficients 5 ft above the floor in the center of a 12 by 24.5 ft room ($D_h = 16.1$ ft). [Equations \(6\)](#) through [\(11\)](#), derived from this research, can be used to calculate heat flux from panels by natural convection.

Natural-convection heat flux between an **all-heated ceiling surface and indoor air**

$$q_c = 0.041 \frac{(t_p - t_a)^{1.25}}{D_e^{0.25}} \tag{6}$$

Natural-convection heat flux between a **heated floor or cooled ceiling surface and indoor air**

$$q_c = 0.39 \frac{|t_p - t_a|^{0.31} (t_p - t_a)}{D_e^{0.08}} \tag{7}$$

Natural-convection heat flux between a **heated or cooled wall panel surface and indoor air**

$$q_c = 0.29 \frac{|t_p - t_a|^{0.32} (t_p - t_a)}{H^{0.05}} \tag{8}$$

where

- q_c = heat flux from natural convection, Btu/h · ft²
- t_p = effective temperature of temperature-controlled panel surface, °F
- t_a = indoor space dry-bulb air temperature, °F
- D_e = equivalent diameter of panel (4 × area/perimeter), ft
- H = height of wall panel, ft

Schutrum and Humphreys (1954) measured panel performance in furnished test rooms that did not have uniform panel surface temperatures and found no variation in performance large enough to be significant in heating practice. Schutrum and Younis (1954) established that the effect of room size was usually insignificant except for very large spaces like hangars and warehouses, for which [Equations \(6\)](#) and [\(7\)](#) should be used. Otherwise, [Equations \(6\)](#), [\(7\)](#), and [\(8\)](#) can be simplified to the following by $D_h = 16.1$ ft and $H = 8.85$ ft:

Natural-convection heat flux between an **all-heated ceiling surface and indoor air**

$$q_c = 0.02(t_p - t_a)^{0.25}(t_p - t_a) \tag{9}$$

Natural-convection heat flux from a heated ceiling may be augmented by leaving cold strips (unheated ceiling sections), which help initiate natural convection. In this case, [Equation \(9\)](#) may be replaced by [Equation \(10\)](#) (Kollmar and Lise 1957):

$$q_c = 0.13(t_p - t_a)^{0.25}(t_p - t_a) \tag{10}$$

For large spaces such as aircraft hangars, if panels are adjoined, [Equation \(10\)](#) should be adjusted with the multiplier (16.1/ D_h)^{0.25}.

Natural-convection heat flux between a **heated floor or cooled ceiling surface and indoor air**

$$q_c = 0.31|t_p - t_a|^{0.31}(t_p - t_a) \tag{11}$$

Natural-convection heat flux between a **heated or cooled wall panel surface and indoor air**

$$q_c = 0.26|t_p - t_a|^{0.32}(t_p - t_a) \tag{12}$$

There are no confirmed data for floor cooling, but [Equation \(10\)](#) may be used for approximate calculations. Under normal conditions, conditions, t_a is the dry-bulb indoor air temperature. In floor-heated or ceiling-cooled spaces with large proportions of outdoor exposed fenestration, t_a may be taken to equal AUST.

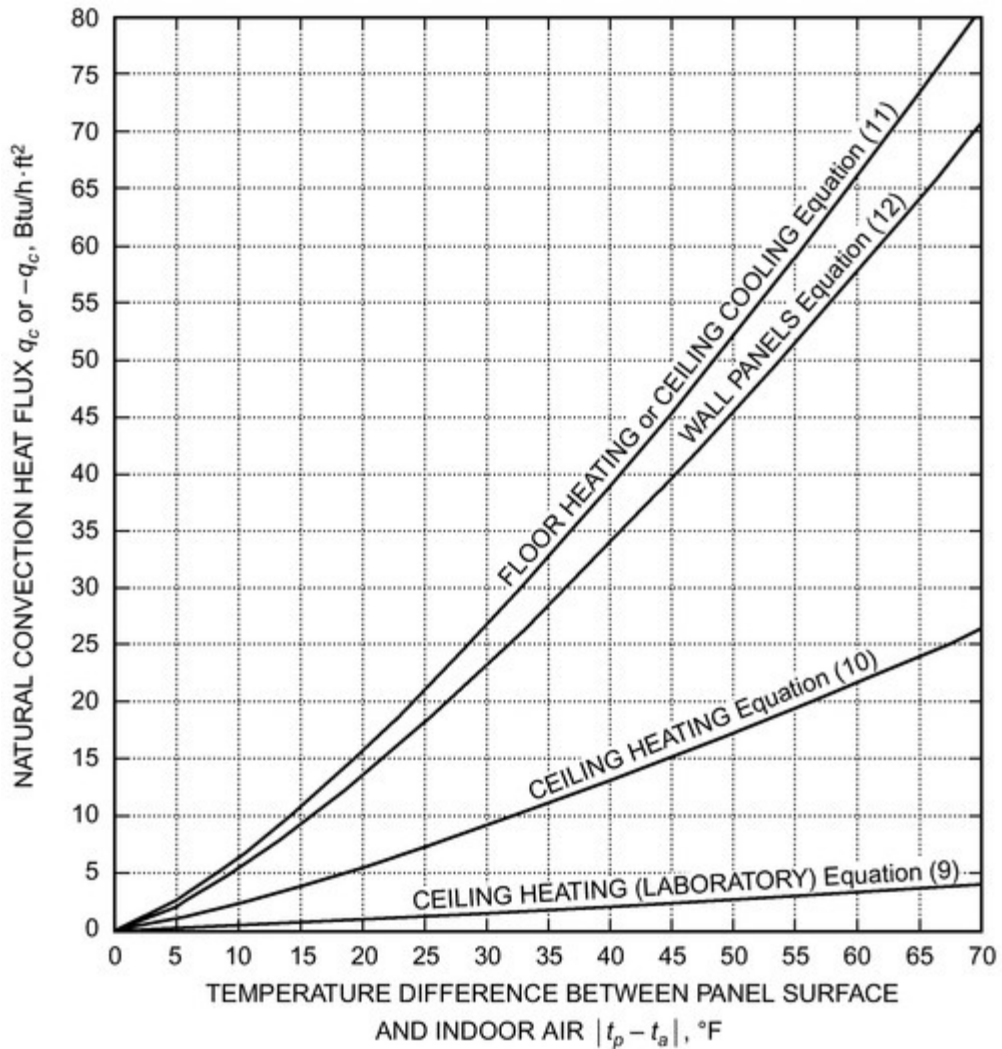


Figure 3. Natural-Convection Heat Transfer at Floor, Ceiling, and Wall Panel Surfaces

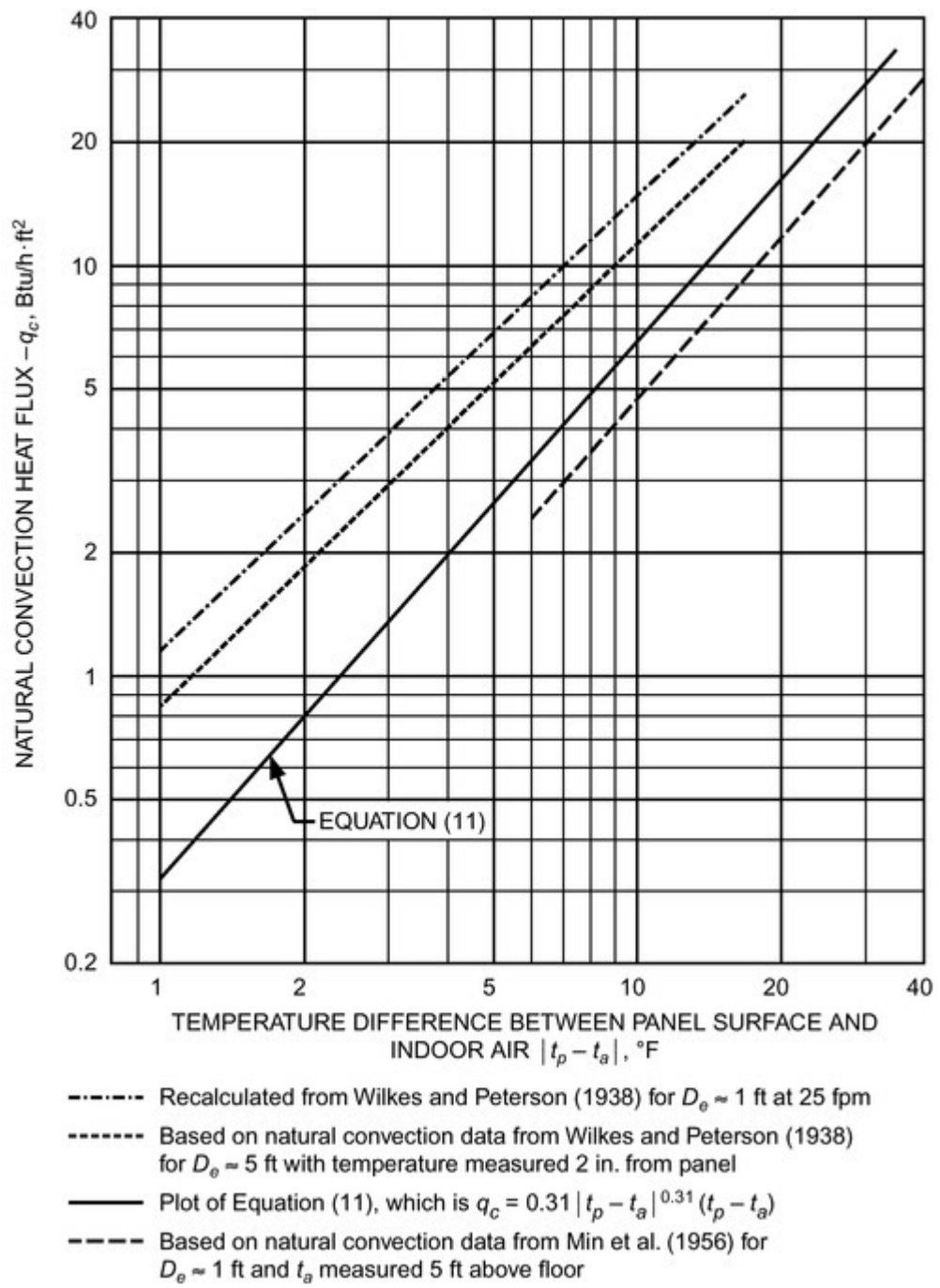


Figure 4. Empirical Data for Heat Removal by Ceiling Cooling Panels from Natural Convection

In cooling, t_p is less than t_a , so q_c is negative. [Figure 3](#) shows heat flux by natural convection at floor, wall, and ceiling heating panels as calculated from [Equations \(11\), \(12\), \(9\), and \(10\)](#), respectively. [Figure 4](#) compares heat removal by natural convection at cooled ceiling panel surfaces, as calculated by [Equation \(11\)](#), with data from Wilkes and Peterson (1938) for specific panel sizes. An additional curve illustrates the effect of forced convection on the latter data. Similar adjustment of the data from Min et al. (1956) is inappropriate, but the effects would be much the same.

Combined Heat Flux (Thermal Radiation and Natural Convection)

The combined heat flux on the active panel surface can be determined by adding the thermal-radiation heat flux q_r as calculated by [Equation \(5\)](#) (or from [Figures 1 and 2](#)) to the natural-convection heat flux q_c as calculated from [Equations \(9\), \(10\), \(11\), or \(12\)](#) or from [Figure 3](#) or [4](#), as appropriate. [Equation \(5\)](#) requires the AUST for the indoor space. In calculating the AUST, the surface temperature of interior walls may be assumed to be the same as the dry-bulb indoor air temperature. The inside surface temperature t_{si} of outdoor exposed walls and outdoor exposed floors or ceilings can be calculated from the following relationship:

$$h(t_{si} - t_{ai}) = U(t_{si} - t_o) \tag{13}$$

or

$$t_{si} = t_{ai} - \frac{U}{h}(t_{si} - t_o) \tag{14}$$

where
 h = natural-convection coefficient of the inside surface of an outdoor exposed wall or ceiling
 U = overall heat transfer coefficient of wall, ceiling, or floor, Btu/h · ft² · °F
 t_a = dry-bulb indoor space design air temperature, °F
 t_{si} = inside surface temperature of outdoor exposed wall, °F
 t_o = dry-bulb outdoor design air temperature, °F
From [Table 10 in Chapter 26 of the 2013 ASHRAE Handbook—Fundamentals](#),
 h = 1.63 Btu/h · ft² · °F for a horizontal surface with heat flow up
 h = 1.46 Btu/h · ft² · °F for a vertical surface (wall)
 h = 1.08 Btu/h · ft² · °F for a horizontal surface with heat flow down

[Figure 5](#) is a plot of [Equation \(13\)](#) for a vertical outdoor wall with 70°F dry-bulb indoor air temperature and $h = 1.46$ Btu/h · ft² · °F. For rooms with dry-bulb air temperatures above or below 70°F, the values in [Figure 5](#) can be corrected by the factors plotted in [Figure 6](#).

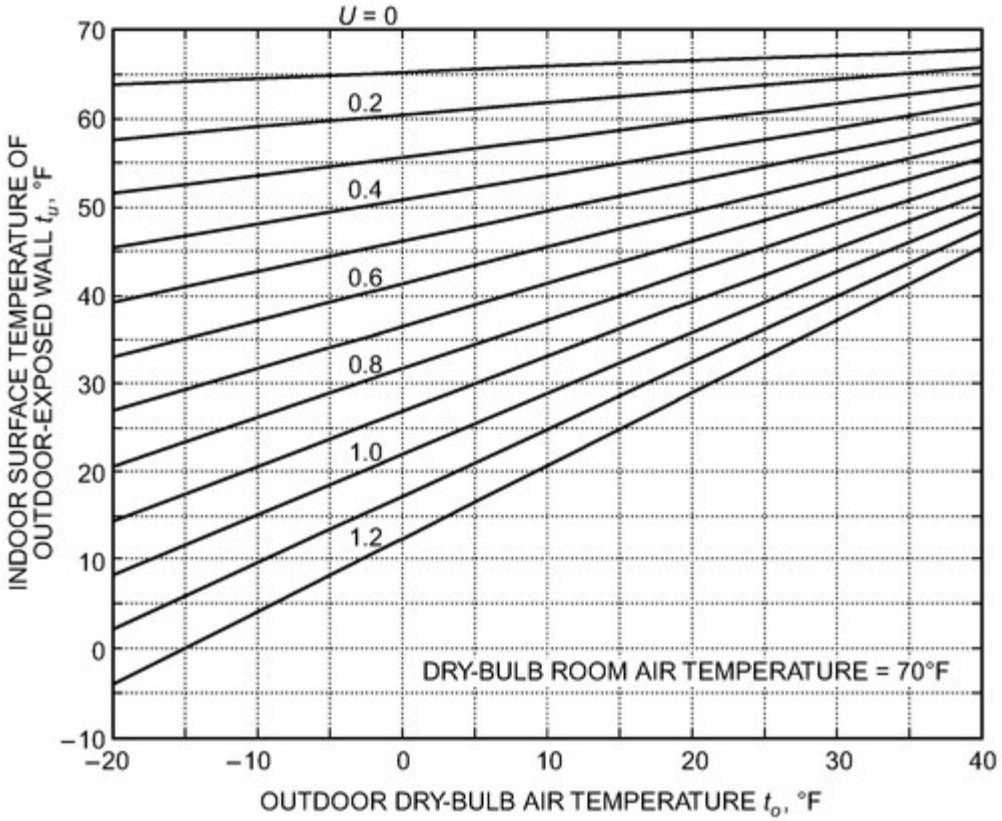


Figure 5. Relation of Inside Surface Temperature to Overall Heat Transfer Coefficient

Tests by Schutrum et al. (1953a, 1953b) and simulations by Kalisperis (1985) based on a program developed by Kalisperis and Summers (1985) show that the AUST and indoor air temperature are almost equal, if there is little or no outdoor exposure. Steinman et al. (1989) noted that this may not apply to enclosures with large fenestration or a high percentage of outdoor exposed wall and/or ceiling surface area. These surfaces may have a lower (in heating) or higher (in cooling) AUST, which increases the heat flux from thermal radiation.

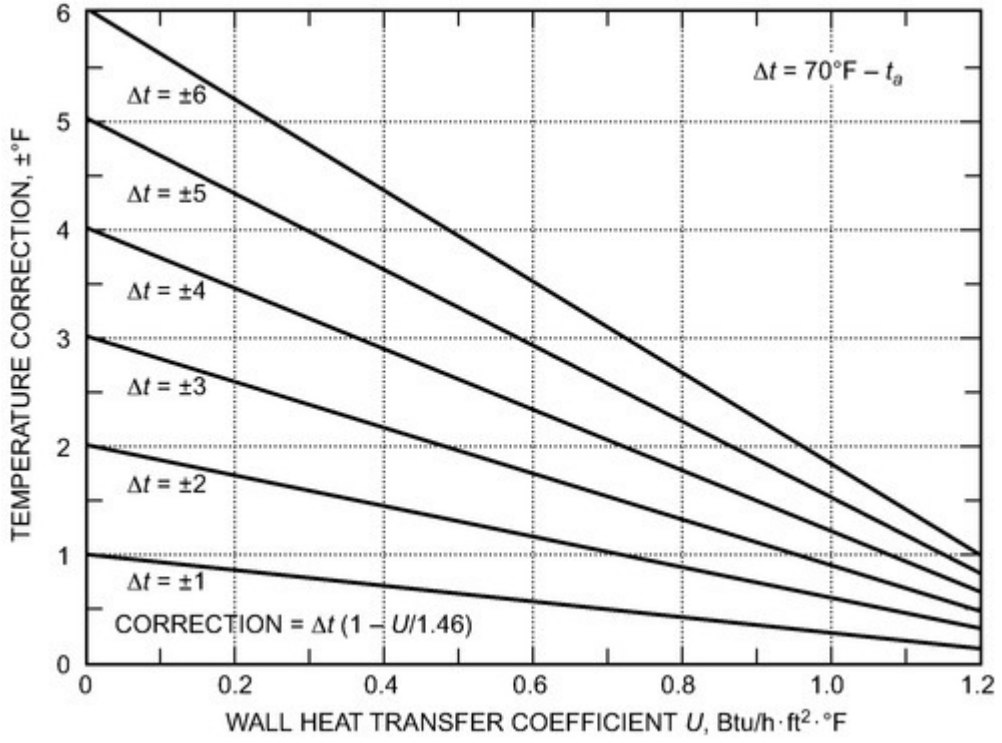


Figure 6. Inside Surface Temperature Correction for Exposed Wall at Dry-Bulb Air Temperatures Other Than 70°F

[Figure 7](#) shows the combined heat flux from thermal radiation and natural convection for cooling, as given in [Figures 2 and 4](#). The data in [Figure 7](#) do not include solar, lighting, occupant, or equipment heat gains.

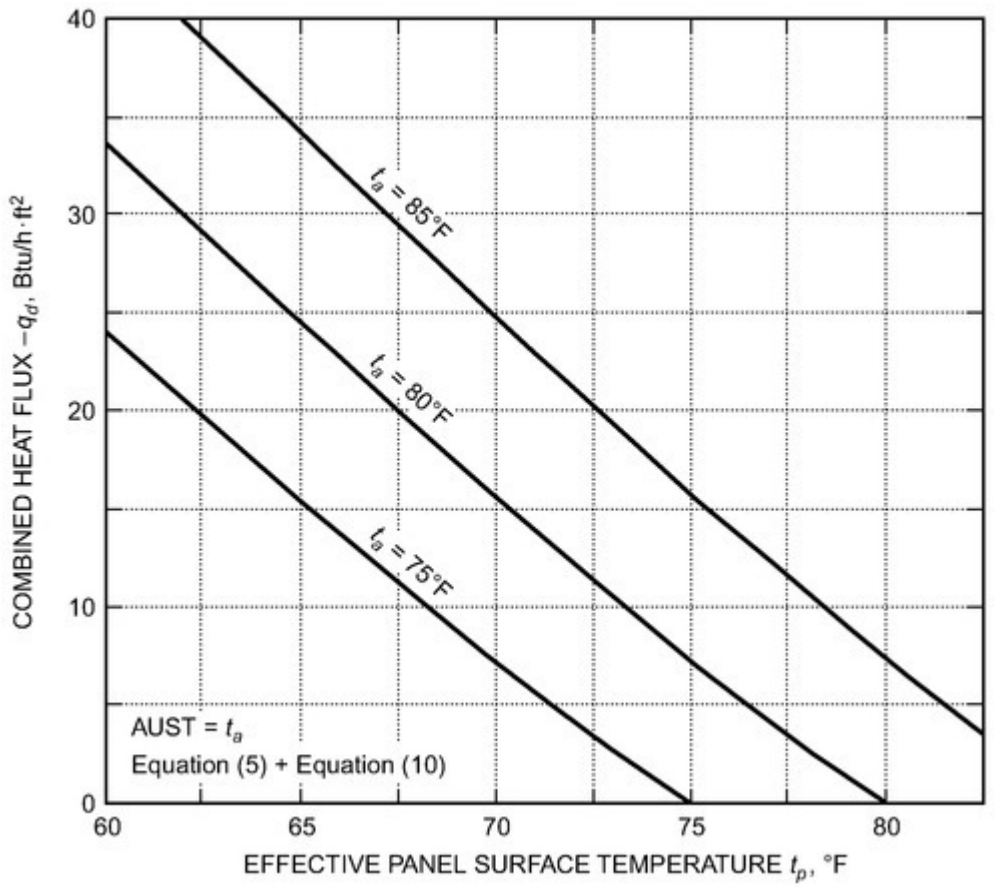


Figure 7. Cooled Ceiling Panel Performance in Uniform Environment with No Infiltration and No Internal Heat Sources

In suspended-ceiling panels, heat is transferred from the back of the ceiling panel to the floor slab above (in heating) or vice versa (in cooling). The ceiling panel surface temperature is affected because of heat transfer to or from the panel and the slab by thermal radiation and, to a much smaller extent, by natural convection. The thermal-radiation component of the combined heat flux can be approximated using [Equation \(5\)](#) or [Figure 1](#). The natural-convection component can be estimated from [Equation \(10\)](#) or [\(11\)](#) or from [Figure 3](#) or [4](#). In this case, the temperature difference is that between the top of the ceiling panel and the midspace of the ceiling. The temperature of the ceiling space should be determined by testing, because it varies with different panel systems. However, much of this heat transfer is nullified when insulation is placed over the ceiling panel, which, for perforated metal panels, also provides acoustical absorption. If artificial lighting fixtures are recessed into the suspended ceiling space, radiation from the top of the fixtures raises the overhead slab temperature and heat is transferred to the indoor air by natural convection. This heat is absorbed at the top of the cooled ceiling panels both by thermal radiation, in accordance with [Equation \(5\)](#) or [Figure 2](#), and by thermal convection, generally in accordance with [Equation \(10\)](#). The amount the top of the panel absorbs depends on the system. Similarly, panels installed under a roof absorb additional heat, depending on configuration and insulation.

1.2 FACTORS AFFECTING HEAT TRANSFER

Panel Thermal Resistance

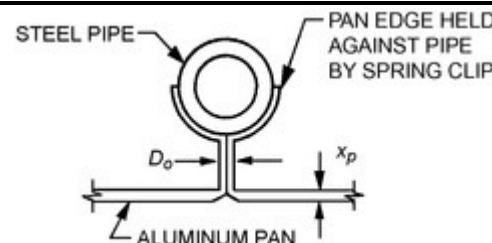
Any thermal resistance between the indoor space and the active panel surface, as well as between the active panel surface and the hydronic tubing or electric circuitry in the panel, reduces system performance. Thermal resistance to heat transfer may vary considerably among different panels, depending on the type of bond between the tubing (electric cabling) and the panel material. Factors such as corrosion or adhesion defects between lightly touching surfaces and the method of maintaining contact may change the bond with time. The actual thermal resistance of any proposed system should be verified by testing. Specific resistance and performance data, when available, should be obtained from the manufacturer based on performance tests such as those described in ANSI/ASHRAE *Standard* 138 or European *Standard* EN 14240 (CEN 2004). Panel thermal resistances include

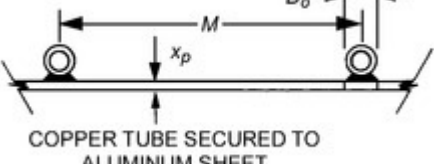
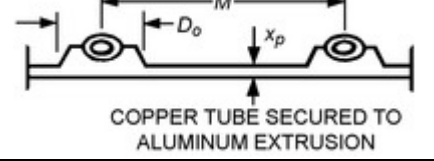
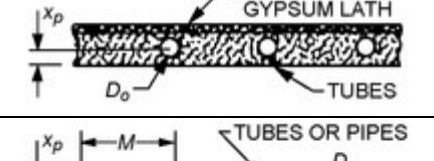
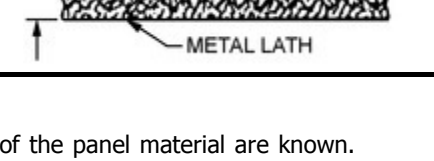
- r_t = thermal resistance of tube wall per unit tube spacing in a hydronic system, ft² · h · °F/Btu · ft
 - r_s = thermal resistance between tube (electric cable) and panel body per unit spacing between adjacent tubes (electric cables), ft² · h · °F/Btu · ft
 - r_p = thermal resistance of panel body, ft² · h · °F/Btu
 - r_c = thermal resistance of active panel surface covers, ft² · h · °F/Btu
 - r_{ra} = characteristic (combined) panel thermal resistance, ft² · h · °F/Btu
- For a given adjacent tube (electric cable) spacing M ,

$$r_{ti} = r_t M + r_s M + r_p + r_c \tag{15}$$

When the tubes (electric cables) are embedded in the slab, r_s may be neglected. However, if they are externally attached to the body of the panel, r_s may be significant, depending on the quality of bonding. [Table 1](#) gives typical r_s values for various ceiling panels.

Table 1. Thermal Resistance of Ceiling Panels

Type of Panel	Thermal Resistance	
	r_p , ft ² · h · °F/Btu	r_{ra} , ft ² · h · °F/Btu · ft
	$\frac{x_p}{k_p}$	0.55
		0.22

Type of Panel	Thermal Resistance	
	$r_p, \text{ft}^2 \cdot \text{h} \cdot ^\circ\text{F}/\text{Btu}$	$r_p, \text{ft}^2 \cdot \text{h} \cdot ^\circ\text{F}/\text{Btu} \cdot \text{ft}$
 COPPER TUBE SECURED TO ALUMINUM SHEET	$\frac{x_p}{k_p}$	
 COPPER TUBE SECURED TO ALUMINUM EXTRUSION	$\frac{x_p}{k_p}$	0.17
 METAL OR GYPSUM LATH TUBES	$\frac{x_p - D_o/2}{k_p}$	≈ 0
 TUBES OR PIPES METAL LATH	$\frac{x_p - D_o/2}{k_p}$	≤ 0.20

The value of r_p may be calculated if the characteristic panel thickness x_p and the thermal conductivity k_p of the panel material are known.
If the tubes (electric cables) are embedded in the panel,

$$r_p = \frac{x_p - D_o/2}{k_p} \tag{16}$$

where D_o = outside diameter of the tube (electric cable). Hydronic floor heating by a heated slab and gypsum-plaster ceiling heating are typical examples.
If the tubes (electric cable) are attached to the panel,

$$r_p = \frac{x_p}{k_p} \tag{17}$$

Metal ceiling panels (see [Table 1](#)) and tubes under subfloor (see [Figure 23](#)) are typical examples.
Thermal resistance per unit on-center spacing (M = unity) of circular tubes with inside diameter D_i and thermal conductivity k_i is

$$r_t = \frac{\ln(D_o/D_i)}{2\pi k_i} \tag{18}$$

For an elliptical tube with semimajor and semiminor axes of a and b , respectively, measured at both the outside and inside of the tube,

$$k_t = \ln \frac{(a_o + b_o)/(a_i + b_i)}{2\pi k_i} \tag{19}$$

In an electric cable, $r_t \approx 0$.

Table 2. Thermal Conductivity of Typical Tube Material

Material	Thermal Conductivity $k_t, \text{Btu}/\text{h} \cdot \text{ft} \cdot ^\circ\text{F}$
Carbon steel (AISI 1020)	30
Aluminum	137
Copper (drawn)	225
Red brass (85 Cu-15 Zn)	92
Stainless steel (AISI 202)	10
Low-density polyethylene (LDPE)	0.18
High-density polyethylene (HDPE)	0.24
Cross-linked polyethylene (VPE or PEX)	0.22
Textile-reinforced rubber hose (HTRH)	0.17
Polypropylene block copolymer (PP-C)	0.13
Polypropylene random copolymer (PP-RC)	0.14

In metal pipes, r_t is virtually the fluid-side thermal resistance:

$$r_t = \frac{1}{hD_i} \tag{20}$$

If the tube has multiple layers, [Equations \(18\)](#) or [\(19\)](#) should be applied to each individual layer and then summed to calculate the tube's total thermal resistance. Thermal resistance of capillary tube mats can also be calculated from either [Equation \(18\)](#) or [\(19\)](#). Typically, capillary tubes are circular, 1/12 in. in internal diameter, and 1/2 in. apart.
Capillary tube mats can be easily applied to existing ceilings in a sand plaster cover layer.
If the tube material is nonmetallic, oxygen ingress may be a problem, especially in panel heating. To avoid oxygen corrosion in the heating system, either (1) tubing with an oxygen barrier layer, (2) corrosion-inhibiting additives in the hydronic system, or (3) a heat exchanger separating the panel circuit from the rest of the system should be used.
[Table 8 in Chapter 4 of the 2013 ASHRAE Handbook—Fundamentals](#) may be used to calculate the forced-convection heat transfer coefficient h . [Table 2](#) gives values of k_t for different tube and pipe materials.

Effect of Floor Coverings

Active panel surface coverings like carpets and pads on the floor can have a pronounced effect on a panel system's performance. The added thermal resistance r_c reduces the panel surface heat flux. To reestablish the required performance, the water temperature must be increased (decreased in cooling). Thermal resistance of a panel covering is

$$r_c = \frac{x_c}{k_c} \tag{21}$$

where

r_c = thermal resistance of panel covering, $\text{ft}^2 \cdot \text{h} \cdot ^\circ\text{F}/\text{Btu}$

x_c = thickness of each panel covering, ft

k_c = thermal conductivity of each panel cover, $\text{Btu}/\text{h} \cdot \text{ft} \cdot ^\circ\text{F}$

If the active panel surface has more than one cover, individual r_c values should be added. [Table 3](#) gives typical r_c values for floor coverings.

If there are covered and bare floor panels in the same hydronic system, it may be possible to maintain a sufficiently high water temperature to satisfy the covered panels and balance the system by throttling the flow to the bare slabs. In some instances, however, the increased water temperature required when carpeting is applied over floor panels makes it impossible to balance floor panel systems in which only some rooms have carpeting unless the piping is arranged to permit zoning using more than one water temperature.

Panel Heat Losses or Gains

Heat transferred from the upper surface of ceiling panels, the back surface of wall panels, the underside of floor panels, or the exposed perimeter of any panel is considered a panel heat loss (gain in cooling). Panel heat losses (gains) are part of the building heat loss (gain) if the heat transfer is between the panel and the outside of the building. If heat transfer is between the panel and another conditioned space, the panel heat loss (gain) is a positive conditioning contribution for that space instead. In either case, the magnitude of panel loss (gain) should be calculated.

Panel heat loss (gain) to (from) space outside the conditioned space should be kept to a reasonable amount by insulation. For example, a floor panel may overheat the basement below, and a ceiling panel may cause the temperature of a floor surface above it to be too high for comfort unless it is properly insulated.

The heat loss from most panels can be calculated by using the coefficients given in [Table 1 in Chapter 26 of the 2013 ASHRAE Handbook—Fundamentals](#). These coefficients should not be used to determine downward heat loss from panels built on grade because heat flow from them is not uniform (ASHAE 1956, 1957; Sartin and Harris 1956). The heat loss from panels built on grade can be estimated from [Figure 8](#) or from [Equations \(41\)](#) and [\(42\) in Chapter 18 of the 2013 ASHRAE Handbook—Fundamentals](#).

Table 3. Thermal Resistance of Floor Coverings

Description	Thermal Resistance $r_c, \text{ft}^2 \cdot \text{h} \cdot ^\circ\text{F}/\text{Btu}$
Bare concrete, no covering	0
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.60
Light carpet with rubber pad	1.00
Light carpet with light pad	1.40
Light carpet with heavy pad	1.70
Heavy carpet	0.80
Heavy carpet with rubber pad	1.20
Heavy carpet with light pad	1.60
Heavy carpet with heavy pad	1.90
3/8 in. hardwood	0.54
5/8 in. wood floor (oak)	0.57
1/2 in. oak parquet and pad	0.68
Linoleum	0.12
Marble floor and mudset	0.18
Rubber pad	0.62
Prime urethane underlayment, 3/8 in.	1.61
48 oz. waffled sponge rubber	0.78
Bonded urethane, 1/2 in.	2.09

Notes:

1. Carpet pad thickness should not be more than 1/4 in.
2. Total thermal resistance of carpet is more a function of thickness than of fiber type.
3. Generally, thermal resistance (R-value) is approximately 2.6 times the total carpet thickness in inches.
4. Before carpet is installed, verify that the backing is resistant to long periods of continuous heat up to 120°F.

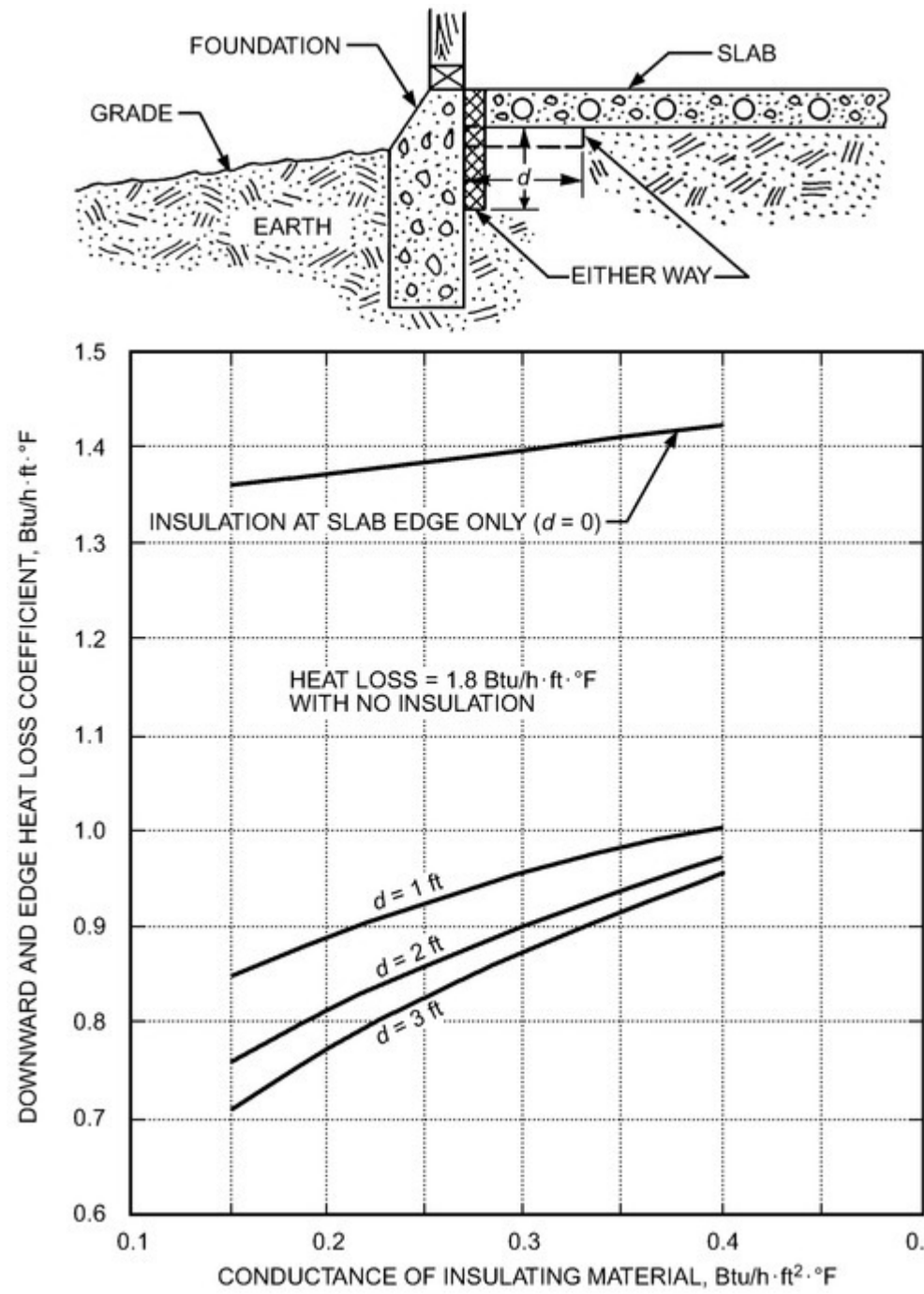


Figure 8. Downward and Edgewise Heat Loss Coefficient for Concrete Floor Slabs on Grade

Panel Performance

As with other electric or hydronic terminal units, panel performance can be described by the following equation:

$$q = C\Delta T|\Delta T|^{m-1} \tag{22}$$

where

q = combined heat flux on panel surface, $\text{Btu}/\text{h} \cdot \text{ft}^2$

C = characteristic performance coefficient, $\text{Btu}/\text{h} \cdot \text{ft}^2 \cdot ^\circ\text{F}^m$

ΔT = temperature difference, either $t_p - t_a$ in electric heating or $t_w - t_a$ in hydronic heating, °F

t_a = operative temperature, °F

$m_b = 2 + r_c/2r_p$

C and m for a particular panel may be either experimentally determined or calculated from the design material given in this chapter. In either case, sufficient data or calculation points must be gathered to cover the entire operational design range (ASHRAE Standard 138).

Radiant panel product performance data can be obtained from the manufacturers. ASHRAE Standard 138 and European Standards EN 14240 and EN 14037 regulate the testing and rating method for radiant panel cooling and heating performance. European Standards EN 1264 and EN 15377 regulate the testing methods for the embedded radiant heating and cooling systems.

1.3 PANEL DESIGN

Either hydronic or electric circuits control the active panel surface temperature. The required effective surface temperature t_p necessary to maintain a combined heat flux q (where $q = q_c + q_a$) at steady-state design conditions can be calculated by using applicable heat flux equations for q_c and q_a , depending on the position of the panel. At a given t_a , AUST must be predicted first. [Figures 9](#) and [10](#) can also be used to find t_p when q and AUST are known. The next step is to determine the required average water (brine) temperature t_w in a hydronic system. It depends primarily on t_p , tube spacing M , and the characteristic panel thermal resistance r_p . [Figure 9](#) provides design information for heating and cooling panels, positioned either at the ceiling or on the floor.

The combined heat flux for ceiling and floor panels can be read directly from [Figure 9](#). Here q_a is the combined heat flux on the floor panel and q_c is the combined heat flux on the ceiling panel. For an electric heating system, t_w scales correspond to the skin temperature of the cable. The following algorithm (TSI 1994) may also be used to design and analyze panels under steady-state conditions:

$$t_d \approx t_a + \frac{(t_p - t_a)M}{2W\eta + D_o} + q(r_p + r_c + r_sM) \tag{23}$$

where

t_d = average skin temperature of tubing (electric cable), °F

q = combined heat flux on panel surface, $\text{Btu}/\text{h} \cdot \text{ft}^2$

t_a = dry-bulb indoor air design temperature, °F. In floor-heated or ceiling-cooled indoor spaces that have large fenestration, t_a may be replaced with AUST.

D_o = outside diameter of embedded tube or characteristic contact width of attached heating or cooling element with panel (see [Table 1](#)), ft

M = on-center spacing of adjacent tubes (electric cables), ft

$2W$ = net spacing between tubing (electric cables), $M - D_o$, ft

η = fin efficiency, dimensionless

The first two terms in [Equation \(23\)](#) give the maximum (minimum in cooling) value of the panel surface temperature profile; consequently, if tube spacing M is too large, hot strips along the panel surface may occur in heating, or local condensation occur on these cold strips in sensible cooling mode.

$$\eta = \frac{\tanh(fW)}{fW} \tag{24}$$

$$\eta \approx 1/fW \quad \text{for } fW > 2 \tag{25}$$

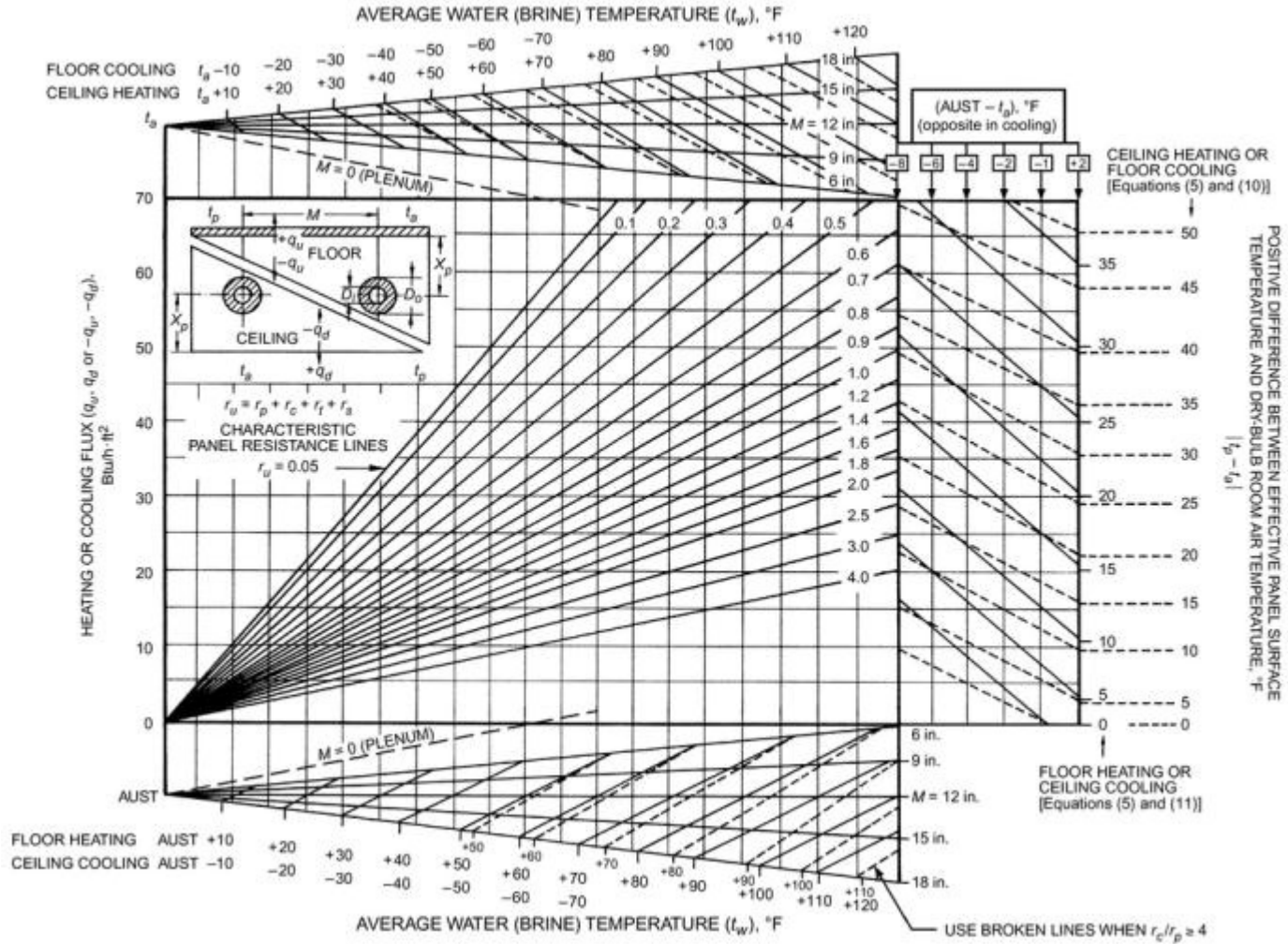


Figure 9. Design Graph for Sensible Heating and Cooling with Floor and Ceiling Panels

The following equation, which includes transverse heat diffusion in the panel and surface covers, may be used to calculate the fin coefficient f :

$$f \approx \left[\frac{q}{m(t_p - t_d) \sum_{i=1}^n k_i x_i} \right]^{1/2} \quad \text{for } t_p \neq t_d \tag{26}$$

where

- f = fin coefficient
- $m = 2 + r_i/2r_p$
- n = total number of different material layers, including panel and surface covers
- x_i = characteristic thickness of each material layer i , ft
- k_i = thermal conductivity of each layer i , Btu/h · ft · °F

For a hydronic system, the required average water (brine) temperature is

$$t_w = (q + q_p)Mr_t + t_d \tag{27}$$

where q_p is the flux of back and perimeter heat losses (positive) in a heated panel or gains (negative) in a cooling panel. This algorithm may be applied to outdoor slab heating systems, provided that the combined heat flux q is calculated according to [Chapter 51 in the 2015 ASHRAE Handbook—HVAC Applications](#), with the following conditions: no snow, no evaporation, q (in panel heating) = q_r (radiation and convection heat flux in snow-melting calculations). With a careful approach, outdoor slab cooling systems may be analyzed by incorporating the solar radiation gain, thermal radiation, and forced convection from the sky and ambient air.

ISO *Standard* 18556-2014 provides an in-depth description of the design, installation, and control of radiant panels.

Special Cases

[Figure 9](#) may also be used for panels with tubing not embedded in the panel:

- $x_p = 0$ if tubes are externally attached.
- In spring-clipped external tubing, $D_i = 0$ and D_o is the clip thickness.

Warm air and electric heating elements are two design concepts influenced by local factors. The warm-air system has a special cavity construction where air is supplied to a cavity behind or under the panel surface. The air leaves the cavity through a normal diffuser arrangement and is supplied to the indoor space. Generally, these systems are used as floor panels in schools and in floors subject to extreme cold, such as in an overhang. Cold outdoor temperatures and heating medium temperatures must be analyzed with regard to potential damage to the building construction. Electric heating elements embedded in the floor or ceiling construction and unitized electric ceiling panels are used in various applications to provide both full heating and spot heating of the space.

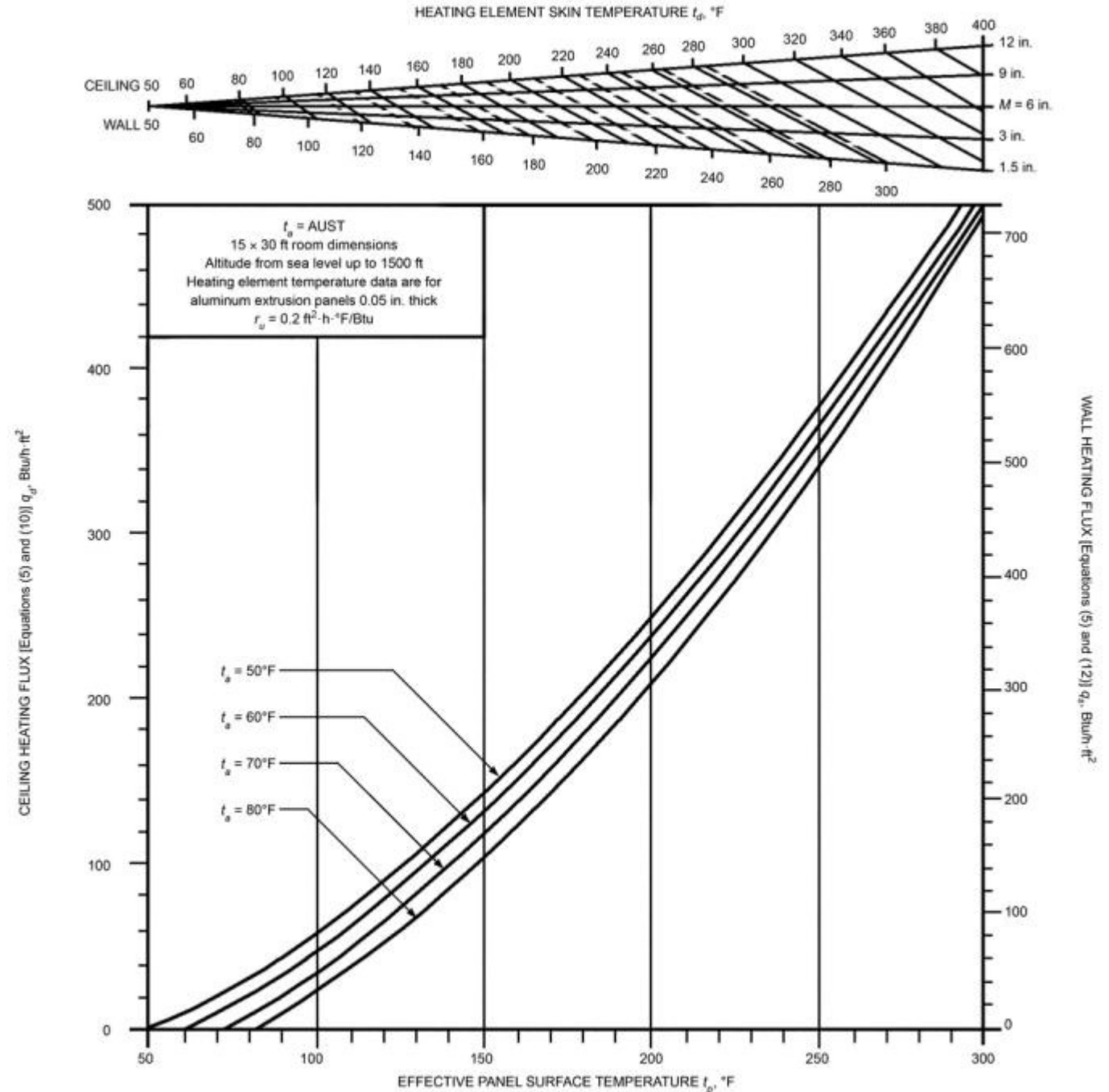


Figure 10. Design Graph for Heating with Aluminum Ceiling and Wall Panels

Examples

Residential heating applications usually consist of tubes or electric elements embedded in masonry floors or plaster ceilings. This construction is suitable where loads are stable and building design minimizes solar effects. However, in buildings where fenestration is large and thermal loads change abruptly, the slow response, thermal lag, and override effect of masonry panels are unsatisfactory. Metal ceiling panels with low thermal mass respond quickly to load changes (Berglund et al. 1982). Panels are preferably located in the ceiling because it is exposed to all other indoor surfaces, occupants, and objects in the conditioned indoor space. It is not likely to be covered, as floors are, and higher surface temperatures can be maintained. [Figure 10](#) gives design data for ceiling and wall panels with effective surface temperatures up to 300°F.

Example 1. An in-slab, on-grade panel (see [Figure 20](#)) will be used for both heating and cooling. $M = 1$ ft (12 in.), $r_o = 0.5$ ft² · h · °F/Btu, and r_i/r_o is less than 4. t_d is 68°F in winter and 76°F in summer. AUST is expected to be 2°F less than t_d in winter heating and 1°F higher than t_d in summer cooling.

What is the average water temperature and effective floor temperature (1) for winter heating when $q_w = 40$ Btu/h · ft², and (2) for summer cooling when $-q_c = 15$ Btu/h · ft²?

Solution:

Winter heating

To obtain the average water temperature using [Figure 9](#), start on the left axis where $q_w = 40$ Btu/h · ft². Proceed right to the intersect $r_o = 0.5$ and then down to the $M = 12$ in. line. The reading is AUST + 56, which is the solid line value because $r_i/r_o < 4$. As stated in the initial problem, AUST = $t_d - 2$ or AUST = 68 – 2 = 66°F. Therefore, the average water temperature would be $t_w = 66 + 56 = 122$ °F.

To find the effective floor temperature, start at $q_w = 40$ Btu/h · ft² in [Figure 9](#) and proceed right to AUST = $t_d - 2$ °F. The solid line establishes 21°F as the temperature difference between the panel and the indoor air. Therefore, the effective floor temperature $t_p = t_d + 2$ or $t_p = 68 + 2 = 70$ °F.

Summer cooling

Using [Figure 9](#), start at the left axis at $-q_c = 15$ Btu/h · ft². Proceed to $r_o = 0.5$, and then up (for cooling) to $M = 12$ in., which reads $t_d - 23$ or 76 – 23 = 53°F average water temperature for cooling.

To obtain the effective floor temperature at $-q_c = 15$ Btu/h · ft², proceed to AUST = $t_d + 1$ °F, which reads –11°F. Therefore, the effective floor temperature is 76 – 11 = 65°F.

Example 2. An aluminum extrusion panel, which is 0.05 in. thick with heat element spacing of $M = 0.5$ ft (6 in.), is used in the ceiling for heating. If a ceiling heat flux q_c of 400 Btu/h · ft² is required to maintain room temperature t_d at 70°F, what is the required heating element skin temperature t_s and effective panel surface temperature t_p ?

Solution:

Using [Figure 10](#) enter the left axis heat flux q_c at 400 Btu/h · ft². Proceed to the line corresponding to $t_d = 70$ °F and then move up to the $M = 6$ in. line. The ceiling heating element temperature t_s at the intersection point is 320°F. From the bottom axis of [Figure 10](#), the effective panel surface temperature t_p is 265°F.

2. GENERAL DESIGN CONSIDERATIONS

Principal **advantages** of panel systems are the following:

- Because not only indoor air temperature but also mean radiant temperature can be controlled, total human thermal comfort may be better satisfied.
- Because the operative temperature for required human thermal comfort may be maintained by primarily controlling the mean radiant temperature of the conditioned indoor space, dry-bulb air temperature may be lower (in heating) or higher (in cooling), which reduces sensible heating or cooling loads (see [Chapter 16](#) for the definition and calculation of operative and mean radiant temperatures).
- Hydronic panel systems may be connected in series, following other hydronic heating or cooling systems (i.e., their return water may be used), increasing exergetic efficiency.
- Comfort levels can be better than those of other space-conditioning systems because thermal loads are satisfied directly and air motion in the space corresponds to required ventilation only.
- Waste and low-enthalpy energy sources and heat pumps may be directly coupled to panel systems without penalty on equipment sizing and operation. Being able to select from a wide range of moderate operation temperatures ensures optimum design for minimum cost and maximum thermal and exergetic efficiency.
- Seasonal thermal distribution efficiency in buildings may be higher than in other hydronic systems.
- In terms of simple payback period, ceiling cooling panels and chilled beams have the highest technical energy savings potential (DOE 2002).
- Part or all of the building structure may be thermally activated (Meierhans and Olesen 2002).
- Space-conditioning equipment is not required at outdoor exposed walls, simplifying wall, floor, and structural systems.
- Almost all mechanical equipment may be centrally located, simplifying maintenance and operation.
- No space within the conditioned space is required for mechanical equipment. This feature is especially valuable in hospital patient rooms, offices, and other applications where space is at a premium, if maximum cleanliness is essential or legally required.
- Significantly reduced need for ductwork allows either lower floor-to-floor heights (reducing façade costs and overall building height) or higher ceilings (an architecturally desirable feature).
- Draperies and curtains can be installed at outdoor exposed walls without interfering with the space-conditioning system.
- When four-pipe systems are used, cooling and heating can be simultaneous, without central zoning or seasonal changeover.
- Supply air requirements usually do not exceed those required for ventilation and humidity control.
- Modular panels provide flexibility to meet changes in partitioning.
- A 100% outdoor air system may be installed with smaller penalties in refrigeration load because of reduced air quantities.
- A common central air system can serve both the interior and perimeter zones.
- Wet-surface cooling coils are eliminated from the occupied space, reducing the potential for septic contamination.
- Modular panel systems can use the automatic sprinkler system piping (see NFPA *Standard* 13, Section 3.6). The maximum water temperature must not fuse the heads.
- Panel heating and cooling with minimum supply air quantities provide a draft-free environment.
- Noise associated with fan-coil or induction units is eliminated.
- Peak loads are reduced as a result of thermal energy storage in the panel structure, as well as walls and partitions directly exposed to panels.
- Panels can be combined with other space-conditioning systems to decouple several indoor requirements (e.g., humidity control, indoor air quality, air velocity) and optimally satisfy them without compromises.
- In-floor heating creates inhospitable living conditions for house dust mites compared to other heating systems (Sugawara et al. 1996).

Disadvantages include the following:

- Response time can be slow if controls and/or heating elements are not selected or installed correctly.
- Improper selection of panel heating or cooling tube or electrical heating element spacing and/or incorrect sizing of heating/cooling source can cause nonuniform surface temperatures or insufficient sensible heating or cooling capacity.
- Radiant systems can satisfy only *sensible* heating and cooling loads. In a stand-alone radiant cooling system, dehumidification and surface condensation may be of prime concern and a secondary air-handling system must provide ventilation air and handle latent loads.
- Radiant systems in which the tubing is embedded in a building surface or in the structural slab typically have a large exposed concrete surface area. Additional acoustical treatments are often required to reduce room reverberation time to a more desirable level.

2.1 HYBRID SYSTEMS

In general, any HVAC system relies on a single heat transfer mode as its major heat transfer mechanism. For example, central air conditioning is a forced-convection system, and a high-intensity radiant system operates almost purely with thermal radiation. Additionally, every system has a typical range for satisfactory and economical operation, with its own advantages, limitations, and disadvantages. A single system may not be sufficient to encompass all requirements of a given building in the most efficient and economical way. Under these circumstances, it may be more desirable to decouple several components of indoor space conditioning, and satisfy them by several dedicated systems. A hybrid heating and cooling system is an optimum partnership of multiple, collocated, simultaneously operating heating and cooling systems, each of which is based on one of the primary heat transfer modes (i.e., radiation and convection). In its most practical form, hybrid heating and cooling may consist of a radiant system and a forced-air system. The forced-air component may be a central HVAC system or a hydronic system such as fan-coils, as shown in [Figure 11](#). Here, a panel system is added downstream of the condensing fan-coils, and ventilation is provided by a separate system with substantially reduced duct size. Dehumidification and ventilation problems that may be associated with stand-alone panel cooling systems may be eliminated by hybrid HVAC systems.

ASHRAE research project RP-1140 (Scheatle 2003) successfully demonstrated the use of panel systems for both heating and cooling, in conjunction with a forced-convection system, to economically achieve year-round thermal comfort in a residence using both active and passive performance of the building and a ground-source heat pump. Skylighting, an energy recovery ventilator, and packaged dehumidifiers were also used. Twenty-four-month-long tests indicated that a radiant/convective system can offer substantial cost savings, given proper design and control.

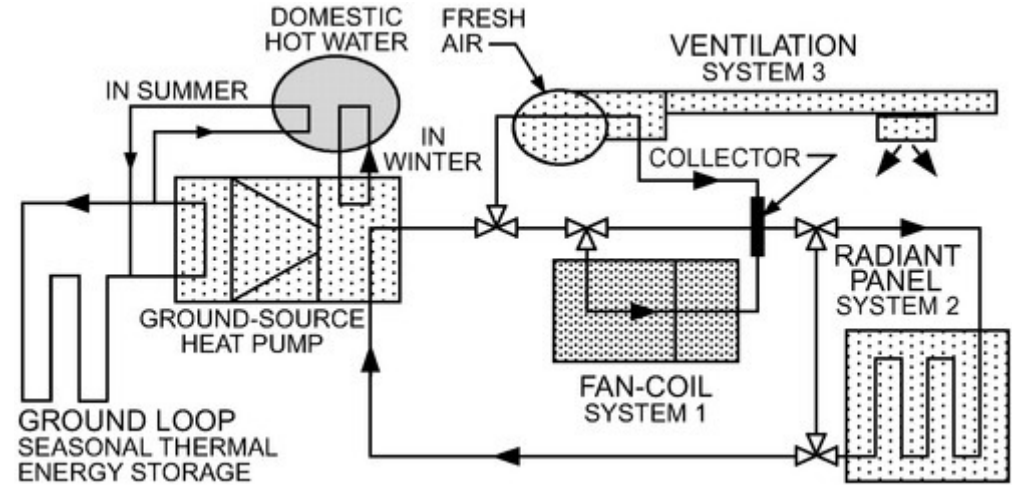


Figure 11. Typical Residential Hybrid HVAC System

3. RADIANT HEATING AND COOLING SYSTEMS

The following are the most common forms of panels applied in radiant heating and cooling systems:

- Hydronic ceiling panels
- Embedded tubing in ceilings, walls, or floors
- Electric ceiling panels for heating only
- Electrically heated ceilings or floors

3.1 HYDRONIC CEILING PANELS

Metal ceiling panels can be integrated into a system that heats and cools. In such a system, a source of dehumidified ventilation air is required in summer, so the system is classed as an air-and-water system. Also, various amounts of forced air are supplied year-round. When metal panels are applied for heating only, a ventilation system may be required, depending on local codes. Ceiling panel systems are an outgrowth of perforated metal suspended acoustical ceilings. These ceiling panel systems are usually designed into buildings where the suspended acoustical ceiling can be combined with panel heating and cooling. The panels can be designed as small units to fit the building module, which provides extensive flexibility for zoning and control, or the panels can be arranged as large continuous areas for maximum economy. Some ceiling installations require active panels to cover only part of the indoor space and compatible matching acoustical panels for the remaining ceiling area.

Three types of metal ceiling systems are available. The first consists of light aluminum panels, usually 12 by 24 in., attached in the field to 0.5 in. galvanized pipe coils. [Figure 12](#) shows a metal ceiling panel system that uses 0.5 in. pipe laterals on 6, 12, or 24 in. centers, hydraulically connected in a sinuous or parallel-flow welded system. Aluminum ceiling panels are clipped to these pipe laterals and act as a heating panel when warm water is flowing or as a cooling panel when chilled water is flowing.

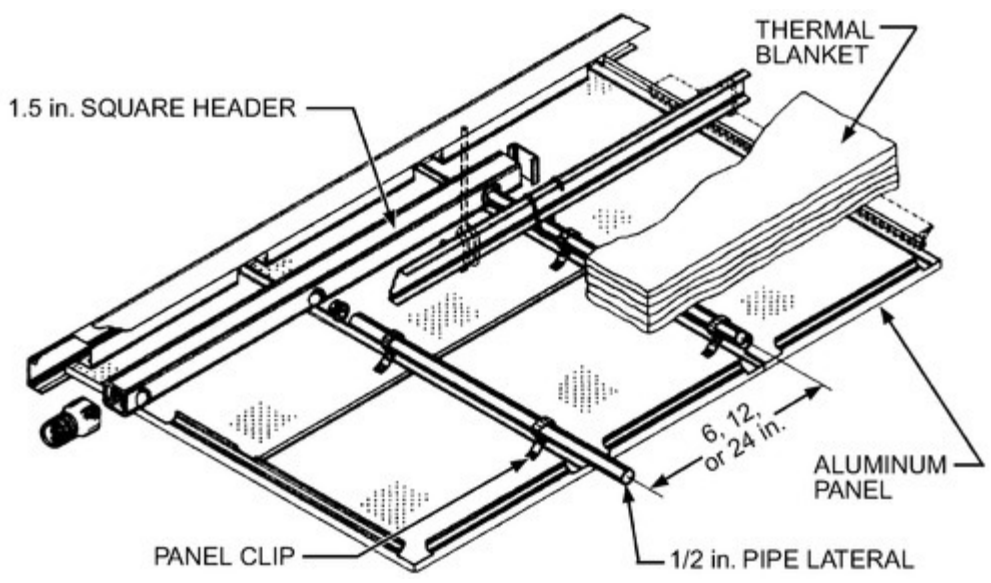


Figure 12. Metal Ceiling Panels Attached to Pipe Laterals

The second type of panel consists of a copper coil secured to the aluminum face sheet to form a modular panel. Modular panels are available in sizes up to about 36 by 60 in. and are held in position by various types of ceiling suspension systems, most typically a standard suspended T-bar 24 by 48 in. exposed grid system. [Figure 13](#) shows metal panels using a copper tube pressed into an aluminum extrusion, although other methods of securing the copper tube have proven equally effective.

Metal ceiling panels can be perforated so that the ceiling becomes sound absorbent when acoustical material is installed on the back of the panels. The acoustical blanket is also required for thermal reasons, so that reverse loss or upward flow of heat from the metal ceiling panels is minimized.

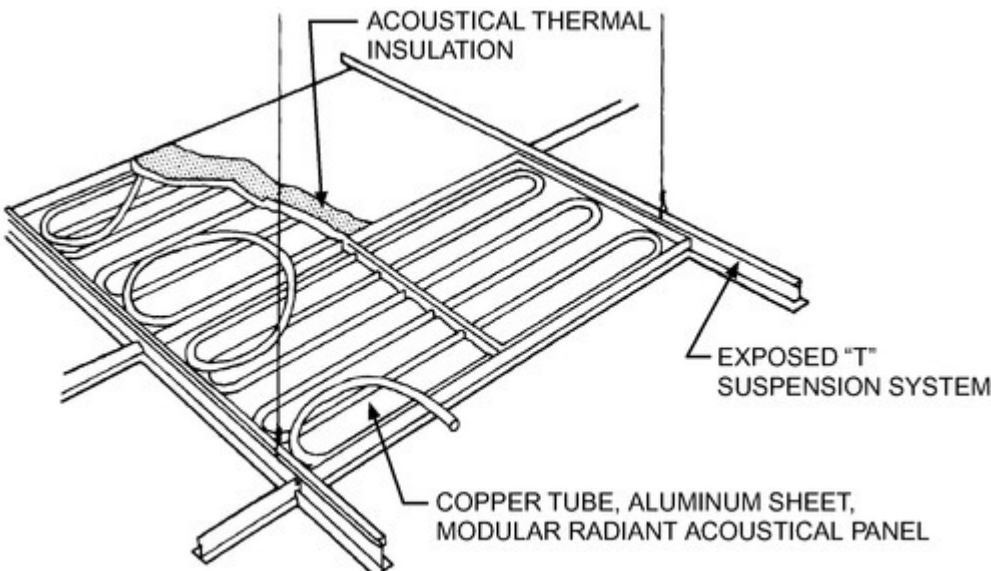


Figure 13. Metal Ceiling Panels Bonded to Copper Tubing

The third type of panel is an aluminum extrusion face sheet with a copper tube mechanically fastened into a channel housing on the back. Extruded panels can be manufactured in almost any shape and size. Extruded aluminum panels are often used as long, narrow panels at the outside wall and are independent of the ceiling system. Panels 15 or 20 in. wide usually satisfy a typical office building's heating requirements. Lengths up to 20 ft are available. [Figure 14](#) shows metal panels using a copper tube pressed into an aluminum extrusion. Some manufacturers also use graphite as a panel material.

Performance data for extruded aluminum panels vary with the copper tube/aluminum contact and test procedures used. Hydronic ceiling panels have a low thermal resistance and respond quickly to changes in space conditions. [Table 1](#) gives thermal resistance values for various ceiling constructions.

Metal ceiling panels can be used with any of the air-air cooling systems described in [Chapter 7, Chapters 12 and 18 of the 2013 ASHRAE Handbook—Fundamentals](#) describe how to calculate heating loads. Double-glazing and heavy insulation in outside walls reduce transmission heat losses. As a result, infiltration and reheating have become of greater concern. Additional design considerations include the following:

- Perimeter radiant heating panels extending not more than 3 ft into the indoor space may operate at higher temperatures, as described in the section on Hydronic Panel Systems.
- Hydronic panels operate efficiently at low temperature and are suitable for condenser water heat reclaim systems.
- Locate ceiling panels adjacent to the outside wall and as close as possible to the areas of maximum load. The panel area within 3 ft of the outside wall should have a heating capacity equal to or greater than 50% of the wall transmission load.
- Ceiling system designs based on passing return air through perforated modular panels into the plenum space above the ceiling are not recommended because much of this heat is lost to the return air system in heating mode.
- When selecting heating design temperatures for a ceiling panel surface or average water temperature, the design parameters are as follows:
 - Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect."
 - Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
 - Give ceiling panel location priority.
 - With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235°F surface temperature. If panels extend beyond 3 ft into the indoor space, the panel surface temperature should be limited to the values as given in [Figure 15](#).
- Allow sufficient space above the ceiling for installation and connection of the piping that forms the panel ceiling.
- Hydronic ceiling panels provide a fast-response system (Watson et al. 1998).

Metal acoustic panels provide heating, cooling, sound absorption, insulation, and unrestricted access to the plenum space. They are easily maintained, can be repainted to look new, and have a life expectancy in excess of 30 years. The system is quiet, comfortable, draft-free, easy to control, and responsive. The system is a basic air-and-water system.

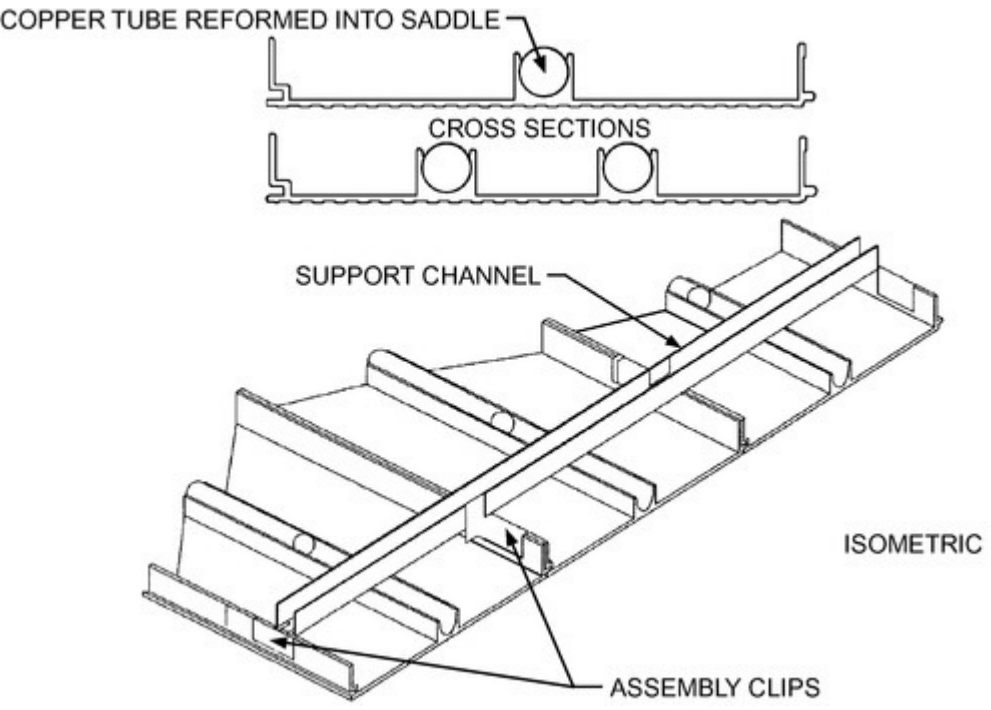


Figure 14. Extruded Aluminum Panels with Integral Copper Tube

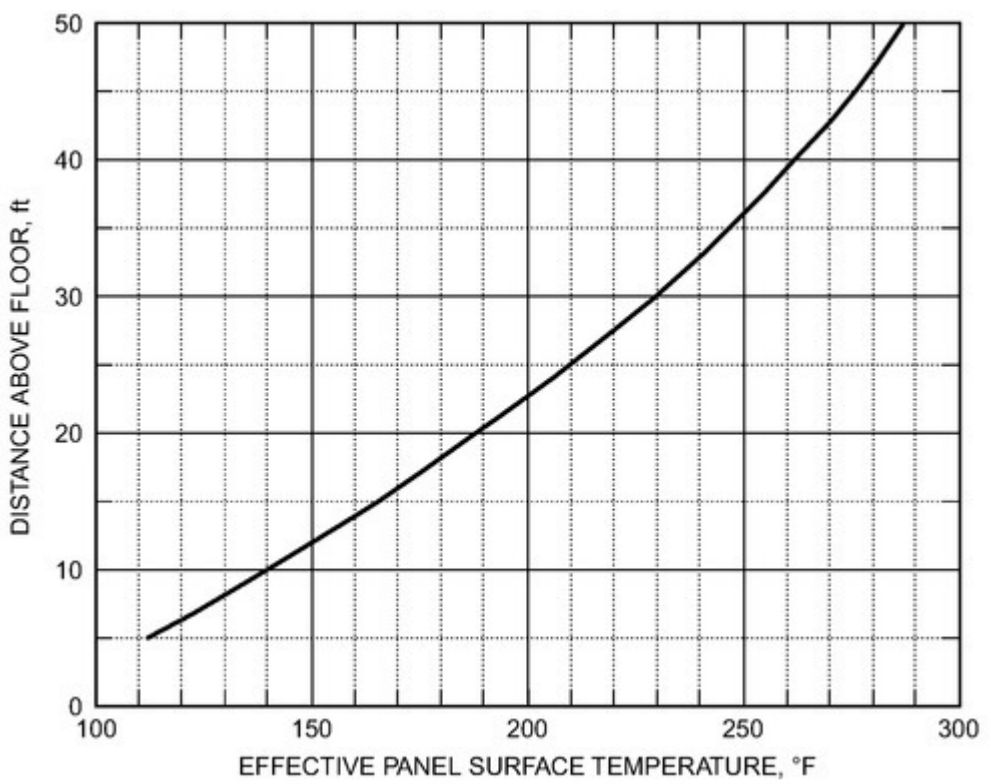


Figure 15. Permitted Design Ceiling Surface Temperatures at Various Ceiling Heights

Metal heating panels, hydronic and electric, are applied to building perimeter spaces for heating in much the same way as finned-tube convectors. They can also be integrated into the ceiling design to provide a narrow band of panel heating around the perimeter of the building. The panel system offers advantages over baseboard or overhead air in appearance, comfort, operating efficiency and cost, maintenance, and product life. ASHRAE *Standard 138* discusses testing and rating of ceiling panels.

3.2 EMBEDDED SYSTEMS WITH TUBING IN CEILINGS, WALLS, OR FLOORS

Layout and design of embedded systems for heating and cooling begin early in the job. The type and location chosen influences the design and, conversely, thermal considerations may dictate which system type to use. ISO *Standard 11855-2012* provides an in-depth description of the design, installation, and control of embedded radiant systems. One of the following types of construction is generally used:

- Pipe or tube is embedded in the lower portion of a concrete slab, generally within 1 in. of its lower surface ([Figure 16](#)). If plaster is to be applied to the concrete, the piping may be placed directly on the wood forms. If the slab is to be used without plaster finish, the piping should be installed not less than 0.75 in. above the undersurface of the slab. The minimum coverage must comply with local building code requirements.
- Pipe or tube is embedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceiling, the lath and heating coils are securely wired to the supporting members so that the lath is below, but in good contact with, the coils. Plaster is then applied to the metal lath, carefully embedding the coil as shown in [Figure 17](#).
- Smaller-diameter copper or thermoplastic tube is attached to the underside of wire or gypsum lath. Plaster is then applied to the lath to embed the tube, as shown in [Figure 18](#).
- Other forms of ceiling construction are composition board, wood paneling, etc., with warm-water piping, tube, or channels built into the panel sections.

Coils are usually laid in a sinusoidal pattern, although some header or grid-type coils have been used in ceilings. Coils are typically thermoplastic tube, spaced from 4 to 12 in. on centers, depending on the required heat flux, pipe or tube size, and other factors.

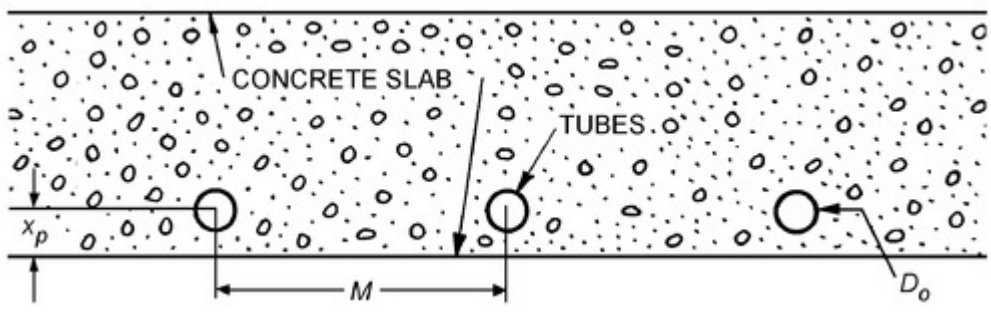


Figure 16. Coils in Structural Concrete Slab

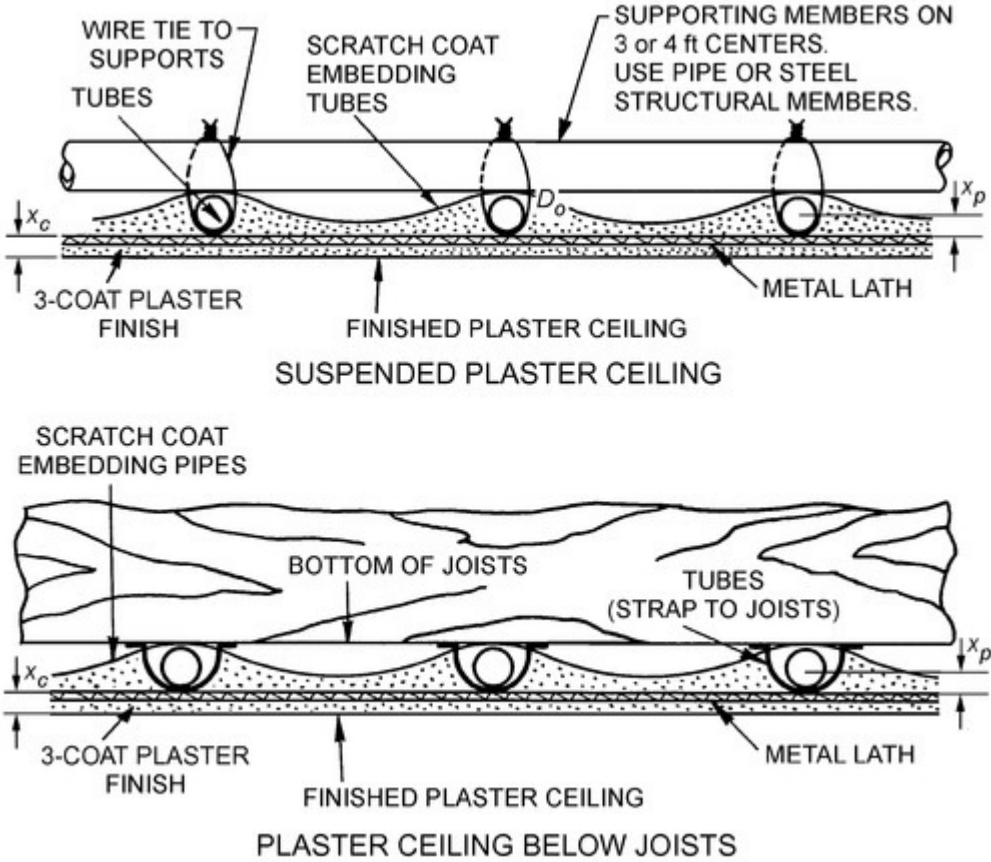


Figure 17. Coils in Plaster Above Lath

Where plastering is applied to pipe coils, a standard three-coat gypsum plastering specification is followed, with a minimum of 3/8 in. of cover below the tubes when they are installed below the lath. Generally, the surface temperature of plaster panels should not exceed 120°F, which may be satisfied by limiting the average water temperature to a maximum temperature of 140°F. Insulation should be placed above the coils to reduce back loss, which is the difference between heat supplied to the coil and net useful heat output to the heated indoor space.

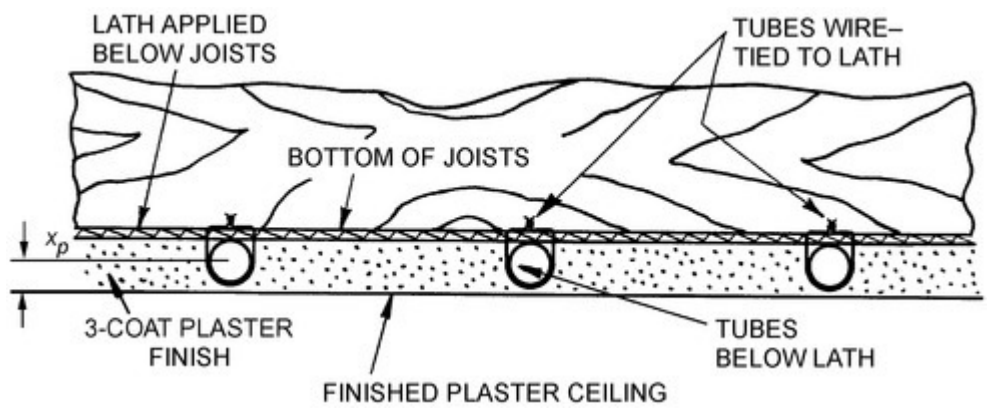


Figure 18. Coils in Plaster Below Lath

To protect the plaster installation and to ensure proper air drying, heat must not be applied to the panels for two weeks after all plastering work has been completed. When the system is started for the first time, water supplied to the panels should not be more than 20°F above the prevailing indoor air temperature and not in excess of 90°F.

Water should be circulated at this temperature for about two days, and then increased at a rate of about 5°F per day to 140°F.

During the air-drying and preliminary warm-up periods, there should be adequate ventilation to carry moisture from the panels. No paint or paper should be applied to the panels before these periods have been completed or while the panels are being operated. After paint and paper have been applied, an additional shorter warm-up period, similar to first-time starting, is also recommended.

Hydronic Wall Panels

Although piping embedded in walls is not as widely used as for floor and ceiling panels, it can be constructed by any of the methods outlined for ceilings or floors. Its design is similar to other hydronic panels [see [Equations \(22\) to \(26\)](#), [Equations \(5\)](#) and [\(12\)](#)] give the heat flux at the surface of wall panels.

Hydronic Floor Panels

Interest has increased in floor heating with the introduction of nonmetallic tubing and new design, application, and control techniques. Whichever method is used for optimum floor output and comfort, it is important that heat be evenly distributed over the floor. Spacing is generally 4 to 12 in. on centers for the coils. Wide spacing under tile or bare floors can cause uneven surface temperatures.

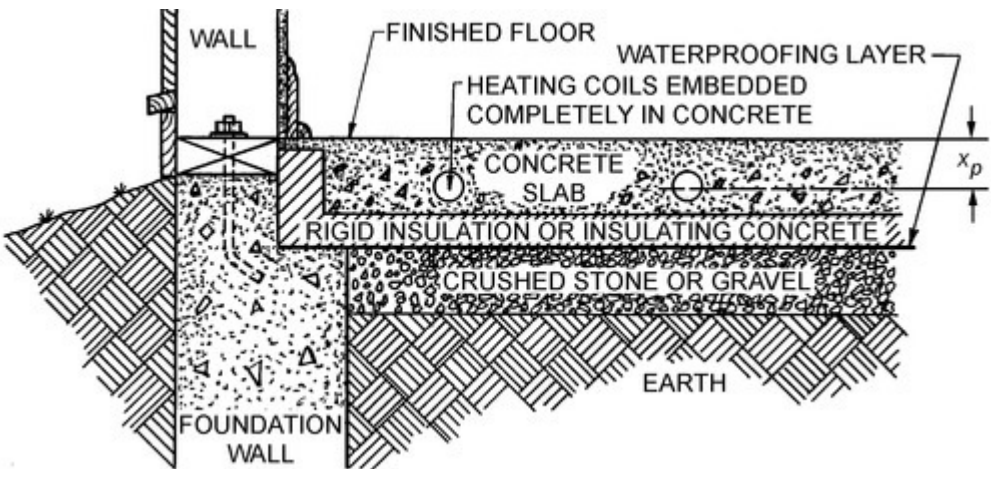


Figure 19. Coils in Floor Slab on Grade

Embedded Tubes or Pipes in Concrete Slab. Thermoplastic, rubber, ferrous, and nonferrous pipes, or composite tubes (e.g., thermoplastic tubes with aluminum sleeves) are used in floor slabs that rest on grade. Hydronic coils are constructed as sinusoidal-continuous coils or arranged as header coils with a spacing of 6 to 18 in. on centers. The coils are generally installed with 1.5 to 4 in. of cover above them. Insulation is recommended to reduce perimeter and back losses where the outer surface is exposed to the exterior. [Figure 19](#) shows application of hydronic coils in slabs resting on grade. Coils should be embedded completely and should not rest on an interface. Any supports used for positioning heating coils should be nonabsorbent and inorganic. Reinforcing steel, angle iron, pieces of pipe or stone, or concrete mounds can be used. No wood, brick, concrete block, or similar materials should support coils. A waterproofing layer is desirable to protect insulation and piping.

Where coils are embedded in structural load-supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed as described for slabs resting on grade.

Embedded systems may fail sometime during their life. Adequate valves and properly labeled drawings help isolate the point of failure.

Suspended Floor Tubing or Piping. Piping may be applied on or under suspended wood floors using several construction methods. Piping may be attached to the surface of the floor and embedded in a layer of concrete or gypsum, mounted in or below the subfloor, or attached directly to the underside of the subfloor using metal panels to improve heat transfer from the piping. An alternative method is to install insulation with a reflective surface and leave an air gap of 2 to 4 in. to the subfloor. Whichever method is used for optimum floor output and comfort, it is important that heat be evenly distributed throughout the floor. Tubing generally has a spacing of 4 to 12 in. on centers. Wide spacing under tile or bare floors can cause uneven surface temperatures.

Figure 20 depicts construction with piping embedded in concrete or gypsum. The embedding material is generally 1 to 2 in. thick when applied to a wood subfloor. Gypsum products specifically designed for floor heating can generally be installed 1 to 1.5 in. thick because they are more flexible and crack-resistant than concrete. When concrete is used, it should be of structural quality to reduce cracking caused by movement of the wood frame or shrinkage. The embedding material must provide a hard, flat, smooth surface that can accommodate a variety of floor covers.

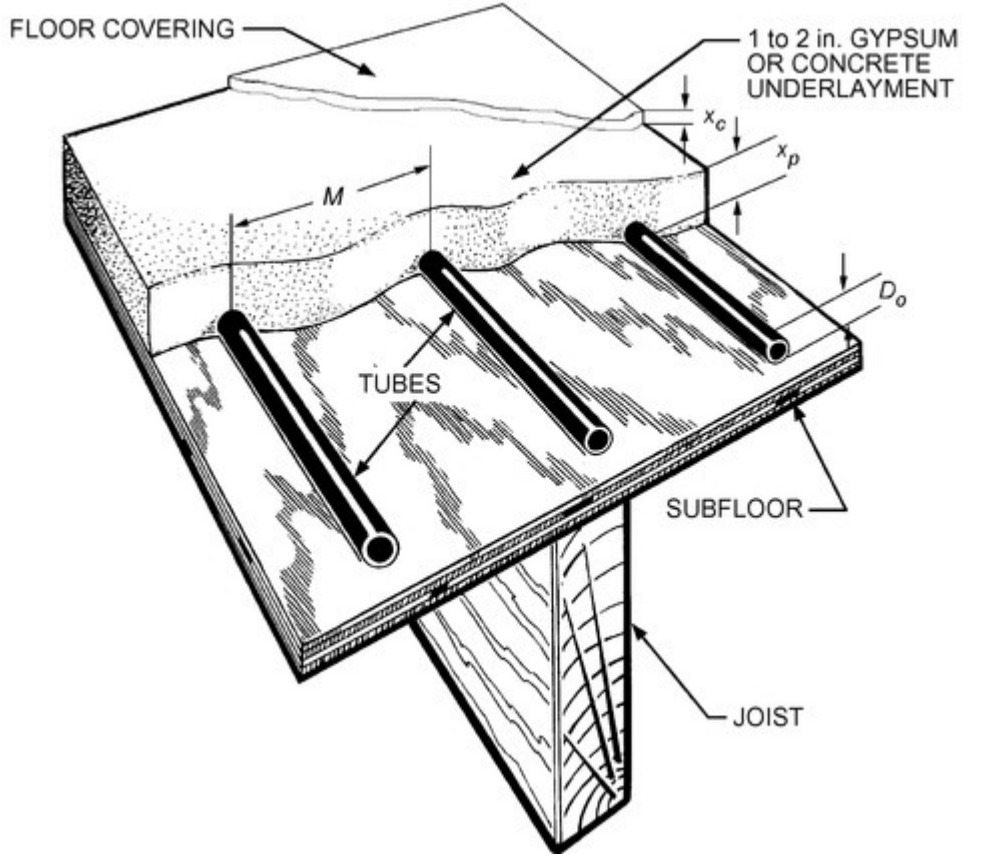


Figure 20. Embedded Tube in Thin Slab

As shown in Figure 21, tubing may also be installed in the subfloor. The tubing is installed on top of the rafters between the subflooring members. Heat diffusion and surface temperature can be improved uniformly by adding metal heat transfer plates, which spread heat beneath the finished flooring. A third construction option is to attach the tube to the underside of the subfloor with or without metal heat transfer plates. The construction is depicted in Figure 22.

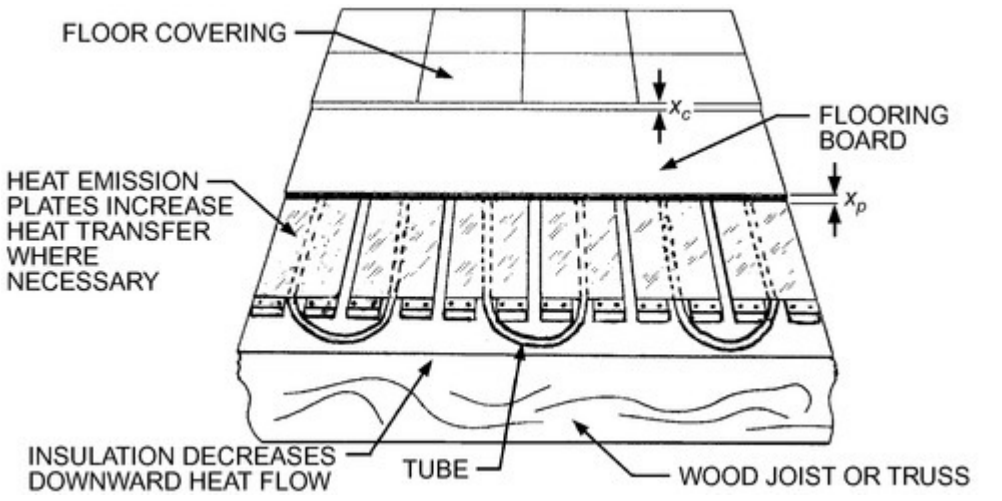


Figure 21. Tube in Subfloor

Transfer from the hot-water tube to the surface of the floor is the important consideration in all cases. The floor surface temperature affects the actual heat transfer to the space. Any hindrance between the heated water tube and the floor surface reduces system effectiveness. The method that transfers and spreads heat evenly through the subfloor with the least resistance produces the best results.

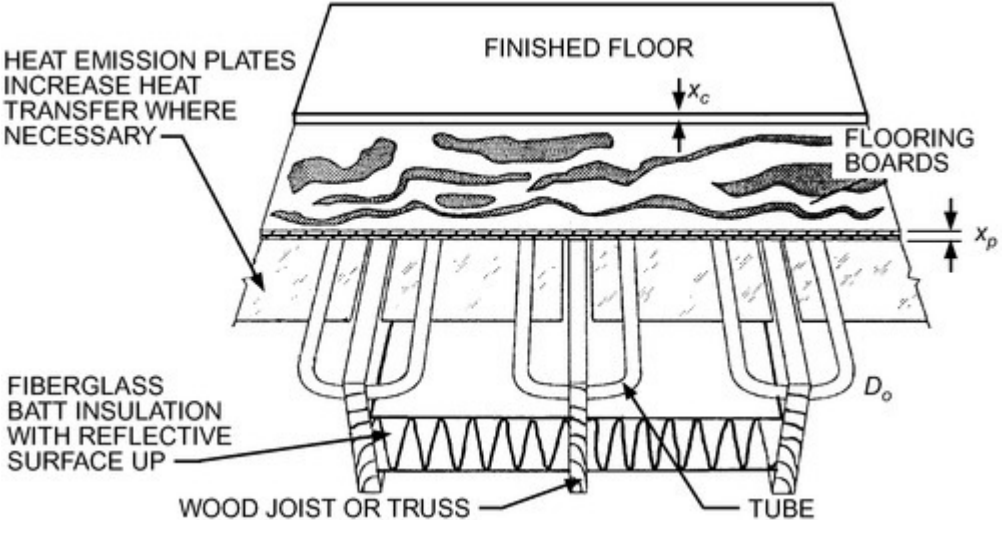


Figure 22. Tube Under Subfloor

3.3 ELECTRICALLY HEATED RADIANT SYSTEMS

Several panel systems convert electrical energy to heat, raising the temperature of conditioned indoor surfaces and the indoor air. These systems are classified by the temperature of the heated system. Higher-temperature surfaces require less area to maintain occupant comfort. Surface temperatures are limited by the ability of the materials to maintain their integrity at elevated temperatures. The maximum effective surface temperature of floor panels is limited to what is comfortable to occupants' feet.

Electric Ceiling Panels

Prefabricated Electric Ceiling Panels. These panels are available in sizes 1 to 6 ft wide by 2 to 12 ft long by 0.5 to 2 in. thick. They are constructed with metal, glass, or semirigid fiberglass board or vinyl. Heated surface temperatures range from 100 to 300°F, with corresponding heat fluxes ranging from 25 to 100 W/ft² for 120 to 480 V services. A panel of gypsum board embedded with insulated resistance wire is also available. It is installed as part of the ceiling or between joists in contact with a ceiling. Heat flux is limited to 22 W/ft² to maintain the board's integrity by keeping the heated surface temperature below 100°F. Nonheating leads are furnished as part of the panel. Some panels can be cut to fit; others must be installed as received. Panels may be either flush or surface-mounted, and in some cases, they are finished as part of the ceiling. Rigid 2 by 4 ft panels for lay-in ceilings (Figure 23) are about 1 in. thick and weigh from 6 to 25 lb. Typical characteristics of an electric panel are listed in Table 4. Panels may also be (1) surface-mounted on gypsum board and wood ceilings or (2) recessed between ceiling joists. Panels range in size from 4 ft wide to 8 ft long. Their maximum power output is 95 W/ft².

Electric Ceiling Panel Systems. These systems are laminated conductive coatings, printed circuits, or etched elements nailed to the bottom of ceiling joists and covered by 1/2 in. gypsum board. Heat flux is limited to 18 W/ft². In some cases, the heating element can be cut to fit available space. Manufacturers' instructions specify how to connect the system to the electric supply. Appropriate codes should be followed when placing partitions, lights, and air grilles adjacent to or near electric panels.

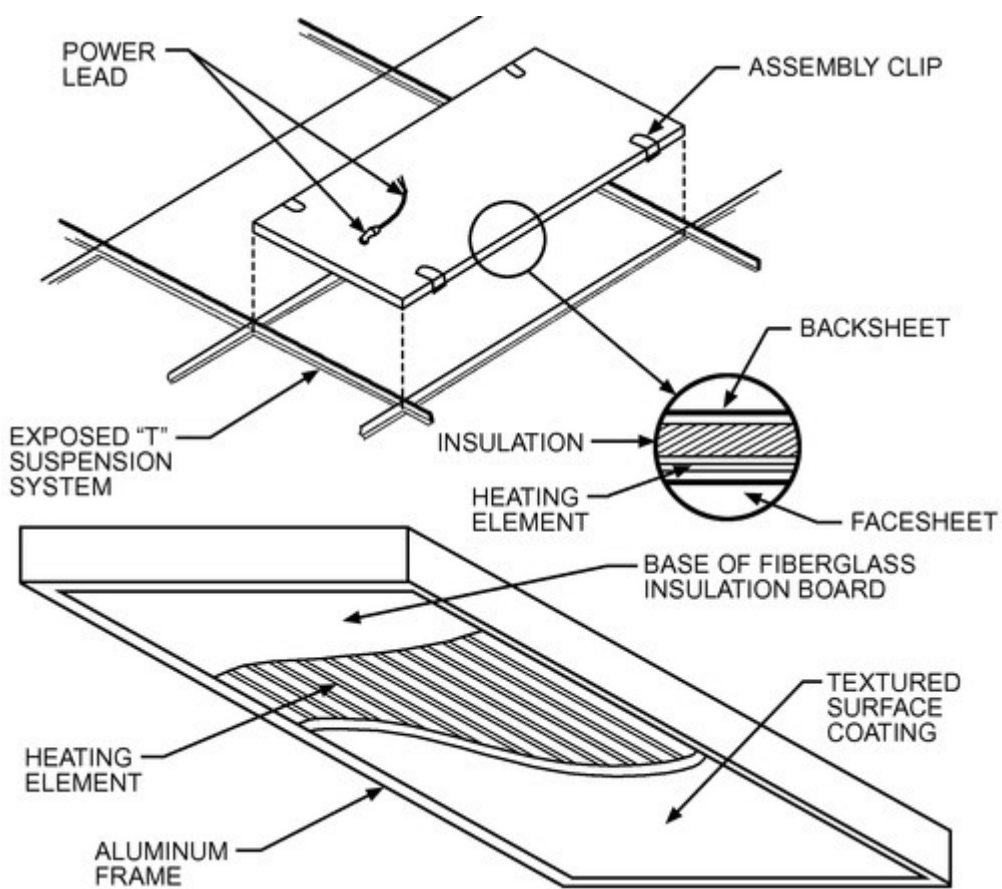


Figure 23. Electric Heating Panels

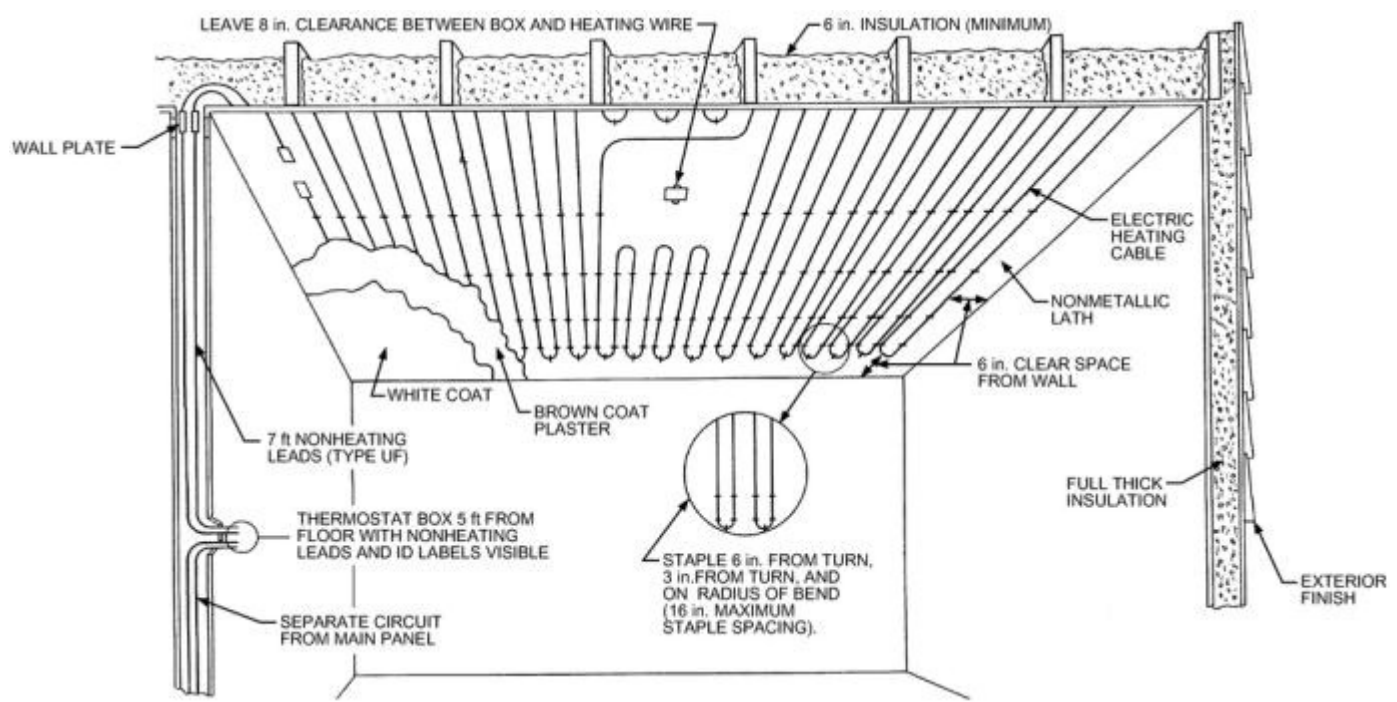


Figure 24. Electric Heating for Wet Plaster Ceiling

Electric Cables Embedded in Ceilings. Electric heating cables for embedded or laminated ceiling panels are factory-assembled units furnished in standard lengths of 75 to 1800 ft. These cable lengths cannot be altered in the field. The cable assemblies are normally rated at 2.75 W per linear foot and are supplied in capacities from 200 to 5000 W in roughly 200 W increments. Standard cable assemblies are available for 120, 208, and 240 V. Each cable unit is supplied with 7 ft nonheating leads for connection at the thermostat or junction box.

Electric cables for panel heating have electrically insulated sleeves resistant to medium temperature, water absorption, aging effects, and chemical action with plaster, cement, or ceiling lath material. This insulation is normally made of polyvinyl chloride (PVC), which may have a nylon jacket. The thickness of the insulation layer is usually about 0.12 in. For plastered ceiling panels, the heating cable may be stapled to gypsum board, plaster lath, or similar fire-resistant materials with rust-resistant staples (Figure 24). With metal lath or other conducting surfaces, a coat of plaster (brown or scratch coat) is applied to completely cover the metal lath or conducting surface before the cable is attached. After the lath is fastened on and the first plaster coat is applied, each cable is tested for continuity of circuit and for insulation resistance of at least 100 kΩ measured to ground.

The entire ceiling surface is finished with a cover layer of thermally noninsulating sand plaster about 0.50 to 0.75 in. thick, or other approved noninsulating material applied according to manufacturer's specifications. The plaster is applied parallel to the heating cable rather than across the runs. While new plaster is drying, the system should not be energized, and the range and rate of temperature change should be kept low by other heat sources or by ventilation until the plaster is thoroughly cured. Vermiculite or other insulating plaster causes cables to overheat and is contrary to code provisions.

For laminated drywall ceiling panels, the heating cable is placed between two layers of gypsum board, plasterboard, or other thermally noninsulating fire-resistant ceiling lath. The cable is stapled directly to the first (or upper) lath, and the two layers are held apart by the thickness of the heating cable. It is essential that the space between the two layers of lath be completely filled with a noninsulating plaster or similar material. This fill holds the cable firmly in place and improves heat transfer between the cable and the finished ceiling. Failure to fill the space between the two layers of plasterboard completely may allow the cable to overheat in the resulting voids and may cause cable failure. The plaster fill should be applied according to manufacturer's specifications.

Table 4. Characteristics of Typical Electric Panels

Resistor material	Graphite or nichrome wire
Relative heat intensity	Low, 50 to 125 W/ft ²
Resistor temperature	180 to 350°F
Envelope temperature (in use)	160 to 300°F
Thermal-radiation-generating ratio ^a	0.7 to 0.8
Response time (heat-up)	240 to 600 s
Luminosity (visible light)	None
Thermal shock resistance	Excellent
Vibration resistance	Excellent
Impact resistance	Excellent
Resistance to drafts or wind ^b	Poor
Mounting position	Any
Envelope material	Steel alloy or aluminum
Color blindness	Very good
Flexibility	Good—wide range of heat flux, length, and voltage practical
Life expectancy	Over 10,000 h

^a Ratio of radiation heat flux to input power density (elements only).

^b May be shielded from wind effects by louvers, deep-drawn fixtures, or both.

Electric heating cables are ordinarily installed with a 6 in. nonheating border around the periphery of the ceiling. An 8 in. clearance must be provided between heating cables and the edges of the outlet or junction boxes used for surface-mounted lighting fixtures. A 2 in. clearance must be provided from recessed lighting fixtures, trim, and ventilating or other openings in the ceiling. Heating cables or panels must be installed only in ceiling areas that are not covered by partitions, cabinets, or other obstructions. However, it is permissible for a single run of isolated embedded cable to pass over a partition.

The *National Electrical Code*[®] (NFPA *Standard* 70) requires that all general electrical power and lighting wiring be run above the thermal insulation or at least 2 in. above the heated ceiling surface, or that the wiring be derated.

In drywall ceilings, the heating cable is always installed with the cable runs parallel to the joist. A 2.5 in. clearance between adjacent cable runs must be left centered under each joist for nailing. Cable runs that cross over the joist must be kept to a minimum. Where possible, these crossings should be in a straight line at one end of the indoor space.

For cable having a heat flux of 2.75 W/ft, the minimum permissible spacing is 1.5 in. between adjacent runs. Some manufacturers recommend a minimum spacing of 2 in. for drywall construction.

The spacing between adjacent runs of heating cable can be determined using the following equation:

$$M = 12A_p/C \tag{28}$$

where

M = cable spacing, in.

A_p = net panel heated area, ft²

C = length of cable, ft

Net panel area A_p in Equation (28) is the net ceiling area available after deducting the area covered by the nonheating border, lighting fixtures, cabinets, and other ceiling obstructions. For simplicity, Equation (28) contains a slight safety factor, and small lighting fixtures are usually ignored in determining net ceiling area.

Electrical resistance of the electric cable must be adjusted according to its temperature at design conditions (Ritter and Kilis 1998):

$$R' = R \frac{[1 + \alpha_o(t_d - 68)]}{[1 + \alpha_o(t_o - 68)]} \tag{29}$$

where

R = electrical resistance of electric cable at standard temperature (68°F), Ω/ft

α_o = thermal coefficient for material resistivity, °F⁻¹

α_o = thermal expansion coefficient, °F⁻¹

t_o = surface temperature of electric cable at operating conditions [see Equation (23)], °F

The 2.5 in. clearance required under each joist for nailing in drywall applications occupies one-fourth of the ceiling area if the joists are 16 in. on centers. Therefore, for drywall construction, the net area A_p must be multiplied by 0.75. Many installations have a spacing of 1.5 in. for the first 2 ft from the cold wall. Remaining cable is then spread over the balance of the ceiling.

Electric Wall Heating

Cable embedded in walls similar to ceiling construction is used in Europe. Because of possible damage from nails driven for hanging pictures or from building alterations, most U.S. codes prohibit such panels. Some of the prefabricated panels described in the preceding section are also used for wall panel heating.

Electric Floor Heating

Electric heating cable assemblies such as those used for ceiling panels are sometimes used for concrete floor heating systems. Because the possibility of cable damage during installation is greater for concrete floor slabs than for ceiling panels, these assemblies must be carefully installed. After the cable has been placed, all unnecessary traffic should be eliminated until the concrete layer has been placed and hardened.

Preformed mats are sometimes used for electric floor slab heating systems. These mats usually consist of PVC-insulated heating cable woven into or attached to metallic or glass fiber mesh. Such mats are available as prefabricated assemblies in many sizes from 2 to 100 ft² and with heat fluxes from 15 to 25 W/ft². When mats are used with a thermally treated cavity beneath the floor, a heat storage system is provided, which may be controlled for off-peak heating.

Mineral-insulated (MI) heating cable is another effective method of slab heating. MI cable is a small-diameter, highly durable, flexible heating cable composed of solid electric-resistance heating wire or wires surrounded by tightly compressed magnesium oxide electrical insulation and enclosed by a metal sheath. MI cable is available in stock assemblies in a variety of standard voltages, heat fluxes (power densities), and lengths. A cable assembly consists of the specified length of heating cable, waterproof hot-cold junctions, 7 ft cold sections, UL-approved end fittings, and connection leads. Several standard MI cable constructions are available, such as single conductor, twin conductor, and double cable. Custom-designed MI heating cable assemblies can be ordered for specific installations.

Other outer-sleeve materials that are sometimes specified for electric floor heating cable include (1) silicone rubber, (2) lead, and (3) tetrafluoroethylene (Teflon[®]).

For a given floor heating cable assembly, the required cable spacing is determined from Equation (24). In general, cable heat flux and spacing should be such that floor panel heat flux is not greater than 15 W/ft². Check the latest edition of the *National Electrical Code*[®] (NFPA *Standard* 70) and other applicable codes to obtain information on maximum panel heat flux and other required criteria and parameters.

Floor Heating Cable Installation. When PVC-jacketed electric heating cable is used for floor heating, the concrete slab is laid in two pours. The first pour should be at least 3 in. thick and, where practical, should be insulating concrete to reduce downward heat loss. For a proper bond between the layers, the finish slab should be placed within 24 h of the first pour, with a bonding grout applied. The finish layer should be between 1.5 and 2 in. thick. This top layer must not be insulating concrete. At least 1 in. of perimeter insulation should be installed as shown in Figure 25.

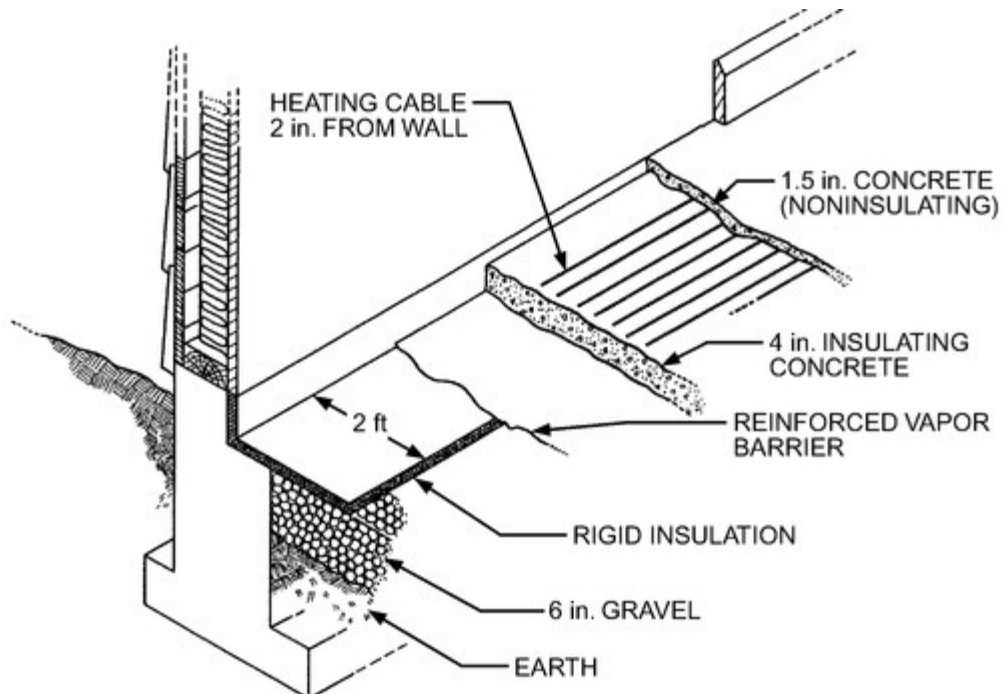


Figure 25. Electric Heating Cable in Concrete Slab

The cable is installed on top of the first pour of concrete no closer than 2 in. from adjoining walls and partitions. Methods of fastening the cable to the concrete include the following:

- Staple the cable to wood nailing strips fixed in the surface of the rough slab. Dabs of cement, plaster of paris, or tape maintain the predetermined cable spacing.
- In light or uncured concrete, staple the cable directly to the slab using hand-operated or powered stapling machines.
- Nail special anchor devices to the first slab to hold the cable in position while the top layer is being poured.

Preformed mats can be embedded in the concrete in a continuous pour. The mats are positioned in the area between expansion and/or construction joints and electrically connected to a junction box. The slab is poured to within 1.5 to 2 in. of the finished level. The surface is rough-screeded and the mats placed in position. The final cap is applied immediately. Because the first pour has not set, there is no adhesion problem between the first and second pours, and a monolithic slab results. A variety of contours can be developed by using heater wire attached to glass fiber mats. Allow for circumvention of obstructions in the slab.

MI electric heating cable can be installed in concrete slab using either one or two pours. For single-pour applications, the cable is fastened to the top of the reinforcing steel before the pour is started. For two-layer applications, the cable is laid on top of the bottom structural slab and embedded in the finish layer. Proper spacing between adjacent cable runs is maintained by using prepunched copper spacer strips nailed to the lower slab.

4. DESIGN PROCEDURE

Panel design requires determining panel area, type, and arrangement as well as the supply water temperature and flow rate. Panel performance is directly related to indoor space conditions. Air-side design also must be established. Heating and cooling loads may be calculated by procedures covered in [Chapters 17](#) and [18 of the 2013 ASHRAE Handbook—Fundamentals](#). For cooling load calculations, the procedure based on the heat balance (HB) method is recommended, because other simplified methods developed for all-air systems [such as the radiant time series (RTS) method] are not as appropriate for radiant-system applications (Feng et al. 2013b, 2014). The procedure is as follows:

Sensible Cooling

1. Determine indoor design dry-bulb temperature, relative humidity, and dew point.
2. Calculate sensible and latent heat gains.
3. Establish minimum supply air quantity for ventilation.
4. Calculate latent cooling available from supply air.
5. Calculate sensible cooling available from supply air.
6. Determine remaining sensible cooling load to be satisfied by panel system.
7. Determine minimum permissible effective cooling panel surface temperature that will not lead to surface condensation at design conditions.
8. Determine AUST.
9. Determine necessary panel area for remaining sensible cooling.
10. Determine average panel cooling water (brine) temperature for given tube spacing, or determine necessary tube spacing if average panel cooling water (brine) temperature is known.

Sensible Heating

1. Designate indoor design dry-bulb temperature for panel heating.
2. Calculate room heat loss.
3. Determine AUST. Use [Equation \(14\)](#) to find surface temperatures of exterior walls and exposed floors and ceilings. Interior walls are assumed to have surface temperatures equal to indoor air temperature.
4. Calculate required effective surface temperature of panel. Refer to [Figures 9](#) and [10](#) if AUST does not greatly differ from indoor air temperature. Otherwise, use [Equations \(5\)](#), [\(9\)](#), [\(10\)](#), [\(11\)](#), and [\(12\)](#) or refer to [Figures 1](#) and [3](#).
5. Determine panel area. Refer to [Figures 9](#) and [10](#) if AUST does not vary greatly from indoor air temperature.
6. Refer to manufacturers' data for panel surface temperatures higher than those given in [Figures 9](#) and [10](#). For panels with several covers, average temperature of each cover and effective panel surface temperature must be calculated and compared to temperature-withstanding capacity for continuous operation of every cover material. For this purpose, [Equation \(23\)](#) may be used: add thermal resistances of all cover layers between panel surface and cover layer in question, then multiply by q and add to first two terms in [Equation \(23\)](#). This is t_{vi} , approximated temperature of the particular layer at design.
7. Determine tube spacing for a given average water temperature or select electric cable properties or electric mat size.
8. In a hydronic panel system, if tube spacing is known, determine required average water temperature.
9. Design panel arrangement.

Other Steps Common for Sensible Heating and Cooling

1. Check thermal comfort requirements in the following steps [see [Chapter 9 of the 2013 ASHRAE Handbook—Fundamentals](#) and NRB (1981)].
 - a. Determine occupant's clothing insulation value and metabolic rate (see [Tables 4](#), [7](#), and [8 in Chapter 9 of the 2013 ASHRAE Handbook—Fundamentals](#)).
 - b. Determine optimum operative temperature at coldest point in conditioned space (see the Comfort Equations for Radiant Heating section in [Chapter 9 of the 2013 ASHRAE Handbook—Fundamentals](#). Note that the same equations may be adopted for panel cooling).
 - c. Determine MRT at the coldest point in the conditioned space [see [Chapter 16](#) and Fanger (1972)]. *Note:* If indoor air velocity is less than 1.3 fps and MRT is less than 122°F, operative temperature may be approximated as the average of MRT and t_a .
 - d. From the definition of operative temperature, establish optimum indoor design air temperature at coldest point in the room. If optimum indoor design air temperature varies greatly from designated design temperature, designate a new temperature.
 - e. Determine MRT at hottest point in conditioned space.
 - f. Calculate operative temperature at hottest point in conditioned space.
 - g. Compare operative temperatures at hottest and coldest points. For light activity and normal clothing, the acceptable operative temperature range is 68 to 75°F [see NRB (1981) and ANSI/ASHRAE *Standard* 55-2013]. If the range is not acceptable, the panel system must be modified.
 - h. Calculate radiant temperature asymmetry (NRB 1981). Acceptable ranges are less than 9°F for warm ceilings, 27°F for cool ceilings, 18°F for cool walls, and 49°F for warm walls at 10% local discomfort dissatisfaction (ANSI/ASHRAE *Standard* 55-2013).
2. Determine water flow rate and pressure drop. Refer to manufacturers' guides for specific products, or use the guidelines in [Chapter 22 of the 2013 ASHRAE Handbook—Fundamentals](#). [Chapter 13](#) of this volume also has information on hydronic heating and cooling systems.
3. The supply and return manifolds must be carefully designed. If there are circuits of unequal coil lengths, the following equation may be used (Hansen 1985; Kilis 1998) for a circuit i connected to a manifold with n circuits:

$$Q_i = (L_{eq}/L_i)^{1/r} Q_{tot} \tag{30}$$

where

$$L_{eq} = \left[\sum_{i=1}^n L_i^{1-r} \right]^{-1/r}, \text{ ft}$$

Q_i = flow rate in circuit i , gpm

Q_{tot} = total flow rate in supply manifold, gpm

L_i = coil length of hydronic circuit i , ft

r = 1.75 for hydronic panels (Segenthaler 1995)

Application, design, and installation of panel systems have certain requirements and techniques:

- As with any hydronic system, look closely at the piping system design. Piping should be designed to ensure that water of the proper temperature and in sufficient quantity is available to every grid or coil at all times. Proper piping and system design should minimize the detrimental effects of oxygen on the system. Reverse-return systems should be considered to minimize balancing problems.
- Hydronic panels can be used with two- and four-pipe distribution systems. [Figure 26](#) shows a typical system arrangement. It is common to design for a 20°F temperature drop for heating across a given grid and a 5°F rise for cooling, but larger temperature differentials may be used, if applicable.
- Individual panels can be connected for parallel flow using headers, or for sinuous or serpentine flow. To avoid flow irregularities in a header-type grid, the water channel or lateral length should be greater than the header length. If the laterals in a header grid are forced to run in a short direction, using a combination series-parallel arrangement can solve this problem. Serpentine flow ensures a more even panel surface temperature throughout the heating or cooling zone.
- Noise from entrained air, high-velocity or high-pressure-drop devices, or pump and pipe vibrations must be avoided. Water velocities should be high enough to prevent separated air from accumulating and causing air binding. Where possible, avoid automatic air venting devices over ceilings of occupied spaces.
- Design piping systems to accept thermal expansion adequately. Do not allow forces from piping expansion to be transmitted to panels. Thermal expansion of ceiling panels must be considered.
- In hydronic systems, thermoplastic, rubber tubes, steel, or copper pipes are used widely in ceiling, wall, or floor panel construction. Where coils are embedded in concrete or plaster, no threaded joints should be used for either pipe coils or mains. Steel pipe should be the all-welded type. Copper tubing should be a soft-drawn coil. Fittings and connections should be minimized. Bending should be used to change direction. Solder-joint fittings for copper tube should be used with a medium-temperature solder of 95% tin, 5% antimony, or capillary brazing alloys. All piping should be subjected to a hydrostatic test of at least three times the working pressure. Maintain adequate pressure in embedded piping while pouring concrete.
- Placing the thermostat on a wall where it can observe both the outdoor exposed wall and the warm panel should be considered. The normal thermostat cover reacts to the warm panel, and thermal radiation from the panel to the cover tends to alter the control point so that the thermostat controls 2 to 3°F lower when the outdoor temperature is a minimum and the panel temperature is a maximum. Experience indicates that panel-heated spaces are more comfortable under these conditions than when the thermostat is located on a back wall.
- If throttling valve control is used, either the end of the main should have a fixed bypass, or the last one or two rooms on the mains should have a bypass valve to maintain water flow in the main. Thus, when a throttling valve modulates, there will be a rapid response.
- When selecting heating design temperatures for a ceiling panel surface, the design parameters are as follows:
 - Excessively high temperatures over the occupied zone cause the occupant to experience a "hot head effect."
 - Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
 - Locate ceiling panels adjacent to perimeter walls and/or areas of maximum load.
 - With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235°F surface temperature. If panels extend beyond 3 ft into the indoor space, the panel surface temperature should be limited to the values given in [Figure 15](#). The surface temperature of concrete or plaster panels is limited by construction.
- Floor panels are limited to surface temperatures of less than 84°F in occupied spaces for comfort reasons. Subfloor temperature may be limited to the maximum exposure temperature specified by the floor cover manufacturer.
- When the panel chilled-water system is started, the circulating water temperature should be maintained at indoor air temperature until the air system is completely balanced, the dehumidification equipment is operating properly, and building relative humidity is at design value.

When the panel area for cooling is greater than the area required for heating, a two-panel arrangement ([Figure 27](#)) can be used. Panel HC (heating and cooling) is supplied with hot or chilled water year-round. When chilled water is used, the controls activate panel CO (cooling only) mode, and both panels are used for cooling.

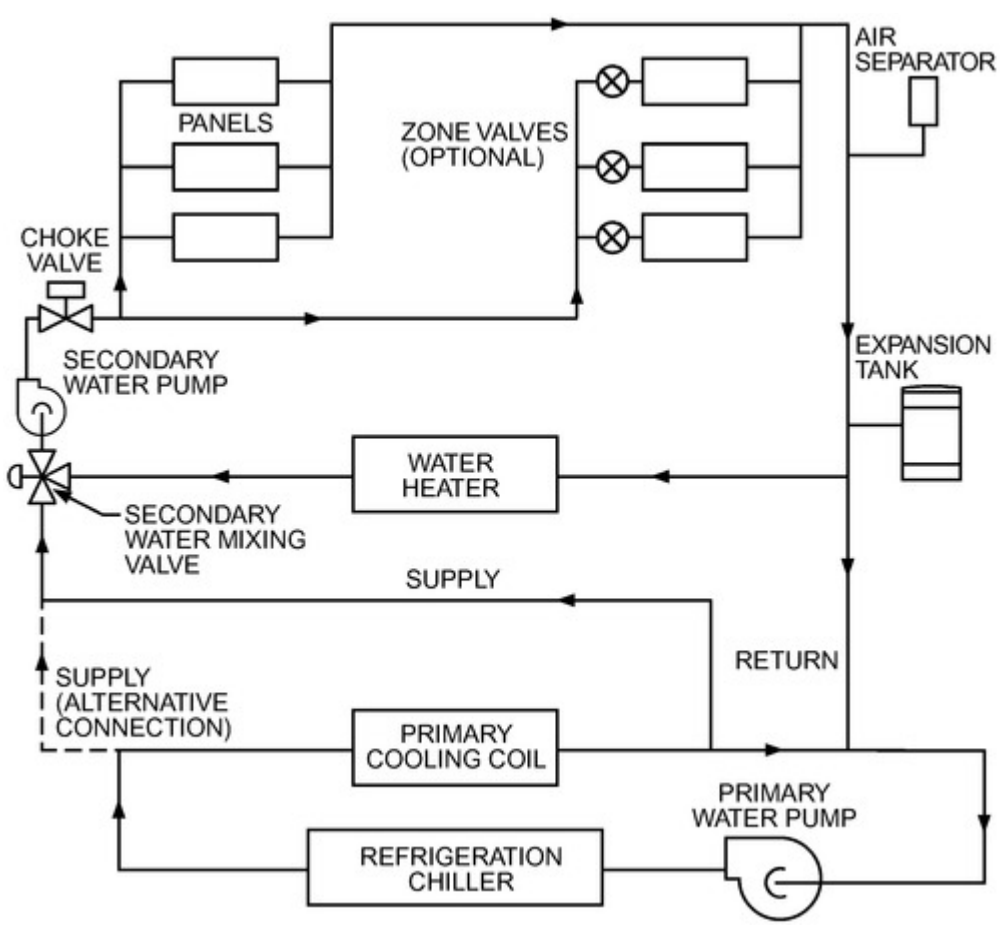


Figure 26. Primary/Secondary Water Distribution System with Mixing Control

- To prevent condensation on the cooling panels, the panel water supply temperature should be maintained at least 1°F above the indoor design dew-point temperature. This minimum difference is recommended to allow for the normal drift of temperature controls for water and air systems, and also to provide a factor of safety for temporary increase in indoor relative humidity.
- Selection of summer design indoor dew point below 50°F generally is not economical.
- The most frequently applied method of dehumidification uses cooling coils. If the main cooling coil is six rows or more, the dew point of leaving air will approach the leaving water temperature. The cooling water leaving the dehumidifier can then be used for the panel water circuit.
- Several chemical dehumidification methods are available to control latent and sensible loads separately. In one application, cooling tower water is used to remove heat from the chemical drying process, and additional sensible cooling is necessary to cool the dehumidified air to the required system supply air temperature.
- When chemical dehumidification is used, hygroscopic chemical dew-point controllers are required at the central apparatus and at various zones to monitor dehumidification.
- When cooled ceiling panels are used with a variable-air-volume (VAV) system, the air supply rate should be near maximum volume to ensure adequate dehumidification before the cooling ceiling panels are activated.

Other factors to consider when using panel systems are

- Evaluate the panel system to take full advantage in optimizing the physical building design.
- Select recessed lighting fixtures, air diffusers, hung ceilings, and other ceiling devices to provide the maximum ceiling area possible for use as panels.
- The air-side design must be able to maintain relative humidity at or below design conditions at all times to eliminate any possibility of condensation on the panels. This becomes more critical if indoor space dry- and wet-bulb temperatures are allowed to drift for energy conservation, or if duty cycling of the fans is used.
- Do not place cooling panels in or adjacent to high-humidity areas, such as near a lobby entrance, or a kitchen.
- Anticipate thermal expansion of the ceiling and other devices in or adjacent to the ceiling.
- The design of operable windows should discourage unauthorized opening.

4.1 CONTROLS

Automatic controls for panel heating may differ from those for convective heating because of the thermal inertial characteristics of the panel and the increase in mean radiant temperature in the space under increasing loads. However, low-mass systems using thin metal panels or thin underlay with low thermal heat capacity may be successfully controlled with conventional control technology using indoor sensors. Many of the control principles for hot-water heating systems described in [Chapters 13](#) and [15](#) also apply to panel heating. Because panels do not depend on air-side equipment to distribute energy, many control methods have been used successfully; however, a control interface between heating and cooling should be installed to prevent simultaneous heating and cooling.

High-thermal-mass panels such as concrete slabs require a control approach different from that for low-mass panels. Because of thermal inertia, significant time is required to bring such massive panels from one operating point to another, say from vacation setback to standard operating conditions. This will result in long periods of discomfort during the delay, then possibly periods of uncomfortable and wasteful overshoot. Careful economic analysis may reveal that nighttime setback is not warranted.

Once a slab is at operating conditions, the control strategy should be to supply the slab with heat at the rate that heat is being lost from the space (MacCluer et al. 1989). For hydronic slabs with constant circulator flow rates, this means modulating the temperature difference between the outgoing and returning water; this is accomplished with mixing valves, fuel modulation, or, for constant thermal power sources, pulse-width modulation (on/off control). For some applications, more advanced control techniques, such as model predictive control, may be used to optimize the balance of thermal comfort and energy consumption (Feng et al. 2015; Okde-wurtel et al. 2012). Slabs with embedded electric cables can be controlled by pulse-width modulators such as the common round thermostat with anticipator or its solid-state equivalent.

Outdoor reset control, another widely accepted approach, measures the outdoor air temperature, calculates the supply water temperature required for steady operation, and operates a mixing valve or boiler to achieve that supply water temperature. If the heating load of the controlled space is primarily a function of outdoor air temperature, or indoor temperature measurement of the controlled space is impractical, then outdoor reset control alone is an acceptable control strategy. When other factors such as solar or internal gains are also significant, indoor temperature feedback should be added to the outdoor reset.

In all panel applications, precautions must be taken to prevent excessive temperatures. A manual boiler bypass or other means of reducing the water temperature may be necessary to prevent new panels from drying out too rapidly.

Sensible Cooling Controls

The average water temperature in the hydronic circuit of radiant systems can be controlled either by mixing, by heat exchange, or by using the water leaving the dehumidifier. Other considerations are listed in the section on General Design Considerations. It is imperative to dry out the building space before starting the panel water system, particularly after extended down periods such as weekends. Such delayed starting action should be automated.

Radiant cooling systems require the following basic areas of temperature control: (1) exterior zones; (2) areas under exposed roofs, to compensate for transmission and solar loads; and (3) each typical interior zone, to compensate for internal loads. For optimum results, each area of the building with significantly different heating or cooling loads should be independently controlled.

Temperature control of the indoor air and panel water supply should not be a function of the outdoor weather. Normal thermostat drift is usually adequate compensation for the slightly lower temperatures desirable during winter weather. This drift should result in an indoor air temperature change of no more than 1.5°F. Control of interior zones is best accomplished by devices that reflect the actual presence of the internal load elements. Frequently, time clocks and current-sensing devices are used on lighting feeders.

Heating Slab Controls

In comfort heating, the effective surface temperature of a heated floor slab is held to a maximum of 80 to 84°F. As a result, when the heated slab is the primary heating system, thermostatic controls sensing air temperature should not be used to control the heated slab temperature; instead, the heating system should be wired in series with a slab-sensing thermostat. The remote sensing thermostat in the slab acts as a limit switch to control maximum surface temperatures allowed on the slab. The ambient sensing thermostat controls the comfort level. For supplementary slab heating, as in kindergarten floors, a remote sensing thermostat in the slab is commonly used to tune in the desired comfort level. Indoor-outdoor thermostats are used to vary the floor temperature inversely with the outdoor temperature. If the building heat loss is calculated for an outdoor temperature between 70 to 0°F, and the effective floor temperature range is maintained between 70 to 84°F with a remote sensing thermostat, the ratio of change in outdoor temperature to the heated slab temperature is 70:14, or 5:1. This means that a 5°F drop in outdoor temperature requires a 1°F increase in the slab temperature. An ambient sensing thermostat is used to vary the ratio between outdoor and slab temperatures. A time clock is used to control each heating zone if off-peak slab heating is desirable.

In heating systems, lowering nighttime air and surface temperatures can produce less satisfactory results in thermally massive radiant systems such as embedded tubing in concrete floors. These systems cannot respond to a quick increase or decrease in heating demand within the relatively short time required, resulting in a very slow reduction of the space temperature at night and a correspondingly slow pickup in the morning. Light panels, such as plaster or metal ceilings and walls, may respond to changes in demand quickly enough for satisfactory results from lowered nighttime air and panel temperatures. Berglund et al. (1982) demonstrated the speed of response on a metal ceiling panel to be comparable to that of convection systems. However, in cooling operation, thermally massive embedded systems can offer cost- and energy-saving opportunities through nighttime precooling when electricity costs are typically lower, as are outdoor dry- and wet-bulb temperatures.

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For additional literature on high-temperature radiant heating, see the Bibliography in Chapter 16.

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