Impact of the Position of the Radiators on Energy Consumption and Thermal Comfort in a Mixed Radiant and Convective Heating System

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ABSTRACT

[1]This paper studies the heating load and thermal comfort distribution in a typical office with a mixed radiant and convective heating system for two different locations of radiant heat sources. Accurately estimating the heating load in a mixed heating space requires careful consideration of the energy balance on each room surface and the comfort level in the space. A pure radiant heating system heats the room surfaces first; then the warm surfaces heat room air. The higher surface temperatures will increase the heat loss from the enclosure to the ambient environment for a fixed air temperature. On the other hand, a radiant heating system creates a higher mean radiant temperature in the space. By keeping the same operative temperature as used with a convective heating system, this system can have a lower room air temperature, which usually reduces the energy to heat infiltrating air or ventilation air, and reduces the convective heat transfer between the room air and enclosure surfaces during the heating season. The reduced room air temperature has the potential to reduce the heat loss from the enclosure to ambient environment. This paper compares the heating load and comfort level as measured by uniformity of operative temperature for two different layouts of radiators in the same geometric space. It is found that when radiators are close to the window, it may increase heating consumption up to 3.6% compared to 100% convective heating in an inadquately ventilated space. In a properly ventilated space, radiant heating can save up to 7.7% of the heating consumption depending on location of the radiator(s) and the outside air supply rate for the cases simulated. Comfort analysis shows that locating the radiator near the window can improve the comfort level in a space.

INTRODUCTION

Radiant Heating

Radiant heating has the reputation of increasing the comfort level in a space and lowering energy bills. A radiant heating system uses one or more temperature controlled indoor surfaces on the floor, walls or ceiling to heat the enclosure surfaces and objects first. The warm surfaces then heat the inside air by convection. Because warm enclosure surfaces radiate more energy to a human body than cold surfaces, people may feel comfortable even if the air temperature is several degrees lower than with a forced air heating system. A temperature controlled surface is called a radiant panel where the temperature is maintained by circulating water, air or electric current. According to the ASHRAE Handbook (2001), the panel surface temperature is normally lower than 300°F. The radiant heating system may be combined with a central forced air system to supply the heating or cooling required by the space. Such systems are called a mixed radiant and convective heating system or a hybrid HVAC system (ASHRAE Handbook 2004).

Floor heating is one of the oldest and most popular radiant heating systems. Stove and flue gas ducts underneath the building floor constitute the ancient heating systems which were used in East Asian countries thousands of years ago. The advantages of floor heating are quiet operation and superior comfort. Several investigations (Dale 1993, Olesen 1994, Gibbs 1994) have evaluated the energy consumption and

[1] This paper was published at *ASHRAE Transactions*, V 113, part 1, 2007

comfort level for this type of heating. Radiant floor heating proponents claim that fuel savings of 15% to 20% (DOE website 2005) over forced air systems are possible.

For office buildings, the most practical application of radiant heating is wall or ceiling mounted heating panels combined with a forced air system. The Hybrid HVAC system provides more flexible control over the space operative temperature, air distribution velocity and humidity level.

Significant research has been done regarding the mean radiant temperature distribution and comfort level of radiant systems. Steinman et al. (1989) proposed a calculation method for mean radiant temperature and noted that the temperature difference between room air and unheated internal surfaces may not be small with a radiant heating system when an enclosure has a large window or a high percentage of exterior walls. Tassou et al (2000) compared radiant and forced air heating systems in two churches. They found that properly located heaters can create a more uniform temperature distribution than an air heating system in a large space. Chapman et al. (1997) visulized mean radiant temperate (MRT) distribution in a bedroom and a kitchen to analyze the thermal comfort conditions, where the heating panels are mounted on the ceiling.

Several studies have examined energy consumption with radiant heating systems and some compare the consumption with radiant systems to that with air heating systems. DeGreef and Chapman (1998) used an improved methodology to analyze the energy consumption of a 48 square foot bathroom with a radiator mounted in the center of the ceiling. Degreef and Chapman (1998) indicated the energy required by a 100% radiant heating system is 25% less than that required by a 100% convective heating system to achieve the same average MRT in the case analyzed. By keeping the operative temperature constant, Chapman et al. (2000) found that the energy consumption of 100% radiant heating is slightly less than (6%) that of a 100% convective heating system in a 3 meter square enclosure without a window. Hanibuchi and Hokoi (2000) compared a floor heating system with a convective heating system; they pointed out that when convective heat exchange is dominant, heat loss through poorly insulated windows is larger than when radiant heat exchange is dominant. Their conclusion was based on keeping the operative temperature at the central point of the tested room constant in the case of floor heating. Most of these studies tend to conclude that radiant heating can save energy compared with a forced air heating system when keeping operative temperature or MRT constant.

One of the important factors untouched by these studies is that the position of radiators relative to windows and the outside air supply rate have an impact on energy consumption and comfort distribution in a space. The location of a radiator can greatly affect the enclosure surface temperature nearby. If a radiator is near windows, it increases the inside surface temperature of the window and counteracts the down draft to make the people near the window feel comfortable. However, this arrangement may increase the heating load. If a radiator is away from windows, the surface temperature of the windows is lower, but the comfort level near window may not be as good as the former layout.

Thermal comfort

The primary objective of HVAC design is to satisfy the thermal comfort requirement of a conditioned space. Any energy management measures must consider this goal first. ASHRAE Standard 55-2004 defines thermal comfort as the “condition of mind which expresses satisfaction with the thermal environment”. Six primary factors affect the thermal comfort of an occupied space: metabolic rate, clothing insulation, air temperature, radiant temperature, air speed and humidity. ASHRAE Standard 55 (2004) specifies the comfort zones appropriate for spaces where 80% of sedentary or slightly active persons find the environment thermally acceptable when their clothing provides between 0.5 clo and 1.0 clo of thermal insulation. Of the six factors noted above, air temperature, radiant temperature, air speed and humidity can be controlled by the HVAC system. Therefore, the comfort zone is expressed as ranges of operative temperature and humidity for environments where the air speeds are not greater than 40 ft/min (0.20 m/s).

From the viewpoint of heat transfer, radiation, convection and evaporation control heat loss from the human body. These three factors are determined by the mean radiant temperature, air temperature, humidity and air speed of a space. Humidity is normally controlled by the HVAC system for the entire area served by one air handler. Air velocity is maintained by the ventilation and air supply system in an individual room at the level needed to provide relatively uniform temperatures and avoid drafts. Air temperature and velocity determine the convection heat transfer rate between the human body and indoor air with heat loss proportional to the temperature difference. Mean radiant temperature (MRT) determines the radiation heat exchange between the human body and surrounding surfaces. In a typical room, the air temperature and MRT are the only two variables the design engineer may control (Palmer and Chapman 2000).

MRT is defined as "the uniform temperature of an imaginary enclosure in which radiant heat transfer from the human body equals the radiant heat transfer in the actual nonuniform enclosure" (ASHRAE Handbook, 2001). MRT can be calculated from the surface temperatures and the corresponding angle factors from the occupant and the surrounding surfaces by the following equation (ASHRAE Handbook, 2001):

$$T_r = \left[\sum_i F_{p-i} T_i^4 \right]^{\frac{1}{4}} \tag{1}$$

Tr is mean radiant temperature; F_{p-i} is the angle factor between person and surface; and Ti is surface temperature. The mean radiant temperature can also be determined by the discrete ordinate method (Degreef and Chapman 1997) using the following equation:

$$T_r = \left[\frac{\sum_j I^j A_P^j w^j}{A_{eff}\sigma} \right]^{\frac{1}{4}} \tag{2}$$

Ij is the intensity coming from a discrete direction; wj is the quadrature weighting factor for the direction; A_p^j is the projected area in the given direction; and Aeff is the effective area of a person. When the temperature differences among surfaces in an enclosure are small, there is no significant difference in the results of these two equations.

Both MRT and room air temperature have a strong influence on thermal comfort although they are not the only conditions influencing human thermal comfort. Operative temperature, which is a term combining air temperature and mean radiant temperature, was suggested by Fanger (1967) as a measure of local thermal comfort. Operative temperature is defined as "the uniform temperature of an imaginary black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual nonuniform space". According to ASHRAE Standard 55-2004, operative temperature can be calculated by the following equation:

$$T_{op} = AT_a + (1 - A)T_r \tag{3}$$

The value of A is a function of relative air speed Vr and can be found from Table 1.

Table 1. Value of A in Equation (3)

Air Speed Vr	<40 fpm (<0.2 m/s)	40 to 120 fpm (0.2 to 0.6 m/s)	120 to 200 fpm (0.6 to 1.0 m/s)
A	0.5	0.6	0.7

(Source: ANSI/ASHRAE Standard 55-2004)

When air speed is small (less than 0.2m/s) or the difference between mean radiant and air temperature is small (less than 4°C or 7°F), the operative temperature can be approximated as the mean of average air temperature and MRT.

When a space has a large area of window as shown in Figure 1, the temperature difference between interior walls and the surface of the exterior window is large in winter. Convective heating systems sometimes have difficulty counteracting the discomfort caused by the cold window surface. Radiant heating is efficient in this situation to neutralize this deficiency and minimize radiation losses by the human body. This leads to the question of how the radiator should be located to achieve energy efficiency and improve the thermal comfort in the space.

This paper analyzes the heating load and operative temperature distribution in two cases. In Case 1, the radiators are located close to a large window. In Case 2, the radiator is located in the center of the ceiling. The heating energy consumption for these two cases is analyzed for different radiant and convective heating ratios. The thermal comfort distributions in these two cases are also analyzed by numerical methods.

SIMULATION CASES

Some people may have experienced discomfort when sitting close to a window or near sliding glass doors in winter. To counteract this effect, panel radiators may be installed close to windows or on the ceiling. Two different radiator positions are studied in a typical office geometry. The office has dimensions of 15 feet long, 10 feet wide and 8 feet high. Radiant heating combined with a central forced-air system is assumed as the heating system. The configuration of this office is shown in Figures 1 and 2. The office is assumed to be in a "middle" floor of an office building. To simplify the calculation, we assume the ceiling, floor, back wall and sidewalls to be adiabatic. The entire exterior wall is assumed to be a double glaze window with an R value of 1.64 hr·ft^2·°F/Btu. This resistance value excludes the internal surface convective and radiative resistance, which will be evaluated seperately later.

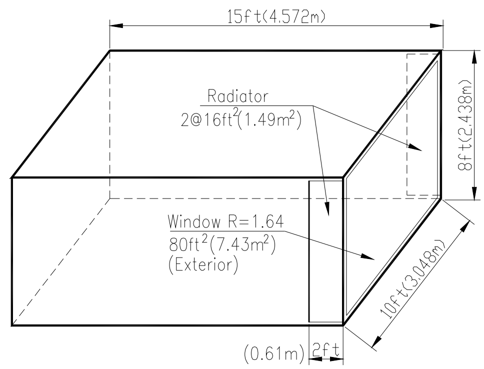

Figure 1. Geometry of an office (Case 1)

In Case 1, two 2x8 foot radiators are positioned as shown next to the window. In Case 2, one 4x8 foot radiator is positioned in the center of the ceiling. The outside temperature is presumed to be 30^{o}F (-1.1^{o}C) and the operative temperature is set to be 73^{o}F (22.8^{o}C). This paper studies the impact of radiator position on heating consumption and thermal comfort for different ratios of radiant and convective heating in these two cases at outside air supply rates of 10 cfm, 20 cfm and 40 cfm. The occupant sensible load is 75W for one person. The lighting and equipment load is 160W.

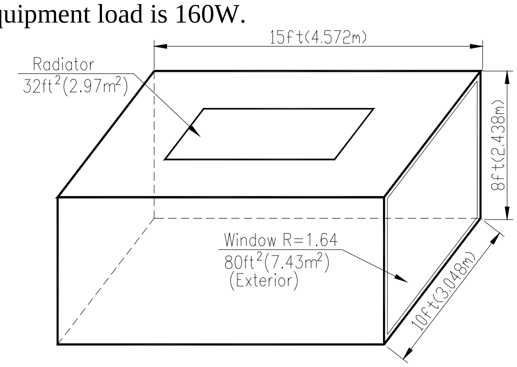

Figure 2. Geometry of an office (Case 2)

HEAT TRANSFER AND ENERGY MODEL

The heat transfer at an internal surface in the enclosure shown in Figures 1 and 2 consists of irradiation from other surfaces, emission to other surfaces, convection between the surface and inside air, and conduction loss to outside. For all adiabatic surfaces, the conduction term vanishes. The window is the only component where heat can be conducted outside. Heat is supplied to the radiator by hot water and can be seen as a generation term, with units of Btu/ft^2.

The heat balance on the occupants includes irradiation from and emission to each surface in the enclosure, convection loss to inside air, and heat generation from the human body. The heat balance can be illustrated as shown in Figure 3. The energy equation for a control surface can be written as

$$\nabla \bullet (k\nabla T_s) + h_c (T_a - T_s) + \dot{q}'' + \dot{q}_r = \rho c_p \frac{\partial T}{\partial t} \tag{4}$$

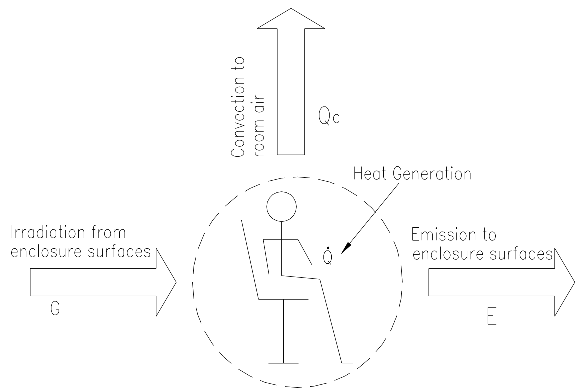

Figure 3. Energy Balance on Occupant

In Equation (4), h_c is the surface convection coefficient which is a temperature dependent variable. Each surface has a different h_c. This will be discussed later. $\dot{q}_r$ can be written as follows:

$$\dot{q}_r = -\sum_i F_{s-i}\varepsilon\sigma\left(T_s^4 - T_i^4\right) \tag{5}$$

F_{s-i} is the view factor from surface s to surface i. At steady state, Equation (4) can be simplified as

$$\frac{T_o - T_s}{R_s} + h_c(T_a - T_s) + \dot{q}'' + \sum_i F_{s-i}\varepsilon\sigma(T_i^4 - T_s^4) = 0 \tag{6}$$

In the simulated cases, all walls are adiabatic except the window. Therefore the energy balance for the entire space can be written as

$$\dot{q}_{radiator} + \dot{q}_{air} + \dot{q}_{gain} - \dot{m}_{fa}c_p(T_a - T_o) - \dot{m}_{vent}c_p(T_a - T_{vent}) - \frac{T_{win} - T_o}{R_{win}} = 0 \tag{7}$$

The term $\dot{q}_{gain}$ is the internal heat gain, which includes heat gain from occupancy, lighting and equipment load. The term, $\dot{m}_{fa}c_p(T_a - T_o)$ is the heat needed to increase the temperature of infiltration air from outside temperature T_o to inside air temperature T_a[2]. T_{vent} is ventilation air temperature and is considered to be T_0. For the remainder of this paper, we will assume that infiltration air is included in the ventilation air $\dot{m}_{vent}$. Therefore, the relevant form for the energy balance equation becomes

$$\dot{q}_{radiator} + \dot{q}_{air} + \dot{q}_{gain} = \dot{m}_{vent}c_p(T_a - T_o) + \frac{T_{win} - T_o}{R_{win}} \tag{7a}$$

If we assume all interior walls are adiabatic and the temperature of each surface is uniform, the energy balance equation for each surface can be expressed as follows by simplifying Equation (6).

For radiator surface temperature T_p:

$$h_{c-p}(T_a - T_p) + \dot{q}'' + \sum_i F_{p-i}\varepsilon\sigma(T_i^4 - T_p^4) = 0 \tag{8}$$

For window surface temperature T_{win}:

$$\frac{T_o - T_{win}}{R_{win}} + h_{c-win}(T_a - T_{win}) + \sum_i F_{win-i}\varepsilon\sigma(T_i^4 - T_{win}^4) = 0 \tag{9}$$

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[2] The heat needed may be less than the amount calculated by this term if the infiltration air (Claridge et al 1990) follows anything other than direct penetration thought the envelope such as a crack around a door.

Equations for side wall temperature T_{w1}, back wall temperature T_{w2}, floor temperature T_{w3}, and ceiling temperature T_{w4} can be generalized as follows (all four equations have the same form).

$$h_{c-wj}(T_a - T_{wj}) + \sum_j F_{wj-i}\varepsilon\sigma(T_i^4 - T_{wj}^4) = 0 \tag{10}$$

Because of symmetry, the two side walls can be assumed to have the same temperature. Therefore, a total of six equations can be obtained. If the inside air temperature T_a and radiation heating value $\dot{q}''$ is given, theoretically, the six surface temperatures can be solved. However, these equations are nonlinear and convection coefficient h_c also depends on the temperature difference between the surface and room air. To simplify these equations, the radiation term can be approximated as (Mills, 1999)

$$\dot{q}_r = \sum_i F_{s-i}\varepsilon\sigma\left(T_i^4 - T_s^4\right) = \sum_i F_{s-i}h_{ri}(T_i - T_s) \tag{11}$$

$$h_{ri} = 4\varepsilon\sigma T_{mi}^3 \tag{12}$$

$$T_{mi} = \frac{T_i + T_s}{2} \tag{13}$$

Mills (1999) has shown that when the temperature difference is less than 100K, the error of this approximation is less than 2%. When the temperature difference is less than 10K, the error is less than 0.03%. Equations (8), (9) and (10) can be simplified to six linear equations as

$$h_{c-p}(T_a - T_p) + \dot{q}'' + \sum_i F_{p-i}h_{rp}(T_i - T_p) = 0 \tag{14}$$

$$\frac{T_o - T_{win}}{R_{win}} + h_{c-win}(T_a - T_{win}) + \sum_i F_{win-i}h_{rwin}(T_i - T_{win}) = 0 \tag{15}$$

$$h_{c-wj}(T_a - T_{wj}) + \sum_j F_{wj-i}h_{rwj}(T_i - T_{wj}) = 0 \tag{16}$$

Convection heat transfer coefficient h_c is not easy to establish. In most of the previous related research, a constant coefficient is used for all vertical walls. This over-simplified method may give inaccurate results. The convection intensity between a cold window pane and inside air is quite different from the convection between warm interior walls and inside air. Natural convection at the interior walls and windows is in the range of turbulent flow according to the laminar flow criterion, $L^3\Delta T < 63$ (US units) (2002). Min et al. (1956) studied the natural convection in a panel heated room. The equations determined by Min and proposed by ASHRAE (2001) are (US units):

For a heated or cooled wall:

$$h_c = 0.26(T_s - T_a)^{0.32} \tag{17}$$

For a partially heated ceiling:

$$h_c = 0.13\left(T_s - T_a\right)^{0.25} \tag{18}$$

For a heated floor or cooled ceiling:

$$h_c = 0.31(T_s - T_a)^{0.31} \tag{19}$$

For a heated ceiling:

$$h_c = 0.02(T_s - T_a)^{0.25} \tag{20}$$

Based on Eqs. 17-20, the convection coefficient needed by Eqs. (14), (15), and (16) can be determined. In the computation program, initial guessed values are given to all h_c and h_r. Then the matrix of six equations is solved. The surface temperatures obtained are then submitted into the coefficient calculation and the matrix is re-solved until the results converge.

SIMULATION RESULTS

The objective of simulation is to obtain the surface temperatures and analyze the heating load at different radiator positions. Once the surface temperatures of the enclosure are known, the heating load of the entire space can be easily found by Equation (7a). Heating load must be compared on the basis of the same comfort level for the two cases. Operative temperature is used as an indicator of comfort. In simulation processes, the operative temperature in the space is set at a constant value. The mean radiant temperature in the center of the space can be calculated by Equation (1). However, the value obtained by this equation only reflects the MRT at a certain point. The weighted surface temperature may better represent the average MRT inside an enclosure. The following equation is used to calculate the mean radiant temperature in the space.

$$T_r = \left(\frac{\sum_i A_i T_{si}^{\ 4}}{A_{total}} \right)^{\frac{1}{4}} \tag{21}$$

T_{si} is the individual surface temperature. The air temperature is assumed uniform and determined from Equation (3), assuming the air speed is less than 40 fpm so A=0.5, resulting in the following equation:

$$T_a = 2T_{op} - T_r \tag{22}$$

Based on the heat transfer model described above, a simulation program has been written and the calculation flow chart is illustrated in Figure 4.

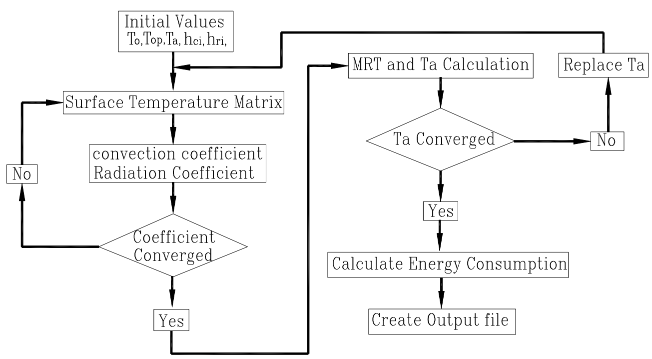

Figure 4. Calculation Flow Chart

By setting outside temperature equal to 30°F (-1.1°C), operative temperature equal to 73°F, and ventilation rate equal to 20CFM, the simulation results are shown in Figures 5 and 6.

Case 1: Two 16 ft^2 Radiators Next to Window

By increasing the radiant heating ratio and keeping operative temperature constant at 73°F (22.78°C), surface temperature, air temperature and mean radiant temperature trends are shown in Figure 5. From left to right, radiant heating ratio increases from 0 to 1. Simultaneously, the convective heating ratio decreases from 1 to 0. At 100% convective heating at the left hand side, window surface temperature is about 61.0°F (16.1°C). This temperature gradually increases to 62.7°F (17.1 °C), when radiant heating increases to 100%. From Figure 5, it can be seen that the slope of window surface temperature is larger than that of the back wall temperature. Because the radiators are much closer to the window than the back wall, radiator surface temperatures have a greater influence on window surface temperature. The increased window surface temperature increases the comfort level for an occupant who is seated near the window. However, the increased surface temperature also raises the temperature difference between the inside surface and the outside environment. The higher temperature difference will result in higher heat loss through the window which may cause the overall heating consumption of the space to rise.

As shown in Figure 5, when the heating system switches from 100% convective to 100% radiant heating (from left to right), the room air temperature can be reduced from 76.6°F (24.8°C) to 72.8 °F (22.6°C), a 4.0°F (2.2°C) difference. The lower room air temperature reduces the energy used to heat up the ventilation air. This is one of the advantages of radiant heating. The mean radiant temperature increases from 69.4°F (20.7 °C) to 73.2 °F(22.9 °C), when the enclosure is heated by 100% radiant heating.

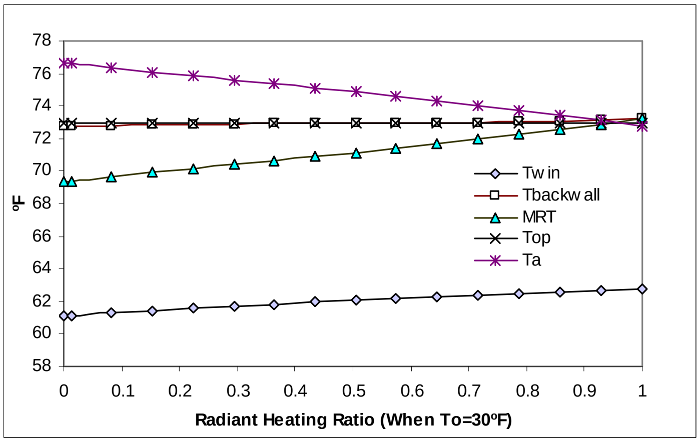

Figure 5. Temperature Trends at Different Radiant Heating Ratios for Case 1

Case 2: One 32 ft² Radiator Located in the Center of the Ceiling

Figure 6 shows the temperature trends when the radiator is located in the center of the ceiling and the radiant heating ratio increases from 0% to 100%. Compared with Figure 5, mean radiant and room air temperatures have the same trend. Room air temperature is reduced from 75.6°F (24.2°C) to 71.4°F (21.8°C), 4.2°F (2.3°C) reduction, as the radiant heating increases to 100%. The big difference between Figure 5 and Figure 6 is the slope of window surface temperature. The slope of window surface temperature is flatter in Figure 6. The increase of window surface temperature is less than 1°F, when the space switches from 100% convective heating to 100% radiant heating. The lower window surface temperature reduces the heat loss from the window, compared with Case 1. However, it may reduce the comfort level near the window. The room air temperature reduction is larger in case 2 than in case 1, which saves more energy.

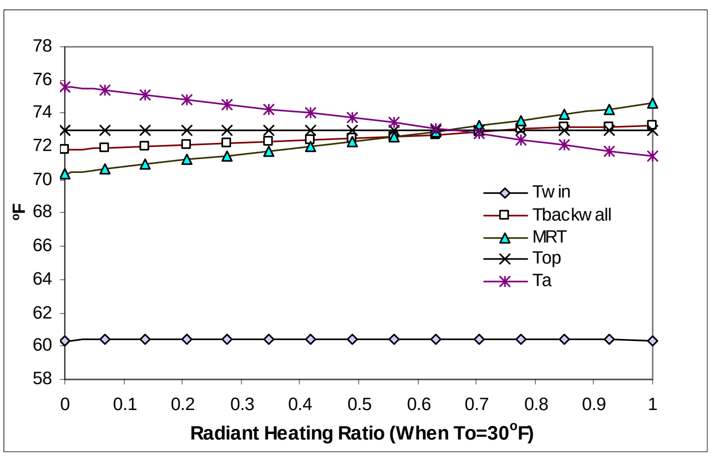

Figure 6. Temperature Trends at Different Radiant Heating Ratios for Case 2

The above observation is based on an outside air flow of 20 cfm. When the outside air flow is changed, the temperature trends of window surface, MRT and room air temperatures are almost the same. The starting and ending points of the trend lines are slightly different.

HEATING LOAD COMPARISON

For the two cases analyzed above, two factors affect heating load: room air temperature and window surface temperature. When radiant heating increases, the reduced room air temperature helps to decrease heating load. On the other hand, the increased window surface temperature adds to energy use. Figure 7 shows the total heating load for these two cases. In Case 1, the heating load increases about 2.5% for 100%

radiant heating compared with 100% convective heating. In Case 2 the heating load decreases about 1.8%. It shows the position of the radiator(s) in a typical office has some impact on heating heating load, but the impact is small. This observation is based on outside air supply of 10 CFM, which is equivalent to 0.5 ACH of infiltration rate. This is the night time operation condition, in which the mechanical ventilation stops and natural infiltration is the only source of outside air.

As the amount of outside air increases, the energy used to heat the ventilation air also increases (see Equation (7a)). This increases the relative importance of changes in room air temperature relative to changes in window surface temperature. We illustrate this by considering two higher ventilation rates for one and two occupants in the daytime.

Figure 8 shows the heating load of the two cases when outside air supply rate equals 20 cfm, which is the fresh air requirement for one occupant in this office, according to ASHRAE Standard 62-2001. For Case 1, the heating load by 100% radiant heating is close to that with 100% convective heating. When the ratio of radiant heating goes up, the heating load first goes up then goes down. This occurs since the term, $\dot{m}_{vent} c_p T_a + \frac{T_{win}}{R_{win}}$, from Equation (7a) first increases in size, then decreases as the radiant heating ratio increases. This causes the total heating consumption to increase a little and then go down. For Case 2, it can be seen the ventilation heating always decreases faster than the window heating increases, and 100% radiant heating can reduce heating by about 3.7%.

If the simulated space (150 ft^2) is occupied by two employees, the outside air requirement would be 40 cfm according to ASHRAE Standard 62-2001. The heating load is shown in Figure 9, when the heating system is switched from convective to radiant heating. It can be seen that the heating load declines about 3.6% for Case 1 and 7.6% for Case 2. Figures 7, 8 and 9 illustrate that outside air supply rate has an important effect on the energy savings of radiant heating. In the real situation, the outside air normally may be preheated to about 55°F before it is supplied to the conditioned space. Infiltration air may be heated very little, or may be heated to essentially room temperature before entering the room, depending on the nature of the openings through which it enters. If it is heated to near room temperature, the assumption of zero infiltration heat exchange is not valid, and the size of the ventilation term will decrease significantly. The maximum radiant heating ratio in the real case is less than 100 percent because of the preheating of ventilation air, providing correspondingly smaller heating savings.

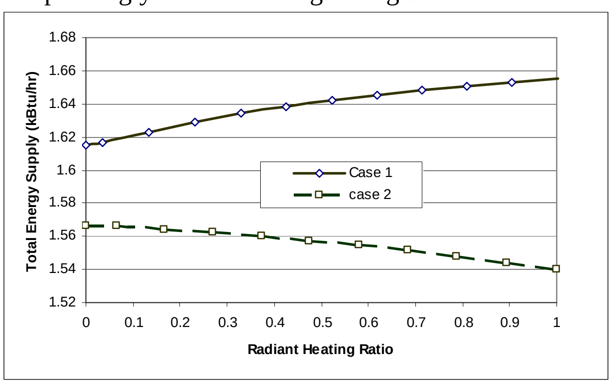

Figure 7. Heating load at Different Radiant Heating Ratios (OA=10cfm)

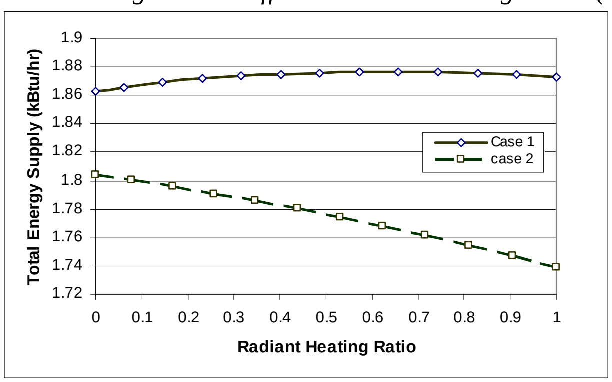

Figure 8. Heating load at Different Radiant Heating Ratios (OA=20cfm)

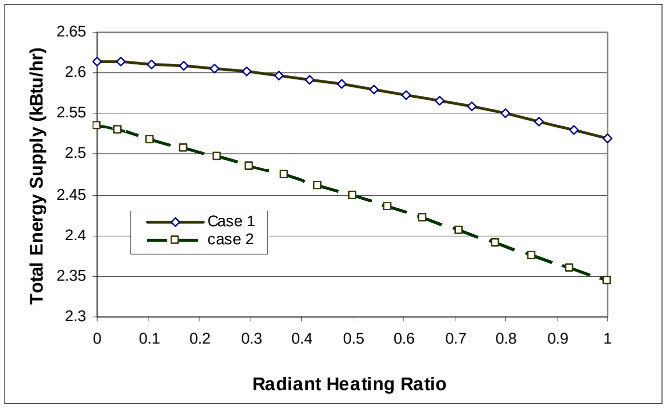

Figure 9. Heating load at Different Radiant Heating Ratios (OA=40cfm)

The above observation is based on keeping the operative temperature constant in order to keep a constant comfort level. In the real radiant heating application, the heating load of radiantly heated space also depends on the type and location of thermostat. If the operative temperature thermostat is used, the energy saving can be expected. If the dry bulb temerature thermostat is used and the setting point is kept as same as air heating, the energy saving may not be achieved. Figure 10 shows the heating load trends at different radiant heating ratio, if the room air temperature is kept constant. Figure 10 indicates the heating load even increase when radiant heating ratio increases. For case 1 at 40CFM ventilation rate, the heating load increases about 11.5% when the space switchs from 100% onvective heating to 100% radiant heating. For case 2 at 40CFM ventilation rate , the heating load increases about 7.7%. The above observation is based on keeping room air temperature at a constant of 73˚F.

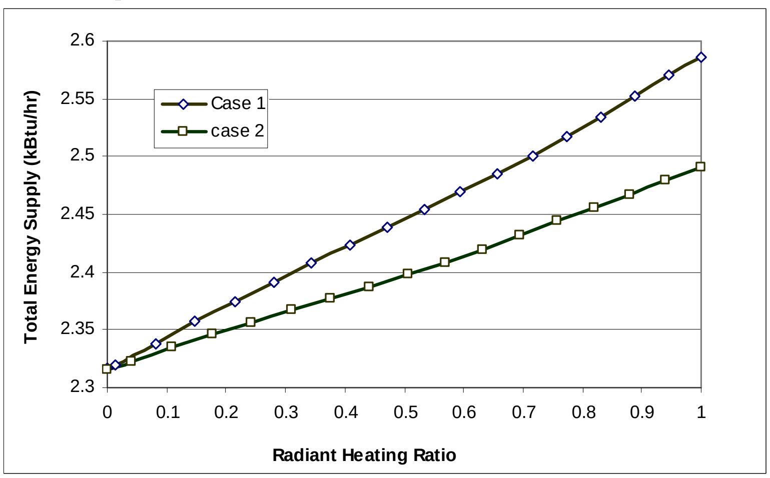

Figure 10. Heating load at Different Radiant Heating Ratios (OA=40cfm, Tair=73˚F)

THERMAL COMFORT DISTRIBUTION

In the theoretical analysis section of this paper, area weighted surface temperature is considered as the average room radiant temperature. The operative temperature is kept at a constant value. However, the mean radiant temperature actually varies with location within the enclosure. When the occupant is close to the radiator, he/she may feel warmer. When the occupant is far from the radiator, he/she may feel cooler. The thermal comfort in the two cases is not uniform. A numerical method was used to calculate the mean radiant temperature, room air temperature, and operative temperature in the three dimensional space with 100% radiant heating. The results at 4 ft (1.22 m) above the floor are shown in Figures 11 and 12.

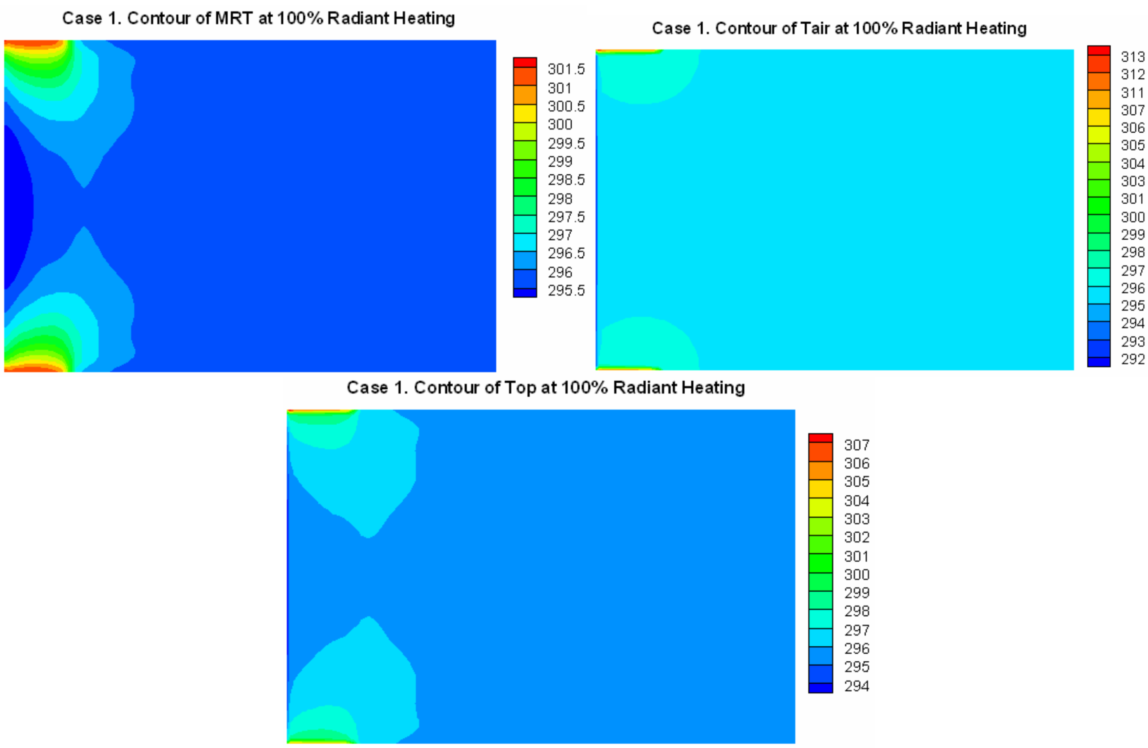

Figure 11. Temperature Distribution (K) of MRT, T_{air}, and T_{op} for 100% Radiant Heating, Case 1, 4 ft level (Window is at the left side of the space)

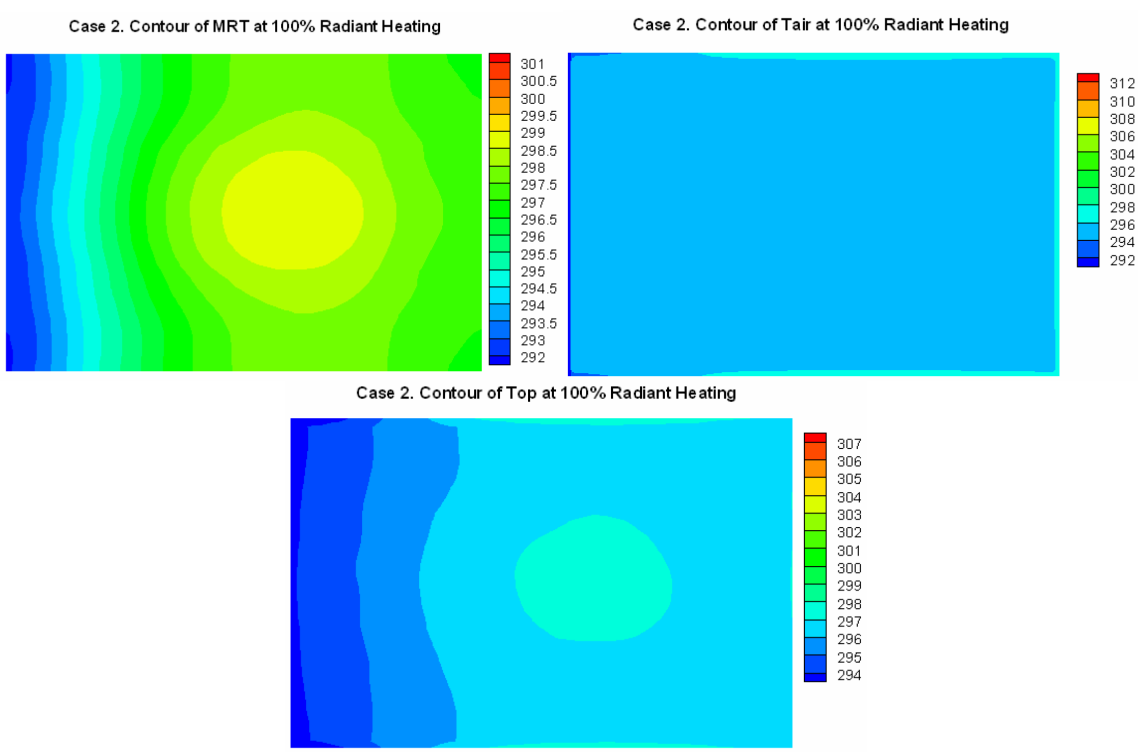

Figure 12. Temperature Distribution (K) of MRT, T_{air}, and T_{op} for 100% Radiant Heating, Case2, 4 ft level

The heat fluxes obtained from the theoretical calculation were used as the boundary conditions for radiators in the numerical analysis. Floor, ceiling, and walls are simulated as adiabatic surfaces with an emissivity of 0.9. The window was simulated as an opaque surface with an emissivity of 0.9. The discrete ordinate model was selected for radiation intensity calculation. This model has been evaluated by Truelove (1987), and by Chapman and Zhang (1995) and shown to provide quite accurate results. The Grashof numbers for window and walls are in the range of 1.3e14 to 1.3e15 which means all surfaces have a turbulent boundary layer. Therefore, the $K-\varepsilon$ turbulence model was selected for the natural convection calculation. Nielson (1998) compared four turbulence models for prediction of room airflow and showed that the $K-\varepsilon$ model was quite accurate for general application.

In Figure 11, room air temperature is 0.5K higher close to the radiator. In the remaining area, the room air temperature is almost uniform. There is a very thin layer close to the window where air temperature is near 60^{o}F (288.5K). The effect of cold window and hot radiator surfaces can be seen clearly in the mean radiant temperature distribution. Close to the radiators, the radiant temperature gradient is much higher, and MRT becomes higher. On the other hand, MRT becomes lower and the negative mean radiant temperature gradient becomes larger when close to the window in Case 1. The operative temperature is around 73^{o}F (296K) except for small areas near the radiators and window. The natural convection is created by the temperature difference between air and window inside surface. Down draft is normally caused by this natural convection in a radiantly heated space. This arrangements has the benefit in reducing down draft effects near the window. When the radiators are close to the window, they heat up the nearby air and the air flows up in an inverse direction of the down draft air. Therefore, downdraft is greatly reduced in window area.

In Figure 12, room air temperature is very even, but the mean radiant temperature has a larger gradient. MRT increases from 64^{o}F (291K, 18˚C) close to the window to 81^{o}F (300K, 27˚C) close to the center of the radiator, then decreases to 77°F (298K, 25˚C) at the back wall. Radiant temperature in one half of the room is clearly higher than the other half. The operative temperature shows thermal comfort is distributed unevenly. One third of the room is lower than 73^{o}F (296K, 23˚C). Further more, the down draft effect may be obvious because there are no measurements to control the natureal convetion near the window. Figure 11 shows that thermal comfort is more uniform in Case 1 than in Case 2. This demonstrates that the radiators near the window prevent cold penetration inside the space and enhance comfort, although this layout uses slightly more energy.

CONCLUSIONS

The position of the radiation source(s) in a radiantly heated office with a double-glazed window for the exterior wall has been shown to impact heating load and thermal comfort distribution inside the room. When radiators are close to the window (Case 1), the increase of window surface temperature is higher than when the radiator is located in the center of the ceiling (Case 2). The layout of Case 1 increases heating heating load at an outside temperature of 30°F by 2.5% in an insufficiently ventilated space. When the radiator is located in the center of ceiling, the window surface temperature increase is very small. This layout uses 1.8% to 7.6% less heating energy than convective heating for the three ventilation rates analyzed as shown in Table 2. The energy savings relative to the convective system depend on the outside air supply rate. When the outside air supply rate is larger than 20 cfm, both layouts can save energy compared with the air heating system. The heating load analysis in this paper does not include fan power savings offered by the radiant heating system.

Table 2. Energy Savings of Radiant Heating vs. Convective Heating

OA	Case 1	Case 2
10 CFM	-2.5%	1.8%
20 CFM	-0.5%	3.7%
40 CFM	3.6%	7.6%

The operation pattern also affect the energy saving of radiant heating systm. If the dry boulb thermostat is used in a radiantly heated space and temperature is set at same point of air heating, the radiant heating can increase the heating load. At the 20cfm of ventilation rate, the heating loads of layout 1 and layout 2 may increase about 11.5% and 7.7% of heating load respectively when the office temperature is set constant as same as in air heating.

On the basis of thermal comfort, radiators located close to window can reduce down draft, prevent cold penetration inside a room and make the operative temperature distribution much more uniform than when the radiator is located in the center of the ceiling. This means radiators close to the window improve the thermal comfort level inside a room although they will cause the heating load to increase a few percent.

NOMENCLATURE

C_p = Specific heat of air
F_{p-i} = View factor between occupant and room surfaces.
F_{s-i} = View factor from surface s to surface i.
F_{wj-i} = View factor from wall j to surface i.
h = Coefficient of convective heat transfer
h_c = Convective heat transfer coefficient of ceiling.
h_{c-p} = Convective heat transfer coefficient of radiator surface(s).
h_{c-win} = Convective heat transfer coefficient of window pane inside surface.
h_{c-wj} = Convective heat transfer coefficient of wall inside surface.
h_{ri} = Radiant heat transfer coefficient between two surfaces.
h_{rp} = Radiant heat transfer coefficient between Surface i and radiation panels.
h_{rwin} = Radiant heat transfer coefficient between Surface i and window.
h_{rwj} = Radiant heat transfer coefficient between Surface i and a wall surface.
K = Coefficient of thermal conductivity
$\dot{m}_{fa}$ = Infiltration mass flow rate
$\dot{m}_{vent}$ = Ventilation mass flow rate
R_s = Thermal resistance of surfaces
R_{win} = Thermal resistance of window
$\dot{q}$ = Heat generation per unit area per unit Time
$\dot{q}_{air}$ = Net heat input by air heating
$\dot{q}_{gain}$ = Internal heat gain
$\dot{q}_r$ = Net radiant heat transfer per unit area per unit time
$\dot{q}_{radiator}$ = Net heat input by radiator
T_a = Room air temperature
T_i, T_s = Surface temperature
T_{mi} = Average temperature of two surfaces
T_o = Outside air temperature
T_{op} = Operative temperature
T_p = Radiator surface temperature
T_r = Mean radiant temperature
T_{win} = Inside surface temperature of window Pane
T_{wj} = Surface temperature of wall j
T_{vent} = Ventilation air temperature
ε = Emissivity
σ = Stefan-Boltzmann constant

REFERENCES

ASHRAE. 2004 *ASHRAE Handbook-HVAC Systems and Equipment*. Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

ASHRAE. 2004 *ANSI/ASHRAE Standard 55-2004, Thermal Environment Conditions for Human Occupancy* . Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

ASHRAE. 2001 *ASHRAE Handbook-Fundamentals*. Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

ASHRAE. 2001 *ANSI/ASHRAE Standard 62-2001,Ventilation for Acceptable Indoor Air Quality* . Atlanta: American Society of Heating, Refrigeration and Air-Conditioning Engineers, Inc.

Chapman, K.S, Rutler, J., Watson, R. , 2000, Impact of Heating System and Wall Surface Temperatures on Room Operative Temperature Fields, *ASHRAE Transactions* Vol. 106(1), 506-514.

Chapman, K.S., DeGreef, Jean and Watson, Richard, 1997, Thermal Comfort Analysis Using BCAP for Retrofitting a Radiantly Heated Residence, *ASHRAE Transactions* Vol. 103(1), 959-965.

Chapman, K.S., P.Zhang, 1995, Radiant Heat Exchange Calculations in Radiantly Heated and Cooled Enclosures, *ASHRAE Transactions* Vol. 101(2), 1236-1247.

Claridge, D. E. and Bhattacharyya, S., 1990, The Measured Energy Impact of Infiltration in a Test Cell, *ASME Journal of Solar Energy Engineering*, Volume 112, 132-139.

Dale, J.D, Ackerman, M.Y., 1993, Thermal Performance of a Radiant Panel Floor-Heating System, *ASHRAE Transactions* Vol 99(1), 23-24.

DeGreef, Jean and Chapman, K.S., 1998, Simplified Thermal Comfort Evaluation of MRT Gradients and Power Consumption Predicted with the BCAP Methodology, *ASHRAE Transactions* Vol. 104(2), 1090-1097.

Fanger, P. 1967. Calculation of Thermal Comfort: Introduction of a Basic Comfort Equation. *ASHRAE Trans.* Vol. 73(2), 1-20.

Gibbs, D.R., 1994, Control of Multizone Hydronic Radiant Floor Heating Systems, *ASHRAE Trans.* Vol 100(1), 1003-1010.

Hanibuchi, Haruo and Hokoi, Shuichi, 2000, Simplified Method of Estimating Efficiency of Radiant and Convective Heating Systems. *ASHRAE Transactions* Vol. 106(1), 487-494.

Kreider, Jan F., Curtiss, Peter S, Rabl, Ari, 2002, *Heating and Cooling of Buildings*, p. 46, ISBN: 0-07-237341-5.

Mills, A. F. 1999, *Heat Transfer*, Prentice Hall, p16, ISBN: 0-13-947624-5.

Min, T.C., L.F.Schutrum, G.V.Parmelee, and J.D. Vouris. 1956. Natural Convection and Radiation in a Panel Heated Room. *ASHRAE Transactions* Vol 62, 337-358.

Nielson, Peter V. 1998, The Selection of Turbulence Models for Prediction of Room Airflow, *ASHRAE Transactions* Vol 104(1), 1119-1127.

Olesen, B.W., 1994, Comparative Experimental Study of Performance of Radiant Floor-Heating Systems and a Wall Panel Heating System Under Dynamic Conditions, *ASHRAE Transactions* Vol 100(1), 1011-1023.

Palmer, Jean and Chapman, K.S., 2000, Direct Calculation of Mean Radiant Temperature Using Radiant Intensities, *ASHRAE Transactions* Vol. 106(1), 477-486.

Steinman, M., Kalisperis, L.N., and Summers, L.H., 1989. The MRT-Correction Method – An Improved Method for Radiant Heat Exchange. *ASHRAE Transactions* Vol. 95(1), 1015-1027.

Tassou, Savvas A., Xiang, Weizhong, and Kolokotroni, Maria, 2000, Church Heating: Numerical Modeling and Comparison between Radiant and Forced Convection Systems, *ASHRAE Transactions* Vol 106(1), 495-505.

Truelove, J.S. 1987 Discrete-Ordinates Solutions of the Radiative Transport Equation. *J. Heat Transfer* 109, 1048-1051.

U.S. Department of Energy, “Radiant Floor Heating and Cooling Systems” http://www.eere.energy.gov/consumerinfo/ factsheets/bc2.html, Accessed in July 2005.