

RADIANT HEATING AND COOLING

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RADIANT heating and cooling applications are classified as panel heating or cooling if the panel surface temperature is below 300°F, and as low-, medium-, or high-intensity radiant heating if the surface or source temperature exceeds 300°F. In thermal radiation heat is transferred by electromagnetic waves that travel in straight lines, and can be reflected. Thermal radiation principally occurs between surfaces or between a source and a surface. In a conditioned space, air is not heated or cooled in this process. Because of these characteristics, radiant systems are effective for both spot heating and space heating or cooling requirements for an entire building.

Sensible heating loads may be reduced by 4 to 16% compared to ASHRAE standard design load. Percent reduction increases with the air change rate (Suryanarayana and Howell 1990).

LOW-, MEDIUM-, AND HIGH-INTENSITY INFRARED HEATING

Low-, medium-, and high-intensity infrared heaters are compact, self-contained direct-heating devices used in hangars, warehouses, factories, greenhouses, and gymnasiums, as well as in areas such as loading docks, racetrack stands, outdoor restaurants, animal breeding areas, swimming pool lounge areas, and areas under marquees. Infrared heating is also used for snow melting and freeze protection (e.g., on stairs and ramps) and process heating (e.g., paint baking and drying). An infrared heater may be electric, gas-fired, or oil-fired and is classified by the source temperature as follows:

- Low-intensity (source temperatures to 1200°F)
- Medium-intensity (source temperatures to 1800°F)
- High-intensity (source temperatures to 5000°F)

The source temperature is determined by such factors as the source of energy, the configuration, and the size. Reflectors can be used to direct the distribution of thermal radiation in specific patterns. [Chapter 15 of the ASHRAE Handbook—HVAC Systems and Equipment](#) covers radiant equipment in detail.

PANEL HEATING AND COOLING

Panel heating and cooling systems provide a comfortable environment by controlling surface temperatures and minimizing air motion within a space. They include the following designs:

- Ceiling panels
- Embedded hydronic tubing or attached piping in ceilings, walls, or floors
- Air-heated or cooled floors or ceilings
- Electric ceiling or wall panels
- Electric heating cable or wire mats in ceilings or floors
- Deep heat, a modified storage system using electric heating cable or embedded hydronic tubing in ceilings or floors

The preparation of this chapter is assigned to TC 6.5, Radiant Space Heating and Cooling.

In these systems, generally more than 50% of the heat transfer between the temperature-controlled surface and other surfaces is by thermal radiation. Panel heating and cooling systems are used in residences, office buildings, classrooms, hospital patient rooms, swimming pool areas, repair garages, and in industrial and warehouse applications. Additional information is available in [Chapter 6 of the ASHRAE Handbook—HVAC Systems and Equipment](#).

Some radiant panel systems, referred to as hybrid HVAC systems, combine radiant heating and cooling with central air conditioning (Scheatzle 2003). They are used more for cooling than for heating (Wilkins and Kosonen 1992). The controlled-temperature surfaces may be in the floor, walls, or ceiling, with temperature maintained by electric resistance or circulation of water or air. The central station can be a basic, one-zone, constant-temperature, or constant-volume system, or it can incorporate some or all the features of dual-duct, reheat, multizone, or variable-volume systems. When used in combination with other water/air systems, radiant panels provide zone control of temperature and humidity.

Metal ceiling panels may be integrated into the central heating and cooling system to provide individual room or zone heating and cooling. These panels can be designed as small units to fit the building module, or they can be arranged as large continuous areas for economy. Room thermal conditions are maintained primarily by direct transfer of radiant energy, normally using four-pipe hot and chilled water. These systems have generally been used in hospital patient rooms. Metal ceiling panel systems are discussed in [Chapter 6 of the ASHRAE Handbook—HVAC Systems and Equipment](#).

ELEMENTARY DESIGN RELATIONSHIPS

When considering radiant heating or cooling for human comfort, the following terms describe the temperature and energy characteristics of the total radiant environment:

- **Mean radiant temperature** (MRT) \bar{t}_r is the temperature of an imaginary isothermal black enclosure in which an occupant would exchange the same amount of heat by radiation as in the actual nonuniform environment.
- **Ambient temperature** t_a is the temperature of the air surrounding the occupant.
- **Operative temperature** t_o is the temperature of a uniform isothermal black enclosure in which the occupant would exchange the same amount of heat by radiation and convection as in the actual nonuniform environment.

For air velocities less than 80 fpm and mean radiant temperatures less than 120°F, the operative temperature is approximately equal to the adjusted dry-bulb temperature, which is the average of the air and mean radiant temperatures.

- **Adjusted dry-bulb temperature** is the average of the air temperature and the mean radiant temperature at a given location. The adjusted dry-bulb temperature is approximately equivalent to the operative temperature for air motions less than 80 fpm and mean radiant temperatures less than 120°F.

• **Effective radiant flux (ERF)** is defined as the net radiant heat exchanged at ambient temperature t_a between an occupant, whose surface is hypothetical, and all enclosing surfaces and directional heat sources and sinks. Thus, ERF is the net radiant energy received by the occupant from all surfaces and sources whose temperatures differ from t_a . ERF is particularly useful in high-intensity radiant heating applications.

The relationship between these terms can be shown for an occupant at surface temperature t_{sf} , exchanging sensible heat H_m in a room with ambient air temperature t_a and mean radiant temperature \bar{t}_r . The linear radiative and convective heat transfer coefficients are h_r and h_c , respectively; the latter is a function of the relative movement between the occupant and air movement V . The heat balance equation is

$$H_m = h_r(t_{sf} - \bar{t}_r) + h_c(t_{sf} - t_a) \quad (1)$$

During thermal equilibrium, H_m is equal to metabolic heat minus work and evaporative cooling by sweating. By definition of operative temperature,

$$H_m = (h_r + h_c)(t_{sf} - t_o) = h(t_{sf} - t_o) \quad (2)$$

The combined heat transfer coefficient is h , where $h = h_r + h_c$. Using Equations (1) and (2) to solve for t_o yields

$$t_o = \frac{h_r \bar{t}_r + h_c t_a}{h_r + h_c} = t_a + \left(\frac{h_r}{h}\right)(\bar{t}_r - t_a) \quad (3)$$

Thus, t_o is an average of \bar{t}_r and t_a , weighted by their respective heat transfer coefficients; it represents how people sense the thermal level of their total environment as a single temperature.

Rearranging Equation (1) and substituting $h - h_r$ for h_c ,

$$H_m + h_r(\bar{t}_r - t_a) = h(t_{sf} - t_a) \quad (4)$$

where $h_r(\bar{t}_r - t_a)$ is, by definition, the effective radiant flux (ERF) and represents the radiant energy absorbed by the occupant from all sources whose temperatures differ from t_a .

The principal relationships between \bar{t}_r , t_a , t_o , and ERF are as follows:

$$\text{ERF} = h_r(\bar{t}_r - t_a) \quad (5)$$

$$\text{ERF} = h(t_o - t_a) \quad (6)$$

$$\bar{t}_r = t_a + \text{ERF}/h_r \quad (7)$$

$$t_o = t_a + \text{ERF}/h \quad (8)$$

$$\bar{t}_r = t_a + (h_r/h_r)(t_o - t_a) \quad (9)$$

$$t_o = t_a + (h_r/h)(\bar{t}_r - t_a) \quad (10)$$

In Equations (1) to (10), the radiant environment is treated as a blackbody with temperature \bar{t}_r . The effect of the emittance of the source, radiating at absolute temperature in degrees Rankine, and the absorptance of skin and clothed surfaces is reflected in the effective values of \bar{t}_r or ERF and not in h_r , which is generally given by

$$h_r = 4\sigma f_{eff} \left[\frac{(\bar{t}_r + t_a)}{2} + 460 \right]^3 \quad (11)$$

where

h_r = linear radiative heat transfer coefficient, Btu/h · ft² · °F

f_{eff} = ratio of radiating surface of the human body to its total DuBois surface area $A_D = 0.71$

σ = Stefan-Boltzmann constant = 0.1712×10^{-8} Btu/h · ft² · °R⁴

The convective heat transfer coefficient for an occupant depends on the relative velocity between the occupant and the surrounding air, as well as the occupant's activity:

• If the occupant is walking in still air,

$$h_c = 0.092 V^{0.53} \quad \{100 < V < 400 \text{ fpm}\} \quad (12a)$$

where V is the occupant's walking speed.

• If the occupant is sedentary with moving air,

$$h_c = 0.061 V_a^{0.6} \quad \{40 < V_a < 800 \text{ fpm}\} \quad (12b)$$

$$h_c = 0.55 \quad \{0 < V_a < 40 \text{ fpm}\} \quad (12c)$$

More information about h_c may be found in [Chapter 8 of the ASHRAE Handbook—Fundamentals](#).

When $\bar{t}_r > t_a$, ERF adds heat to the body; when $t_a > \bar{t}_r$, heat is lost from the body because of thermal radiation. ERF is independent of the occupant's surface temperature and can be measured directly by a black globe thermometer or any blackbody radiometer or flux meter using the ambient air t_a as its heat sink.

In these definitions and for radiators below 1700°F (2160°R), the body clothing and skin surface are treated as blackbodies, exchanging radiation with an imaginary blackbody surface at temperature \bar{t}_r . The effectiveness of a radiating source on human occupants is governed by the absorptance α of the skin and clothing surface for the color temperature (in °R) of that radiating source. The relationship between α and temperature is illustrated in [Figure 1](#). Values for α are those expected relative to the matte black surface normally found on globe thermometers or radiometers measuring radiant energy. A gas radiator usually operates at 1700°F (2160°R); a quartz lamp, for example, radiates at 4000°F (4460°R) with 240 V; and the sun's radiating temperature is 10,000°F (10,460°R). The use of α in estimating the ERF and t_o caused by sources radiating at temperatures above 1700°F (2160°R) is discussed in the section on Testing Instruments for Radiant Heating.

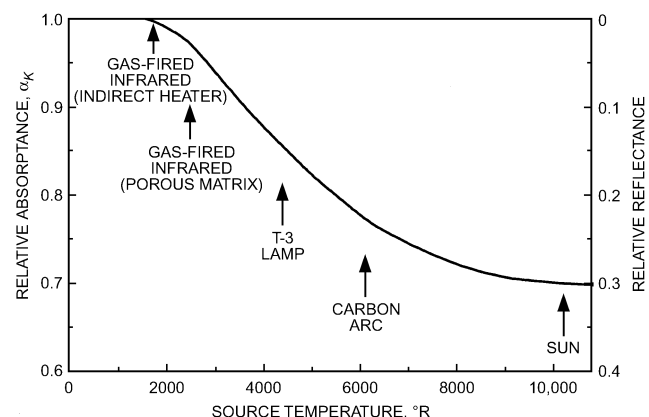


Fig. 1 Relative Absorptance and Reflectance of Skin and Typical Clothing Surfaces at Various Color Temperatures

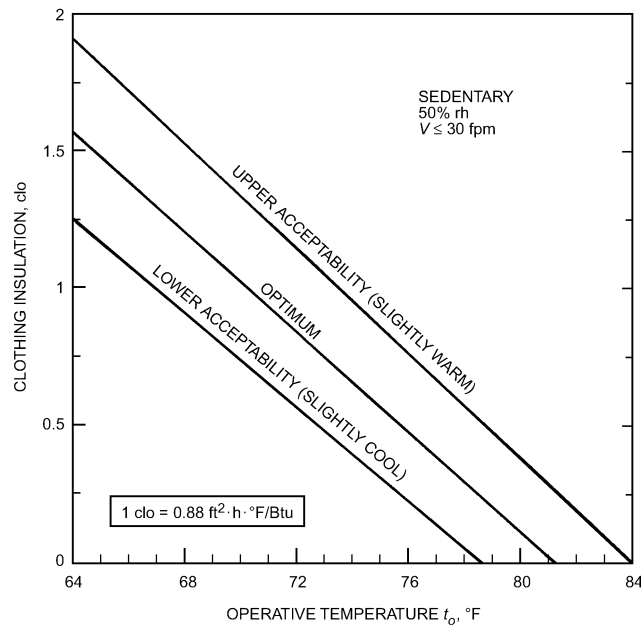


Fig. 2 Range of Thermal Acceptability for Sedentary People with Various Clothing Insulations and Operative Temperatures

DESIGN CRITERIA FOR ACCEPTABLE RADIANT HEATING

Perceptions of comfort, temperature, and thermal acceptability are related to activity, body heat transfer from the skin to the environment, and the resulting physiological adjustments and body temperature. The split between convective and radiant heat transfer from or to a body is also a matter of subjective human comfort. The optimum split is about 60% by thermal radiation and 40% by thermal convection. Heat transfer is affected by ambient air temperature, thermal radiation, air movement, humidity, and clothing worn. Thermal sensation is described as feelings of hot, warm, slightly warm, neutral, slightly cool, cool, and cold. An acceptable environment is defined as one in which at least 80% of the occupants perceive a thermal sensation between “slightly cool” and “slightly warm.” Comfort is associated with a neutral thermal sensation during which the human body regulates its internal temperature with minimal physiological effort for the activity concerned. In contrast, warm discomfort is primarily related to the physiological strain necessary to maintain the body’s thermal equilibrium rather than to the temperature sensation experienced. For a full discussion of the interrelation of physical, psychological, and physiological factors, refer to [Chapter 8 of the ASHRAE Handbook—Fundamentals](#).

ANSI/ASHRAE Standard 55-1992 shows a linear relationship between clothing insulation worn and the operative temperature t_o for comfort (Figure 2). Figure 3 shows the effect of both activity and clothing on the t_o for comfort. Figure 4 shows the slight effect humidity has on the comfort of a sedentary person wearing average clothing. Figures 2, 3, and 4 are adapted from ANSI/ASHRAE Standard 55, Thermal Environmental Conditions for Human Occupancy.

A comfortable t_o at 50% rh is perceived as slightly warmer as humidity increases, or as slightly cooler as humidity decreases. Changes in humidity have a much greater effect on warm and hot discomfort. In contrast, cold discomfort is only slightly affected by humidity and is very closely related to a cold thermal sensation.

Determining the specifications for a radiant heating installation designed for human occupancy and acceptability involves the following steps:

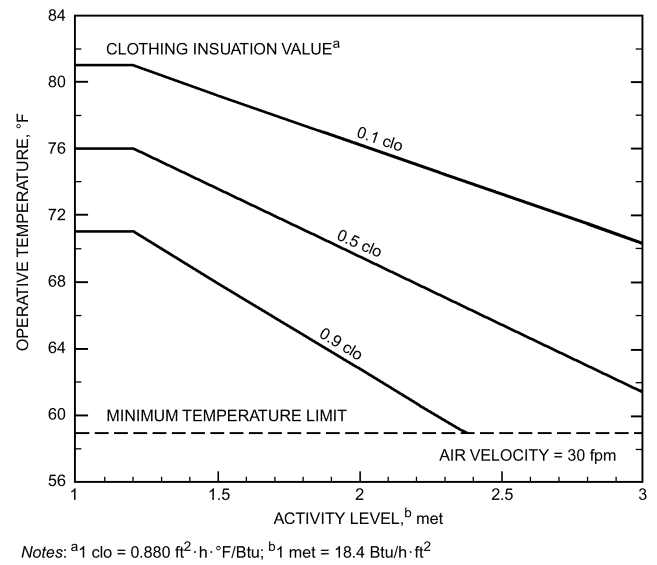


Fig. 3 Optimum Operative Temperatures for Active People in Low-Air-Movement Environments

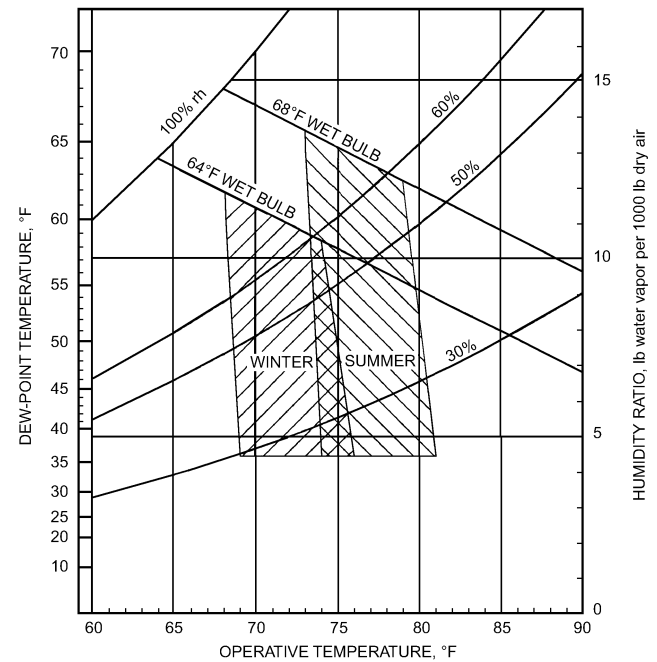


Fig. 4 ASHRAE Comfort Chart for Sedentary Occupants

1. Define the probable activity (metabolism) level of, and clothing worn by, the occupant and the relative air velocity. The following are two examples:

Case 1: Sedentary (1.1 met)

Clothing insulation = 0.6 clo; $V = 30$ fpm

Case 2: Light work (walking) (2 met)

Clothing insulation = 0.9 clo; $V = 100$ fpm

2. From Figure 2 (sedentary) or 3 (active), determine the optimum t_o for comfort and acceptability:

Case 1: $t_o = 75^\circ\text{F}$; Case 2: $t_o = 62.5^\circ\text{F}$

3. For ambient air temperature t_a , calculate the mean radiant temperature \bar{t}_r and/or ERF necessary for comfort and thermal acceptability.

Case 1: For $t_a = 60^\circ\text{F}$ and 50% rh and assuming $\bar{t}_r = 86^\circ\text{F}$,
Solve for h_r from Equation (11):

$$h_r = 4 \times 0.1712 \times 10^{-8} \times 0.71 [(86 + 60)/2 + 460]^3 = 0.736$$

Solve for h_c from Equation (12c):

$$h_c = 0.55$$

Then,

$$h = h_r + h_c = 0.736 + 0.55 = 1.286 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F})$$

$$\frac{h_r}{h} = \frac{0.736}{1.286} = 0.57$$

From Equation (6), for comfort,

$$\text{ERF} = 1.286(75 - 60) = 19.3 \text{ Btu}/\text{h}\cdot\text{ft}^2$$

From Equation (9),

$$\bar{t}_r = 60 + (1.286/0.736)(75 - 60) = 86.2^\circ\text{F}$$

Case 2: For $t_a = 50^\circ\text{F}$ and assuming $\bar{t}_r = 81^\circ\text{F}$,

$$h_r = 4 \times 0.1712 \times 10^{-8} \times 0.71 [(81 + 50)/2 + 460]^3 = 0.706$$

$$h_c = 0.092(100)^{0.53} = 1.06$$

$$h = 0.706 + 1.06 = 1.766 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F})$$

$$\frac{h_r}{h} = \frac{0.706}{1.766} = 0.40$$

$$\text{ERF} = 1.766(62.5 - 50) = 22.1 \text{ Btu}/\text{h}\cdot\text{ft}^2$$

From Equation (7),

$$\bar{t}_r = 50 + 22.1/0.706 = 81.3^\circ\text{F}$$

The t_o for comfort, predicted by [Figure 2](#), is on the cool side when the humidity is low; for high humidities, the predicted t_o for comfort is warm. This effect on comfort for sedentary occupants can be seen in [Figure 4](#). For example, for high humidity at $t_{dp} = 59^\circ\text{F}$, the t_o for comfort is

Case 1: $t_o = 73.5^\circ\text{F}$, compared to 75°F at 50% rh

When thermal acceptability is the primary consideration in an installation, humidity can sometimes be ignored in preliminary design specifications. However, for conditions where radiant heating and the work level cause sweating and high heat stress, humidity is a major consideration and a hybrid HVAC system should be used.

Equations (3) to (12) can also be used to determine the ambient air temperature t_a required when the mean radiant temperature MRT is maintained by a specified radiant system.

When calculating heat loss, t_a must be determined. For a radiant system that is to maintain a MRT of \bar{t}_r , the operative temperature t_o can be determined from [Figure 4](#). Then, t_a can be calculated by recalling that t_o is approximately equal to the average of t_a and t_r .

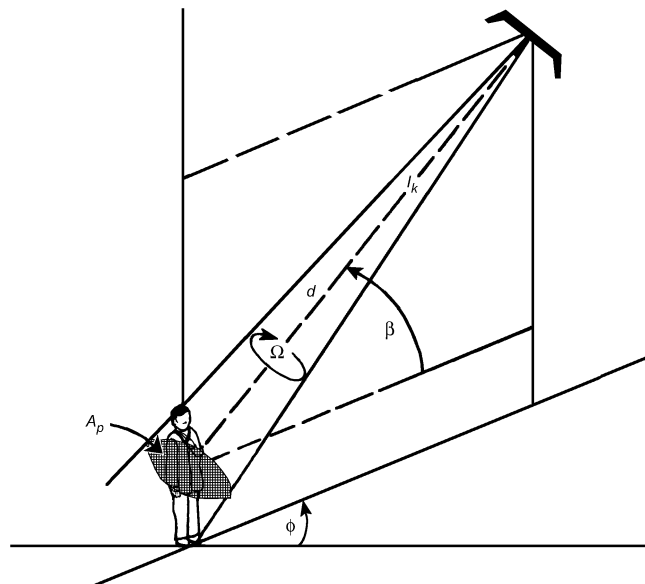


Fig. 5 Geometry and Symbols for Describing Beam Heaters

For example, for a t_o of 73°F and a radiant system designed to maintain an MRT of 78°F , the t_a would be 68°F .

When the surface temperature of outside walls, particularly those with large areas of glass, deviates too much from room air temperature and from the temperature of other surfaces, simplified calculations for load and operative temperature may lead to errors in sizing and locating the panels. In such cases, more detailed radiant exchange calculations may be required, with separate estimation of heat exchange between the panels and each surface. A large window area may lead to significantly lower MRTs than expected. For example, Athienitis and Dale (1987) reported an MRT 5.4°F lower than room air temperature for a room with a glass area equivalent to 22% of its floor area.

DESIGN FOR BEAM RADIANT HEATING

Spot beam radiant heat can improve comfort at a specific location in a large, poorly heated work area. The design problem is specifying the type, capacity, and orientation of the beam heater.

Using the same reasoning as in Equations (1) to (10), the effective radiant flux ΔERF added to an unheated work space with an operative temperature t_{uo} to result in a t_o for comfort (as given by [Figure 2](#) or [3](#)) is

$$\Delta\text{ERF} = h(t_o - t_{uo}) \quad (13)$$

or

$$t_o = t_{uo} + \Delta\text{ERF}/h \quad (14)$$

The heat transfer coefficient h for the occupant in Equation (13) is given by Equations (11) and (12), with $h = h_r + h_c$.

By definition, ERF is the energy absorbed per unit of total body surface A_D (DuBois area) and *not* the total effective radiating area A_{eff} of the body.

Geometry of Beam Heating

[Figure 5](#) illustrates the following parameters that must be considered in specifying a beam radiant heater designed to produce the ERF, or mean radiant temperature \bar{t}_r , necessary for comfort at an occupant's workstation:

- Ω = solid angle of heater beam, steradians (sr)
- I_k = irradiance from beam heater, Btu/h·sr

- K = subscript for absolute temperature of beam heater, °R
- β = elevation angle of heater, degrees (at 0°, beam is horizontal)
- ϕ = azimuth angle of heater, degrees (at 0°, beam is facing subject)
- d = distance from beam heater to center of occupant, ft
- A_p = projected area of occupant on a plane normal to direction of heater beam (ϕ , β), ft²
- α_K = absorptance of skin-clothing surface at emitter temperature (see [Figure 1](#))

ERF may also be measured as the heat absorbed at the occupant's clothing and skin surface from a beam heater at absolute temperature:

$$\text{ERF} = \frac{\alpha_K I_K A_p}{d^2 A_D} \quad (15)$$

where ERF is in Btu/hft² and (A_p/d^2) is the solid angle subtended by the projected area of the occupant from the radiating beam heater I_K , which is treated here as a point source. A_D is the DuBois area:

$$A_D = 0.0621 W^{0.425} H^{0.725}$$

where

- W = occupant weight, lb
- H = occupant height, ft

For additional information on radiant flux distribution patterns and sample calculations of radiation intensity I_K and ERF, refer to [Chapter 15 of the ASHRAE Handbook—HVAC Systems and Equipment](#).

Floor Reradiation

In most low-, medium-, and high-intensity radiant heater installations, local floor areas are strongly irradiated. The floor absorbs most of this energy and warms to an equilibrium temperature t_f , which is higher than the ambient air temperature t_a and the unheated room enclosure surfaces. Part of the energy directly absorbed by the floor is transmitted by conduction to the colder layers beneath (or, for slabs-on-grade, to the ground), part is transferred by natural convection to room air, and the rest is reradiated. The warmer floor will raise ERF or \bar{t}_r over that caused by the heater alone.

For a person standing on a large, flat floor that has a temperature raised by direct radiation t_f , the linearized \bar{t}_r due to the floor and unheated walls is

$$\bar{t}_{rf} = F_{p-f} t_f + (1 - F_{p-f}) t_a \quad (16)$$

where the unheated walls, ceiling, and ambient air are assumed to be at t_a , and F_{p-f} is the angle factor governing radiation exchange between the heated floor and the person.

The ERF_f from the floor affecting the occupant, which is due to the $(t_f - t_a)$ difference, is

$$\begin{aligned} \text{ERF}_f &= h_r (\bar{t}_{rf} - t_a) \\ &= h_r F_{p-f} (t_f - t_a) \end{aligned} \quad (17)$$

where h_r is the linear radiative heat transfer coefficient for a person as given by Equation (11). For a standing or sitting subject when the walls are farther than 16 ft away, F_{p-f} is 0.44 (Fanger 1973). For an average-sized 16 by 16 ft room, a value of 0.35 for F_{p-f} is suggested. For detailed information on floor reradiation, see [Chapter 15 in the ASHRAE Handbook—HVAC Systems and Equipment](#).

In summary, when radiant heaters warm occupants in a selected area of a poorly heated space, the radiation heat necessary for comfort consists of two additive components: (1) ERF directly caused by the heater and (2) reradiation ERF_f from the floor. The

effectiveness of floor reradiation can be improved by choosing flooring with a low specific conductivity. Flooring with high thermal inertia may be desirable during radiant transients, which may occur as heaters are cycled by a thermostat set to the desired operative temperature t_o .

Asymmetric Radiant Fields

In the past, comfort heating has required flux distribution in occupied areas to be uniform, which is not possible with beam radiant heaters. Asymmetric radiation fields, such as those experienced when lying in the sun on a cool day or when standing in front of a warm fire, can be pleasant. Therefore, a limited amount of asymmetry, which is allowable for comfort heating, is referred to as "reasonable uniform radiation distribution" and is used as a design requirement.

To develop criteria for judging the degree of asymmetry allowable for comfort heating, Fanger et al. (1980) proposed defining radiant temperature asymmetry as the difference in the plane radiant temperature between two opposing surfaces. Plane radiant temperature is the equivalent t_{r1} caused by radiation on one side of the subject, compared with the equivalent t_{r2} caused by radiation on the opposite side. Gagge et al. (1967) conducted a study of subjects (eight clothed and eight unclothed) seated in a chair and heated by two lamps. Unclothed subjects found a $(\bar{t}_r - t_a)$ asymmetry as high as 20°F to be comfortable, but clothed subjects were comfortable with an asymmetry as high as 31°F.

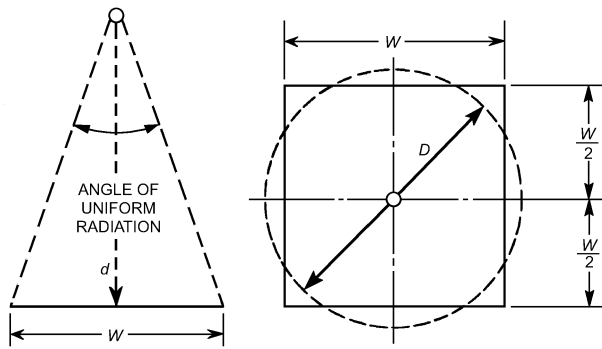
For an unclothed subject lying on an insulated bed under a horizontal bank of lamps, neutral temperature sensation occurred for a t_o of 72°F, which corresponds to a $(t_o - t_a)$ asymmetry of 20°F or a $(\bar{t}_r - t_a)$ asymmetry of 27°F, both averaged for eight subjects (Stevens et al. 1969). In studies of heated ceilings, 80% of eight male and eight female clothed subjects voted conditions as comfortable and acceptable for asymmetries as high as 20°F. The study compared the floor and heated ceilings. Asymmetry in the MRTs for direct radiation from three lamps and for floor reradiation is about 1°F, which is negligible.

In general, the human body has a great ability to sum sensations from many hot and cold sources. For example, a study of Australian aborigines sleeping unclothed next to open fires in the desert at night found a t_a of 43°F (Scholander 1958). The \bar{t}_r caused directly by three fires was 171°F, and the sky \bar{t}_r was 30°F; the resulting t_o was 82°F, which is acceptable for human comfort.

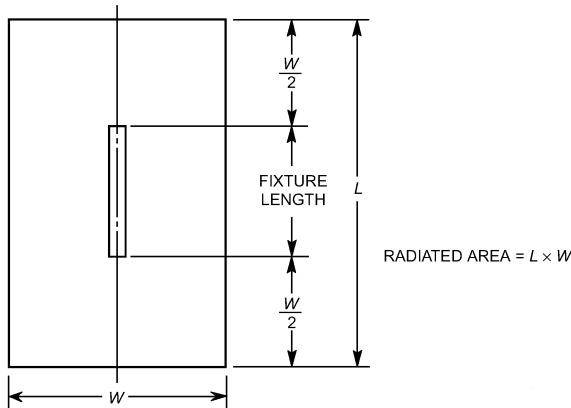
According to the limited field and laboratory data available, an allowable design radiant asymmetry of $22 \pm 5^\circ\text{F}$ should cause little discomfort over the comfortable t_o range used by ASHRAE *Standard 55* and in [Figures 2](#) and [3](#). Increased clothing insulation allows increases in the acceptable asymmetry, but increased air movement reduces it. Increased activity also reduces human sensitivity to changing \bar{t}_r or t_o and, consequently, increases the allowable asymmetry. The design engineer should use caution with an asymmetry greater than 27°F, as measured by a direct beam radiometer or estimated by calculation.

RADIATION PATTERNS

[Figure 6](#) indicates basic radiation patterns commonly used in design for radiation from point or line sources (Boyd 1962). A point source radiates over an area proportional to the square of the distance from the source. The area for a (short) line source also varies substantially as the square of the distance, with about the same area as the circle actually radiated at that distance. For line sources, the pattern width is determined by the reflector shape and position of the element within the reflector. The rectangular area used for installation purposes as the pattern of radiation from a line source assumes a length equal to the width plus the fixture length. This assumed length is satisfactory for design, but is often two or three times the pattern width.



A. PATTERN OF RADIATION FROM POINT SOURCE



B. PATTERN OF RADIATION FROM LINE SOURCE

Note: The projected area W^2 normal to a beam that is Ω steradians wide at distance d is Ωd^2 . The floor area irradiated by a beam heater at an angle elevation β is $W^2/\sin \beta$. Fixture length L increases the area irradiated by $(1 + L/W)$.

Fig. 6 Basic Radiation Patterns for System Design
(Boyd 1962)

Electric infrared fixtures are often identified by their beam pattern (Rapp and Gagge 1967), which is the radiation distribution normal to the line source element. The beam of a high-intensity infrared fixture may be defined as that area in which the intensity is at least 80% of the maximum intensity encountered anywhere within the beam. This intensity is measured in the plane in which maximum control of energy distribution is exercised.

The beam size is usually designated in angular degrees and may be symmetrical or asymmetrical in shape. For adaptation to their design specifications, some manufacturers indicate beam characteristics based on 50% maximum intensity.

The control used for an electric system affects the desirable maximum end-to-end fixture spacing. Actual pattern length is about three times the design pattern length, so control in three equal stages is achieved by placing every third fixture on the same circuit. If all fixtures are controlled by input controllers or variable voltage to electric units, end-to-end fixture spacing can be nearly three times the design pattern length. Side-to-side minimum spacing is determined by the fixture's distribution pattern and is not influenced by the control method.

Low-intensity equipment typically consists of a steel tube hung near the ceiling and parallel to the outside wall. Circulation of combustion products inside the tube elevates the tube temperature, and radiant energy is emitted. The tube is normally provided with a reflector to direct the radiant energy down into the space to be conditioned.

Radiant ceiling panels for heating only are installed with spacing between them to enhance downward convection, and are usually the primary heat source for the space. Usually the source is a surface. Details of radiant and convective heat transfer for panel heating and

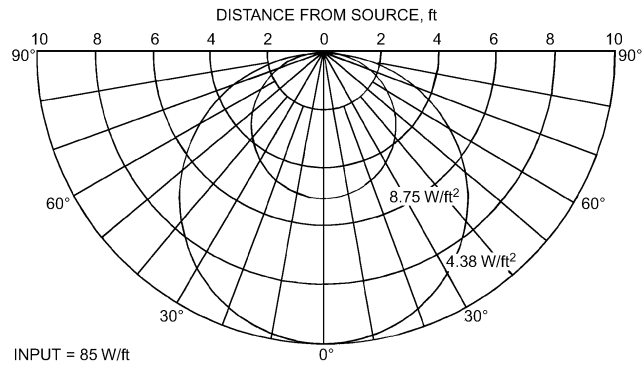


Fig. 7 Lines of Constant Radiant Flux for a Line Source

cooling are given in [Chapter 6 of the ASHRAE Handbook—HVAC Systems and Equipment](#). Flux density is inversely proportional to distance from the source if the source is a recessed line, and is inversely proportional to the square of the distance from the source if the source is a point. In practice, actual dependence will be between these limits.

The rate of radiation exchange between a panel and a particular object depends on their temperatures, surface emittances, and geometrical orientation (i.e., shape factor). It also depends on the temperatures and configurations of all the other objects and walls within the space.

The heat flux for an ideal line source is shown in [Figure 7](#). All objects (except the radiant source) are at the same temperature, and the radiant source is suspended symmetrically in the room.

DESIGN FOR TOTAL SPACE HEATING

Radiant heating differs from conventional heating by a moderately elevated ERF, \bar{t}_r , or t_o over ambient temperature t_a . Standard methods of design are normally used, although informal studies indicate that radiant heating results in a lower heating load than convection heating (Zmeureanu et al. 1987). Buckley and Seel (1987) demonstrated that a combination of elevated floor temperature, higher mean radiant temperature, and reduced ambient temperature results in lower thermostat settings, reduced temperature differential across the building envelope, and thus lower heat loss and heating load for the structure. In addition, peak load may be decreased because of heat (cool) stored in the structure when embedded panels in the floor, walls, or ceiling are used (Kilkis 1992).

Most gas radiation systems for full-building heating concentrate the bulk of capacity at mounting heights of 10 to 16 ft at the perimeter, directed at the floor near the walls. Units can be mounted considerably higher. Successful application depends on supplying the proper amount of heat in the occupied area. Heaters should be located to take maximum advantage of the radiation pattern produced. Exceptions to perimeter placement include walls with high transmission losses and extreme height, as well as large roof areas where roof heat losses exceed perimeter and other heat losses.

Electric infrared systems installed indoors for complete building heating use layouts that uniformly distribute the radiation throughout the area used by people, as well as layouts that emphasize perimeter placement, such as in ice hockey rinks. Some electric radiant heaters emit significant visible radiation and provide both heating and illumination.

The orientation of equipment and people is less important for general area heating (large areas within larger areas) than for spot heating. With reasonably uniform radiation distribution in work or living areas, the exact orientation of the units is not important. Higher intensities of radiation may be desirable near walls with outside exposure. Radiation shields (reflective to infrared) fastened a

few inches from the wall to allow free air circulation between the wall and shield are effective for frequently occupied work locations close to outside walls.

In full-building heating, units should be placed where their radiant and convective output best compensates for the structure's heat loss. The objective of a complete heating system is to provide a warm floor with low conductance to the heat sink beneath the floor. This thermal storage may permit cycling of units with standard controls.

TESTING INSTRUMENTS FOR RADIANT HEATING

In designing a radiant heating system, calculation of radiant heat exchange may involve some untested assumptions. During field installation, the designer must test and adjust the equipment to ensure that it provides acceptable comfort conditions. The black globe thermometer and directional radiometer can be used to evaluate the installation.

Black Globe Thermometer

The classic (Bedford) globe thermometer is a thin-walled, matte black, hollow sphere with a thermocouple, thermistor, or thermometer placed at the center. It can directly measure \bar{t}_r , ERF, and t_o . When a black globe is in thermal equilibrium with its environment, the gain in radiant heat from various sources is balanced by the convective loss to ambient air. Thus, in terms of the globe's linear radiative and convective heat transfer coefficients, h_{rg} and h_{cg} , respectively, the heat balance at equilibrium is

$$h_{rg}(\bar{t}_{rg} - t_g) = h_{cg}(t_g - t_a) \quad (18)$$

where \bar{t}_{rg} is the mean radiant temperature measured by the globe and t_g is the temperature in the globe.

In general, the \bar{t}_{rg} of Equation (18) equals the \bar{t}_r affecting a person when the globe is placed at the center of the occupied space and when the radiant sources are distant from the globe.

The effective radiant flux measured by a black globe is

$$\text{ERF}_g = h_{rg}(\bar{t}_{rg} - t_a) \quad (19)$$

which is analogous to Equation (5) for occupants. From Equations (18) and (19), it follows that

$$\text{ERF}_g = (h_{rg} + h_{cg})(t_g - t_a) \quad (20)$$

If the ERF_g of Equation (20) is modified by the skin-clothing absorptance α_K and the shape f_{eff} of an occupant relative to the black globe, the corresponding ERF affecting the occupant is

$$\text{ERF (for an occupant)} = f_{eff}\alpha_K\text{ERF}_g \quad (21)$$

where α_K is defined in the section on Geometry of Beam Heating and f_{eff} , which is defined after Equation (11), is approximately 0.71 and equals the ratio h_r/h_{rg} . The t_o affecting an occupant, in terms of t_g and t_a , is given by

$$t_o = Kt_g + (1 - K)t_a \quad (22)$$

where the coefficient K is

$$K = \alpha_K f_{eff} (h_{rg} + h_{cg}) / (h_r + h_c) \quad (23)$$

Ideally, when K is unity, the t_g of the globe would equal the t_o affecting an occupant.

For an average comfortable equilibrium temperature of 77°F and noting that f_{eff} for the globe is unity, Equation (11) yields

Table 1 Value of K for Various Air Velocities and Globe Thermometer Diameters ($\alpha_g = 1$)

| Air Velocity V_a , fpm | Approximate Globe Thermometer Diameter D , in. | | | |
|-----------------------------|--|------|------|------|
| | 2 | 4 | 6 | 8 |
| 50 | 1.35 | 1.15 | 1.05 | 0.99 |
| 100 | 1.43 | 1.18 | 1.06 | 0.99 |
| 200 | 1.49 | 1.21 | 1.07 | 1.00 |
| 400 | 1.54 | 1.23 | 1.08 | 1.00 |
| 800 | 1.59 | 1.26 | 1.09 | 1.00 |

$$h_{rg} = 1.06 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}) \quad (24)$$

and Equation (12) yields

$$h_{cg} = 0.296D^{-0.4}V_a^{0.53} \quad \{100 \text{ fpm} < V_a < 400 \text{ fpm}\} \quad (25)$$

where

D = globe diameter, in.
 V_a = air velocity, fpm

Equation (25) is Bedford's convective heat transfer coefficient for a 6 in. globe's convective loss, modified for D . For any radiating source below 1700°F, the ideal diameter of a sphere that makes $K = 1$ that is independent of air movement is 8 in. (see Table 1). Table 1 shows the value of K for various values of globe diameter D and ambient air movement V . The table shows that the uncorrected temperature of the traditional 6 in. globe would overestimate the true $(t_o - t_a)$ difference by 6% for velocities up to 200 fpm, and the probable error of overestimating t_o by t_g uncorrected would be less than 0.9°F. Globe diameters between 6 and 8 in. are optimum for using the uncorrected t_g measurement for t_o . The exact value for K may be used for smaller globes when estimating t_o from t_g and t_a measurements. The value of \bar{t}_r may be found by substituting Equations (24) and (25) in Equation (18), because \bar{t}_r (occupant) is equal to t_{rg} . The smaller the globe, the greater the variation in K caused by air movement. Globes with D greater than 8 in. will overestimate the importance of radiation gain versus convection loss.

For sources radiating at high temperatures (1300 to 10,000°F), the ratio α_m/α_g may be set near unity by using a pink-colored globe surface, whose absorptance for the sun is 0.7, a value similar to that of human skin and normal clothing (Madsen 1976).

In summary, the black globe thermometer is simple and inexpensive and may be used to determine \bar{t}_r [Equation (18)] and the ERF [Figure 1 and Equations (20) and (21)]. When the radiant heater temperature is less than 1700°F, the uncorrected t_g of a 6 to 8 in. black globe is a good estimate of the t_o affecting the occupants. A pink globe extends its usefulness to sun temperatures (10,000°F). A globe with a low mass and low thermal capacity is more useful because it reaches thermal equilibrium in less time.

Using the heat exchange principles described, many heated and unheated instruments of various shapes have been designed to measure acceptability in terms of t_o , \bar{t}_r , and ERF, as sensed by their own geometric shapes. Madsen (1976) developed an instrument that can determine the predicted mean vote (PMV) from the $(t_g - t_a)$ difference, as well as correct for clothing insulation, air movement, and activity (ISO 1994).

Directional Radiometer

The angle of acceptance (in steradians) in commercial radiometers allows the engineer to point the radiometer directly at a wall, floor, or high-temperature source and read the average temperature of that surface. Directional radiometers are calibrated to measure either the radiant flux accepted by the radiometer or the equivalent blackbody radiation temperature of the emitting surface. Many radiometers collimate (make parallel) input to sense small areas of body,

clothing, wall, or floor surfaces. A directional radiometer allows rapid surveys and analyses of important radiant heating factors such as the temperature of skin, clothing surfaces, and walls and floors, as well as the radiation intensity I_K of heaters on the occupants. One radiometer for direct measurement of the equivalent radiant temperature has an angle of acceptance of 2.8° or 0.098 sr, so that at 1 ft, it measures the average temperature over a projected circle about 3/8 in. in diameter.

APPLICATIONS

When installing radiant heaters in specific applications, consider the following factors:

- Gas and electric high-temperature infrared heaters must not be placed where they could ignite flammable dust or vapors, or decompose vapors into toxic gases.
- Fixtures must be located with recommended clearances to ensure proper heat distribution. Stored materials must be kept far enough from fixtures to avoid hot spots. Manufacturers' recommendations must be followed.
- Unvented gas heaters inside tight, poorly insulated buildings may cause excessive humidity with condensation on cold surfaces. Proper insulation, vapor barriers, and ventilation prevent these problems.
- Combustion heaters in tight buildings may require makeup air to ensure proper venting of combustion gases. Some infrared heaters are equipped with induced-draft fans to relieve this problem.
- Some transparent materials may break because of uneven application of high-intensity infrared. Infrared energy is transmitted without loss from the radiator to the absorbing surfaces. The system must produce the proper temperature distribution at the absorbing surfaces. Problems are rarely encountered with glass 0.25 in. or less in thickness.
- Comfort heating with infrared heaters requires a reasonably uniform heat flux distribution in the occupied area. Although thermal discomfort can be relieved in warm areas with high air velocity, such as on loading docks, the full effectiveness of a radiant-heater installation is reduced by high air velocity.
- Radiant spot heating and zoning in large undivided areas with variable occupancy patterns provides localized heating just where and when people are working, which reduces the heating cost.

By building periodic microstructure into a thin silicon carbide wafer, infrared panel surfaces that are highly directional and coherent over many wavelengths can be obtained (Greffet et al. 2002).

Low-, Medium-, and High-Intensity Infrared Applications

Low-, medium-, and high-intensity infrared equipment is used extensively in industrial, commercial, and military applications. This equipment is particularly effective in large areas with high ceilings, such as in buildings with large air volumes and in areas with high infiltration rates, large access doors, or large ventilation requirements.

Factories. Low-intensity radiant equipment suspended near the ceiling around the perimeter of facilities with high ceilings enhances the comfort of employees because it warms floors and equipment in the work area. For older, uninsulated buildings, the energy cost for low-intensity radiant equipment is less than that of other heating systems. High-intensity infrared for spot heating and low-intensity infrared for zone temperature control effectively heat large unheated facilities.

Warehouses. Low- and high-intensity infrared are used for heating warehouses, which usually have a large volume of air, poor insulation, and high infiltration. Low-intensity infrared equipment is installed near the ceiling around the building's perimeter. High-level mounting near the ceiling leaves floor space available for product storage. Both low- and high-intensity infrared are arranged to

control radiant intensity and provide uniform heating at the working level and frost protection areas, which is essential for perishable goods storage.

Garages. Low-intensity infrared provides comfort for mechanics working near or on the floor. With elevated MRT in the work area, comfort is provided at a lower ambient temperature.

In winter, opening the large overhead doors to admit equipment for service allows a substantial entry of cold outdoor air. On closing the doors, the combination of reradiation from the warm floor and radiant heat warming the occupants (not the air) provides rapid recovery of comfort. Radiant energy rapidly warms the cold vehicles. Radiant floor panel heating systems are also effective.

Low-intensity equipment is suspended near the ceiling around the perimeter, often with greater concentration near overhead doors. High-intensity equipment is also used to provide additional heat near doors.

Aircraft Hangars. Equipment suspended near the roofs of hangars, which have high ceilings and large access doors, provides uniform radiant intensity throughout the working area. A heated floor is particularly effective in restoring comfort after an aircraft has been admitted. As in garages, the combination of reradiation from the warm floor and radiation from the radiant heating system provides rapid regain of comfort. Radiant energy also heats aircraft moved into the work area.

Greenhouses. In greenhouse applications, a uniform flux density must be maintained throughout the facility to provide acceptable growing conditions. In a typical application, low-intensity units are suspended near, and run parallel to, the peak of the greenhouse.

Sometimes the soil is warmed by embedding pipes in the ground. The same system may also contribute to sensible heating of the greenhouse.

Outdoor Applications. Applications include loading docks, racetrack stands, outdoor restaurants, and under marquees. Low-, medium-, and high-intensity infrared are used in these facilities, depending on their layout and requirements.

Other Applications. Radiant heat may be used in a variety of large facilities with high ceilings, including religious compounds, day-care facilities, gymnasiums, swimming pools, enclosed stadiums, and facilities that are open to the outdoors. Radiant energy is also used to control condensation on surfaces such as large glass exposures. Careful sizing and design of direct gas-fired systems may improve thermal comfort in places of assembly by affecting air and wall surface temperatures. A numerical study with computational fluid design (CFD) models may be essential to achieve these benefits and reduce condensation risk (Xiang et al. 2001).

Low-, medium-, and high-intensity infrared are also used for other industrial applications, including process heating for component or paint drying ovens, humidity control for corrosive metal storage, and snow control and freeze-protection for parking or loading areas.

Panel Heating and Cooling

Residences. Embedded pipe systems, electric resistance panels, and forced warm-air panel systems have all been used in residences. The embedded pipe system is most common, using plastic or rubber tubing in the floor slab or copper tubing in older plaster ceilings. These systems are suitable for conventionally constructed residences with normal amounts of glass. Lightweight hydronic metal panel ceiling systems have also been applied to residences, and prefabricated electric panels are advantageous, particularly in rooms that have been added on.

Office Buildings. A panel system is usually applied as a perimeter heating system. Panels are typically piped to provide exposure control with one riser on each exposure and all horizontal piping incorporated in the panel piping. In these applications, the air system provides individual room control. Perimeter radiant panel systems have also been installed with individual zone controls.

However, this type of installation is usually more expensive and, at best, provides minimal energy savings and limited additional occupant comfort. Radiant panels can be used for cooling as well as heating. Cooling installations are generally limited to retrofit or renovation jobs where ceiling space is insufficient for the required duct sizes. In these installations, the central air supply system provides ventilation air, dehumidification, and some sensible cooling. Two- and four-pipe water distribution systems may be used. Hot-water supply temperatures are commonly reset by outside temperature, with additional offset or flow control to compensate for solar load. Panel systems are readily adaptable to accommodate most changes in partitioning. Electric panels in lay-in ceilings have been used for full perimeter heating.

Schools. In all areas except gymnasiums and auditoriums, panels are usually selected for heating only, and may be used with any type of approved ventilation system. The panel system is usually sized to offset the transmission loads plus any reheating of the air. If the school is air conditioned by a central air system and has perimeter heating panels, single-zone piping may be used to control the panel heating output, and the room thermostat modulates the supply air temperature or volume. Heating and cooling panel applications are similar to those in office buildings. Panel heating and cooling for classroom areas has no mechanical equipment noise to interfere with instructional activities.

Hospitals. The principal application of heating and cooling radiant panels has been for hospital patient rooms. Perimeter radiant heating panels are typically applied in other areas of hospitals. Compared to conventional systems, radiant heating and cooling systems are well suited to hospital patient rooms because they (1) provide a draft-free, thermally stable environment, (2) have no mechanical equipment or bacteria and virus collectors, and (3) do not take up space in the room. Individual room control is usually achieved by throttling the water flow through the panel. The supply air system is often 100% outdoor air; minimum air quantities delivered to the room are those required for ventilation and exhaust of the toilet room and soiled linen closet. The piping system is typically a four-pipe design. Water control valves should be installed in corridors so that they can be adjusted or serviced without entering patient rooms. All piping connections above the ceiling should be soldered or welded and thoroughly tested. If cubicle tracks are applied to the ceiling surface, track installation should be coordinated with the radiant ceiling. In panel cooling, surface condensation may need a separate latent system (Isoardi and Brasselet 1995). Security panel ceilings are often used in areas occupied by mentally disturbed patients so that equipment cannot be damaged by a patient or used to inflict injury.

Swimming Pools. A partially clothed person emerging from a pool is very sensitive to the thermal environment. Panel heating systems are well suited to swimming pool areas. Floor panel temperatures must be controlled so they do not cause foot discomfort. Ceiling panels are generally located around the perimeter of the pool, not directly over the water. Panel surface temperatures are higher to compensate for the increased ceiling height and to produce a greater radiant effect on partially clothed bodies. Ceiling panels may also be placed over windows to reduce condensation.

Apartment Buildings. For heating, pipe coils are embedded in the masonry slab. The coils must be carefully positioned so as not to overheat one apartment while maintaining the desired temperature in another. The slow response of embedded pipe coils in buildings with large glass areas may be unsatisfactory. Installations for heating and cooling have been made with pipes embedded in hung plaster ceilings. A separate minimum-volume dehumidified air system provides the necessary dehumidification and ventilation for each apartment. The application of electric resistance elements embedded in floors or behind a skim coat of plaster at the ceiling has increased. Electric panels are easy to install and simplify individual room control.

Industrial Applications. Panel systems are widely used for general space conditioning of industrial buildings in Europe (Petras

2001). For example, the walls and ceilings of an internal combustion engine test cell are cooled with chilled water. Although the ambient air temperature in the space reaches up to 95°F, the occupants work in relative comfort when 55°F water is circulated through the ceiling and wall panels.

Other Buildings. Metal panel ceiling systems can be operated as heating systems at elevated water temperatures and have been used in airport terminals, convention halls, lobbies, and museums, especially those with large glass areas. Cooling may also be applied. Because radiant energy travels through the air without warming it, ceilings can be installed at any height and remain effective. One particularly high ceiling installed for a comfort application is 50 ft above the floor, with a panel surface temperature of approximately 285°F for heating. The ceiling panels offset the heat loss from a single-glazed, all-glass wall.

The high lighting levels in television studios make them well suited to panels that are installed for cooling only and are placed above lighting to absorb the radiation and convection heat from the lights and normal heat gains from the space. The panel ceiling also improves the acoustical properties of the studio.

Metal panel ceiling systems are also installed in minimum- and medium-security jail cells and in facilities where disturbed occupants are housed. The ceiling is strengthened by increasing the gage of the ceiling panels, and security clips are installed so that the ceiling panels cannot be removed. Part of the perforated metal ceiling can be used for air distribution.

District Energy Systems. Panel heating and cooling increases the thermal efficiency and use rate of low-enthalpy energy resources like waste heat. Buildings equipped with panels may be cascaded with buildings with high-enthalpy HVAC systems, which increases the use rate of district energy systems (Kilkis 2002).

New Techniques. The introduction of thermoplastic and rubber tubing and new design techniques have improved radiant panel heating and cooling equipment. The systems are energy-efficient and use low water temperatures available from solar collectors and heat pumps (Kilkis 1993). Metal radiant panels can be integrated into ceiling design to provide a narrow band of radiant heating around the building perimeter. These new radiant systems are more attractive, provide more comfortable conditions, operate more efficiently, and have a longer life than some baseboard or overhead air systems.

SYMBOLS

| | |
|------------------|---|
| A_D | = total DuBois surface area of person, ft ² |
| A_{eff} | = effective radiating area of person, ft ² |
| A_p | = projected area of occupant normal to the beam, ft ² |
| clo | = unit of clothing insulation equal to 0.880 ft ² ·°F·h/Btu |
| D | = diameter of globe thermometer, in. |
| d | = distance of beam heater from occupant, ft |
| ERF | = effective radiant flux (person), Btu/h·ft ² |
| ERF _f | = radiant flux caused by heated floor on occupant, Btu/h·ft ² |
| ERF _g | = effective radiant flux (globe), Btu/h·ft ² |
| F_{p-f} | = angle factor between occupant and heated floor |
| f_{eff} | = ratio of radiating surface (person) to its total area (DuBois) |
| H_m | = net metabolic heat loss from body surface, Btu/h·ft ² |
| h | = combined heat transfer coefficient (person), Btu/h·ft ² ·°F |
| h_c | = convective heat transfer coefficient for person, Btu/h·ft ² ·°F |
| h_{cg} | = convective heat transfer coefficient for globe, Btu/h·ft ² ·°F |
| h_r | = linear radiative heat transfer coefficient (person), Btu/h·ft ² ·°F |
| h_{rg} | = linear radiative heat transfer coefficient for globe, Btu/h·ft ² ·°F |
| I_K | = irradiance from beam heater, Btu/h·sr |
| K | = coefficient that relates t_a and t_g to t_o [Equation (22)] |
| K | = subscript indicating absolute irradiating temperature of beam heater, °R |
| L | = fixture length, ft |
| met | = unit of metabolic energy equal to 18.4 Btu/h·ft ² |
| t_a | = ambient air temperature near occupant, °F |
| t_f | = floor surface temperature, °F |
| t_g | = globe temperature, °F |
| t_o | = operative temperature, °F |

\bar{t}_r = mean radiant temperature affecting occupant, °F
 t_{rf} = linearized t_r caused by floor and unheated walls on occupant, °F
 t_{sf} = exposed surface temperature of occupant, °F
 t_{wo} = operative temperature of unheated workspace, °F
 V = relative velocity between occupant and air, fpm
 V_a = air velocity, fpm
 W = width of a square equivalent to the projected area of a beam of angle Ω steradians at a distance d , ft
 α = relative absorptance of skin-clothing surface to that of matte black surface
 α_g = absorptance of globe
 α_K = absorptance of skin-clothing surface at emitter temperature
 α_m = absorptance of skin-clothing surface at emitter temperatures above 1700°F
 β = elevation angle of beam heater, degrees
 σ = Stefan-Boltzmann constant = 0.1712×10^{-8} Btu/h · ft² · °R⁴
 ϕ = azimuth angle of heater, degrees
 Ω = radiant beam width, sr

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