Duty Cycle of a Radiant Ceiling Heating System under Extremely Cold-Temperature Conditions: A Theoretical Assessment

Mohamed Lateb and Hachimi Fellouah

Department of Mechanical Engineering, Université de Sherbrooke, 2500 Blvd. de l’Université, Sherbrooke, QC J1K 2R1, Canada

Abstract: This work focuses on the estimation of a duty cycle of a radiant ceiling heating system with a panel surface temperature of 35 °C and a heat flux of 65 W/m² that corresponds to a thermal comfort for sedentary occupants. The results obtained are based on the theoretical heat transfer equations that govern the radiant and natural convection heat exchange mechanisms, and experimental heat transfer coefficients available in the literature. The results of the examined radiant heating system with specific conditions showed that a duty cycle of 6.46 min alternated by 13.36 min in shutting-down position is required to assure an acceptable thermal comfort for the enclosure space occupants. In addition, the study showed that for extremely cold-temperature conditions the heating system requires a daily operating load of about 61.2% which clearly proves the efficiency of these radiant heating systems in terms of energy consumption.

Key words: Radiant ceiling panels, natural convection transfer, thermal comfort, radiant heating systems, duty cycle, energy consumption.

1. Introduction

Radiant heating systems are different from the typical heating, ventilation and air-conditioning (HVAC) and/or conventional systems because of their specific thermal heat exchange. These systems heat surfaces rather than the surrounding air, and operate at a lower temperature which offers them a large advantage of saving energy and providing higher level of thermal comfort [1]. A system is specifically called radiant system if 50% or more of the total heat transfer is exchanged by radiation mechanism compared to the other transfer phenomena associated to the designed system [2]. Their main advantages are well summarized in works of Oxizidis and Papadopoulos [3]. Among these advantages one can cite: (i) the provision of better thermal comfort conditions by transferring heat mainly by radiation and less by convection, (ii) the elimination of the local thermal discomfort due to the low temperature of heating, (iii) the limitation of the air exchange heat losses during the heating by decreasing the mean radiant temperature (MRT), (iv) the absence of noise and draught due to the fan operation, and (v) the use of low-temperature heating enables the efficient use of the heat sources. However, these systems are often slow in the temperature adjustment and installation work can be costly and difficult, especially when renovating old buildings [4].

Regarding the principle of the radiation heat transfer mechanism, the thermal radiation is caused by energy emission in the form of electromagnetic waves. For radiant systems in buildings, the radiant heating occurs mostly in the infrared (long-wave) region because of the low temperature of bodies in buildings environment [5]. Furthermore, according to Versteeg and Malalasekera [6] most engineering systems emit thermal radiation at infrared wavelengths (0.7-100 μm) and the peak wavelength of sources at room temperature is about 10 μm. Buckley [7] claimed that
even at severe climatic conditions, it is possible to maintain with thermal radiation correct conditions for human thermal comfort. Thermal comfort sensations of a space occupants’ are in general affected by climatic parameters [3] that are air temperature, mean radiant temperature, air velocity, relative humidity and atmospheric pressure [8]. The mean radiant temperature parameter has a strong influence on human thermal comfort and the magnitude of the effect is slightly greater than that of air temperature at low air velocities typical to most indoor spaces [5].

Karmann et al. [9] reviewed the main existing methods for evaluating the thermal comfort. The authors listed three main approaches that are: (i) building performance simulation (BPS), (ii) physical measurements1, and (iii) human subject testing/occupant based surveys. Building performance simulations are based on computer simulation programs. Among them, one can cite computational fluid dynamics (CFD) able to simulate the detailed airflow patterns and temperature distribution within the space, and whole building energy simulation used to model zones and predict indoor conditions energy use for buildings. These techniques allow the prediction of the building behaviour from the early design steps, and the detail assessment of the contribution of all the different building components (e.g. opaque envelope, glazing and shading systems, HVAC systems, control strategies, etc.), and can provide information about the global performance even with various levels of complexity and uncertainty [10]. Physical measurements are a common way to quantify thermal comfort through the measure of climatic parameters2. These approaches are very expensive—compared to BPS methods—since they involve personnel, instrumentation, buildings/rooms and other indirect costs in addition to the time-consuming.

In addition, with certain activity levels and clothing types of occupants, radiant temperature asymmetry, turbulence intensity, and contaminant concentration are very important parameters to quantitatively investigate for evaluating the thermal comfort [11]. Consequently, the physical measurement campaigns are not straightforward tasks to achieve in order to obtain detailed and reliable measurement data. The last approach, namely the human subject testing/occupant based surveys, is based on subjectively perceived comfort by a human. It applies to a group of participants, under specified thermal conditions, that fill a survey after experimental test series. The methodology is very expensive as well, for the same reasons than the physical measurements approach. It implies comfort reference indices that are “predicted mean vote” (PMV) and “predicted percentage of dissatisfied” (PPD) which, in turn, are based on the occupant thermal satisfaction depending on the local properties such as air and mean radiant temperatures [10]. According to Olesen [12], the recommended criteria proposed in ISO 7730 for an acceptable moderate thermal environment using one of these two reference indices are 
\[
-0.5 < PMV < +0.5 \text{ or } PPD < 10\%.
\]

This work focuses on theoretical calculations to estimate the duty cycle and daily operating loads of a radiant ceiling heating system for an optimal thermal comfort and energy consumption. For that, the paper is organized as follows. Section 2 introduces the reader to the radiant ceiling heating/cooling systems. Section 3 details the various transfer mechanisms implied in the radiant heating systems and heat fluxes compromised in the energy balances. The required time of cooling and heating is estimated in Section 4. The analysis and discussion of results are presented in Section 5. Finally, the main findings of the work are summarized in Section 6.

2. Radiant Ceiling Heating/Cooling Systems

Heat transfer performance of radiant ceiling
heating/cooling systems is mainly conducted by radiation and natural convection transfer mechanisms [5]. The heat transfer performance is the most important design parameter during dimensioning processes of the engineering applications [13]. Numerous authors have investigated thermal performances of radiant systems including heating and cooling operations, e.g. Refs. [14-16]. According to works of Miriel et al. [14], the authors reported that the radiation transfer represents about 80% and 66% of the total heat transfer amount of a radiant ceiling heating and cooling system, respectively. While the simplified calculation model proposed by Okamoto et al. [15] provided portions of radiant heat transfer—from the total heat transfer—varying in a range of 70-80% in case of ceiling radiant heating panels and 60% in case of cooling panels.

Radiant systems are classified by the temperature of the heated system. Radiant heating applications are classified as panel heating if the surface temperature is below 150 °C [17]. They provide a comfortable environment by controlling surface temperatures and minimizing air motion within a space. Surface temperatures are limited to the ability of the material and to what is comfortable to occupants’ feet (e.g. the maximum effective surface temperature of floor panels). In case of electric ceiling panels [5], prefabricated models—constructed with metal, glass, or semirigid fiberglass board or vinyl—have heated surface temperatures ranging from 40 to 150 °C, with corresponding heat fluxes ranging from 270 to 1,100 W/m². Other models exist—like a panel of gypsum board embedded with insulated resistance wire generally installed as a part of the ceiling—and have heat flux limited to 240 W/m² to maintain the heated surface temperature below 40 °C. In the case of this work, an electric ceiling panel with a surface temperature of 35 °C and heat flux of 65 W/m² corresponding to thermal comfort of sedentary occupants [17] is selected to estimate the necessary duty cycle for optimizing the energy consumption. It is to note that these characteristic data correspond to a building that has radiant heating systems already installed.

3. Theoretical Energy Balances and Heat Flux Transfers

Knowing that sensible heating panels transfer heat through temperature-controlled surface to the indoor space and its enclosure surfaces by natural convection and thermal radiation [5], the energy balance can be traduced by the equality between the total heat flux emitted from the ceiling panel \( q_{\text{tot}} \) and the summation of the radiant heat flux transferred to the unheated surfaces \( q_{\text{rad}} \) and the natural convective heat flux received by the ambient air \( q_{\text{conv}} \).

In the following sections, the various heat transfer balance equations of the different transfer modes (i.e. total heat flux transfer, convective and radiant heat flux transfers) and the calculation methodologies are detailed. As introduced in previous sections, some researchers [8, 14-16, 18, 19] have studied the heat transfer coefficients (HTC) specific to the convective \( h_{\text{conv}} \) and radiant \( h_{\text{rad}} \) transfer modes [13]. The experimental values of HTC are developed in what follows for theoretical estimation of each heat flux transfer.

3.1 Total Heat Transfer

The total heat transfer equation can be written as follows:

\[
q_{\text{tot}} = q_{\text{rad}} + q_{\text{conv}}
\]  

(1)

When detailing the combined heat fluxes (thermal radiation and natural convection) emitted from the heating panel surface, the equation above (Eq. (1)) can be written—using a total heat transfer coefficient—as follows:

\[
q_{\text{tot}} = h_{\text{tot}}(T_{\text{pan}} - T_{\text{ap}})
\]  

(2)

where \( h_{\text{tot}} \) is the total heat transfer coefficient that comprises the radiant and convective phenomena (in
W/m²·K), and $T_{\text{pan}}$ and $T_{\text{op}}$ represent the panel surface and operative temperatures (in K), respectively.

Many total HTC values are gathered by ASHRAE [20]. Among them the value of 8.29 W/(m²·K) is specific for a horizontal surface with heat flow down that is well representing the case of radiant ceiling heating system. A very close value of 8.4 W/(m²·K) was obtained by Koca et al. [1] in their experimental works. As noted by the standard EN 15377-1 [21], a value of 8 W/(m²·K) is generally recommended. However, in this work, the value of 8.29 W/(m²·K) is judged to better represent the studied case configuration, therefore the value will be considered in the theoretical estimation of the total heat flux transfer.

In order to estimate the total heat transfer that can be exchanged between the heating source and the enclosure surfaces, the heat flux transferred from the panel surface temperature $T_{\text{pan}}$ (35 °C) to $T_{\text{op}}$ (22 °C)—that will be referred as the testing case hereafter—using the selected HTC of 8.29 W/(m²·K) is calculated. The numerical calculation gives a result of $q_{\text{tot}} = 107.77$ W/m².

### 3.2 Heat Transfer by Convection

The equation of the heat transfer exchanged by the natural convection between the ambient air of the space and the panel surface temperature is:

$$q_{\text{conv}} = h_{\text{conv}} (T_{\text{pan}} - T_{\text{amb}})$$  \hspace{1cm} (3)

where $h_{\text{conv}}$ is the convective heat transfer coefficient (in W/m²·K), and $T_{\text{pan}}$ and $T_{\text{amb}}$ represent the panel surface and the ambient air temperatures (in K), respectively.

Heat flux from natural convection ($q_{\text{conv}}$) occurs between the indoor air and the temperature-controlled panel surface. In this mechanism, warming the boundary layer air at the panel surface generates air motion, and the induced thermal convection coefficient is not easily established [5]. The complex determination of the coefficient can be due to many factors (e.g. the indoor space configuration, the occupants’ movement, the mechanical ventilation systems...) that affect the natural convection.

In this view, many research works have been carried out for the thermal convection coefficient determination. All the reviewed values of convective HTC's listed hereafter are exclusively investigated in case of heated radiant ceiling studies. Miriel et al. [14] have proposed a convective HTC of 3.0 W/(m²·K), while the experimental works of Koca et al. [1] suggested a value of 2.7 W/(m²·K). Finally, regarding the value proposed by the experimental studies, the value of 3.0 W/(m²·K) is selected as HTC for the natural convection for the rest of the study.

As done for the total heat flux ($q_{\text{tot}}$) with the testing case, the calculated value of heat flux using the selected HTC of 3.0 W/(m²·K) gives a result of $q_{\text{conv}} = 39$ W/m². In term of percentage, the heat transferred by natural convection represents 35% of the total heat flux transferred for the testing case (see the footnote 4).

### 3.3 Heat Transfer by Radiation

The estimation of the radiant heat transfer uses a reference temperature that is defined as the average unheated surface temperature $T_{\text{AUST}}$ and obtained by using surface temperatures and view factors [13]. For instance, any unheated surface in the same plan with the panel is not account for the average unheated surface temperature (AUST). The equation of the radiant heat flux can be obtained by two calculation methods.

- Using an estimated radiative heat transfer coefficient (HTC), the equation is as follows:

$$q_{\text{rad}} = h_{\text{rad}} (T_{\text{pan}} - T_{\text{AUST}})$$  \hspace{1cm} (4)

where $T_{\text{pan}}$ represents the surface temperature of the ceiling panel (in K).

Regarding the experimental work—including the simulation of the typical condition of occupancy—carried out by Causone et al. [18], the
authors argued that the value of radiant HTC can be considered constant at 5.6 W/(m²·K) for both cases (heated and cooled ceiling). Zhang et al. [19] proposed a value of 5.5 W/(m²·K) while Koca et al. [1] suggested a value of 5.7 W/(m²·K) for radiant HTC. For this work, the radiant HTC value of 5.6 W/(m²·K) is selected and will be considered for the rest of this study.

For the testing case, the HTC value of 5.6 W/(m²·K) for radiation transfer mode gives a result of \( q_{rad} = 72.8 \) W/m². In terms of percentage, the heat transferred by radiation convection represents about 65% of the total heat flux exchanged in the testing case. When the total heat flux is calculated by adding the radiation heat flux and the natural convection heat flux, it is found that the total heat flux is about 111.8 W/m². This value is very close to the previous total heat flux value (107.77 W/m²) calculated in previous section using Eq. (2). The relative error is about 3% which widely justifies the selected values of HTCs for this study. In addition, the percentage of the portions transferred by radiation (65%) and natural convection (35%) shows that the heating system remains conform to what is found in the literature.

• Using the basic equation—for multi-surface enclosure with diffuse isothermal grey surfaces derived by radiosity formulation methods [22]—simplified by reducing a multi-surface enclosure to a two-surface approximation by means of the mean radiation temperature (MRT) method, the heat transfer equation writes as:

\[ q_{rad} = \sigma F_r \left( T_{pan}^4 - T_{AUST}^4 \right) \]  

(5)

where \( \sigma \) is the Stephan-Boltzmann constant (= 5.67 × 10⁻⁸ W/m²·K⁴) and \( F_r \) the radiation exchange factor (dimensionless).

The detailed equation of the radiation interchange factor \( F_r \) for two-surface radiation heat exchange can be found in ASHRAE [5]. The equation expression depends on the radiation angle factor from panel to fictitious surface, the area of the panel and the fictitious surfaces. According to the document [5], the term \( (\sigma F_r) \) can be well approximated by the value of \( 5 \times 10^{-8} \) W/(m²·K⁴). Therefore, this value is selected for use when it is question of radiant heat transfer using the MRT method.

The testing case provided a value of 71.60 W/m² using the selected \( (\sigma F_r) \) value. This result is in total agreement with the value found (72.8 W/m²) using Eq. (4), and an insignificant relative error of 2% is found.

Finally, the consistency of all the heat transfer coefficients (HTCs) found in the literature and mainly selected for the present study is demonstrated and justified.

4. Estimation of the Required Heating Time

The calculation of the heating and/or cooling time is evaluated from the first law of thermodynamics. The latter may often be used to determine an unknown temperature at an instant of time. In this case, it will be used to evaluate the time necessary to a warmed ambient space to decrease its internal thermal energy by losing heat up to the minimum threshold of the comfort temperature (18 °C). These calculations will be achieved mainly by calculating the losses of heat induced by conduction transfer through the outdoor frontage wall for which the outdoor side is assumed to be at a temperature of 0 °C. It is supposed that no mechanical ventilation and no infiltration occur in the indoor space. In the case of radiant heating systems, the assumption of losing the main heat by conduction seems reasonable because these systems mainly heat the enclosure surfaces like walls that, in turn, transmit their heat by conduction to the outdoor frontages.

The outdoor frontage wall is mainly composed by: (i) one layer of gypsum board from the indoor side with a
thickness of 0.01 m, thermal resistance of 0.11 m² K/W, density of 650 kg/m³ and specific heat of 1,880 J/(kg·K), and (ii) one layer of concrete at the outdoor side with a thickness of 0.30 m, thermal resistance of 0.217 m² K/W, density of 2,000 kg/m³ and specific heat of 920 J/(kg·K). After calculations, the overall heat transfer coefficient (U) of the wall is about 3.05 W/(m²·K). All these values can be found in data tables of ASHRAE [20].

The different equations for the various transfer of heat due to energy generation, thermal storage and heat transfer by convection, conduction and radiation are detailed below:

1. Storage of the heat energy by the indoor side wall: the variation of the internal thermal energy of the gypsum board layer is equal to the radiant heat received from the heating system. The equation below is obtained when using an estimated radiative heat transfer coefficient (see Eq. (4)).

$$\rho \cdot V \cdot C_g \cdot \frac{dT}{dt} = q_{\text{rad}} A + h_{\text{rad}} A (T_{\text{pan}} - T_{\text{AUST}})$$ (6)

where $q_{\text{rad}}$ is the portion (65% of 65 W/m² = 42.25 W/m²) of the continuous heat flux—exchanged between the wall and the heating source by radiation—during all the period of heating to reach the wanted temperature. $A$ is the surface area equal to unity ($A = 1 \text{ m}^2$), and this value will remain valid for all this work.

The initial conditions are assumed to be equivalent to the first operation of the heating system (i.e. $T_{\text{pan}}$ and $T_{\text{AUST}}$ are equal to 35 and 0 °C, respectively). Eq. (6) provides the increasing of the temperature $T_{\text{AUST}}$ as a function of time. It allows to determine how time the heating system should be operating to reach a given indoor wall surface temperature value.

2. In the same manner the evolution of the indoor ambient temperature heated by convection transfer mode is obtained by the solution of the equation below:

$$\rho_{\text{air}} V_{\text{air}} C_{\text{air}} \frac{dT}{dt} = q_{\text{conv}} A + h_{\text{conv}} A (T_{\text{pan}} - T_{\text{amb}})$$ (7)

where $q_{\text{conv}}$ is the portion (35% of 22.75 W/m²) of the continuous heat flux—exchanged between the ambient and the heating source by convection—during all the period of heating.

The solution of Eqs. (6) and (7) above highlights the required time for the internal energy of the considered material (i.e. gypsum and air) to lower (by losing heat) or increase (by storing heat) its internal temperature. The opposite is also valid, i.e. the temperature can be determined at each time. The exact solution (more details can be found by Incropera et al. [23]) may be obtained if all quantities, excluding the temperature, are independent of time. If this is the case, the equation of time $t$ can be expressed as:

$$t = -\frac{\rho V C}{hA} \ln \left(\frac{T - T_{\text{pan}} - q/q_{\text{rad}}}{T_{\text{in}} - T_{\text{pan}} - q/q_{\text{conv}}}\right)$$ (8)

where $T_{\text{in}}$ corresponds to $T_{\text{AUST}}$ or $T_{\text{amb}}$ depending of the equation in use (i.e. Eq. (6) or (7)) and $T$ represents the wanted temperatures of the material. The parameters $\rho$, $V$, $C$ and $A$ refer to the corresponding material (i.e. gypsum or air), while $q$ and $h$ refer to the heat transfer mode (i.e. radiation or convection) regarding Eqs. (6) or (7).

3. Losses of the energy heat from the indoor side towards the outdoor side of the wall: the variation of the internal thermal energy of the gypsum board layer is equal to the conduction heat that crosses the wall towards the outdoor ambient at 0 °C. The equation writes as:

$$\rho \cdot V \cdot C_g \cdot \frac{dT}{dt} = -UA (T_{\text{in}} - T_{\text{out}})$$ (9)

where $U$ is the overall heat transfer coefficient of the wall (= 3.05 W/m²·K), $T_{\text{in}}$ and $T_{\text{out}}$ are the indoor and outdoor side temperatures of the wall, respectively. For the initial conditions, their values are respectively equal to 22 and 0 °C. The solution of Eq. (9) provides the necessary time for the indoor side wall to evacuate
a part of the stored heat to the outdoor side, thus lowering its surface temperature from 22 to 18 °C.

In the same manner as for Eq. (8) and with the same assumptions, the exact solution can be expressed as follows:

$$ t = -\frac{\rho g V^2 c_p}{UA} \ln \left( \frac{T - T_{out}}{T_{in} - T_{out}} \right) $$  \hspace{1cm} (10)

5. Results and Discussion

5.1 Operating Period of the Heating System

Fig. 1—deduced from Eq. (8)—shows the evolution in time of the temperature obtained on: (i) the enclosure surfaces by the radiation transfer mode and (ii) the indoor ambient air by the convection transfer mode. As can be seen in Fig. 1 the necessary time to the enclosure surface to reach the temperature of 22 °C from 0 °C is about 26.46 min by the radiation transfer mode, while the indoor air ambient takes more time (about 49.40 min) to reach that temperature by the convection transfer mode. As the radiant transfer is the main mechanism of heating when using radiant heating systems, therefore the time of 26.46 min is retained as the necessary time. For case where the enclosure space
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Fig. 3  Duty cycle of a radiant heating system operating with a heat flux of 65 W/m² under a temperature range of 22 °C and 18 °C.

Fig. 4  Daily operating load of a radiant heating system depending on the outdoor temperature.

is already at a temperature of 18 °C, the required time is 6.46 min to increase the temperature up to 22 °C.

5.2 Shutting-Down Period of the Heating System

As can be deduced from Fig. 2, the necessary time to the indoor wall surface to lower its surface temperature from 22 °C to 18 °C by the conduction heat transfer mode is about 13.36 min.

5.3 Duty Cycle Calculation

This subsection will summarize the operating duty cycle to optimize the use of the radiant heating system. Fig. 3 shows the operating and shutdown periods of radiant heating system using a power heat of 65 W/m² with a panel surface temperature of 35 °C.

When the curve is analyzed, one can note that for a space that is not heated at all (i.e. the temperature within the ambient is supposed to be at 0 °C), the radiant heater has to operate during 26.46 min to

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* The required value of 6.46 min is deduced from the evolution curve of the radiant transfer mode (see Fig. 1) between 18 and 22 °C.
increase the surfaces temperature to 22 °C. Therefore, the maximum limit of thermal comfort is reached. Stopping the radiant heating system during 13.36 min will permit to the temperature to lower to 18 °C that is the minimum limit of the thermal comfort commonly indicated by standard handbooks such as ASHRAE [17]. After that, the heating system should be switched on to maintain the comfort until the temperature becomes 22 °C once again. This temperature increase requires 6.46 min for operating the heating system. Finally, the radiant system requires a duty cycle of 6.46 min alternated by 13.36 min in shutting-down position.

5.4 Daily Operating Load

The daily operating load of the heating system as function of the outdoor temperature \(T_{out}\) is represented in Fig. 4.

For instance, the curve indicates that for an outdoor temperature of 0 °C, the daily operating load for the studied radiant heating system is about 32.6%. In case of extremely cold-temperature conditions like −45 °C, the heating system will operate with a load of 61.2%. This value highlights the efficiency of radiant heating systems in terms of energy consumption since other systems like baseboard heating systems that are used by 95% of single-family houses—require up to 95-99% of daily operating load [24] to ensure a thermal comfort for this range of extreme cold temperatures.

6. Conclusions

This work focused on theoretical duty cycle estimation of a radiant heating system with a heat flux of 65 W/m² that corresponds to sedentary occupants’ comfort and a panel surface temperature of 35 °C. The results obtained are based on various parameters—such as heat transfer coefficients—available in the literature and for which the consistency is well demonstrated and justified. As a finding for the specific system examined with taking into account all the supposed assumptions, conclusions can be summarized as follows:

- The heat transfer in radiant heating systems is mainly driven by the radiant transfer and natural convection. Their portions within the total heat transfer are approximately about 65% and 35% for the radiant and convection mechanisms, respectively.
- For an enclosure initially at a temperature of 0 °C, the radiant heating system will take about 26.46 min to increase surface temperatures to 22 °C by the radiant transfer, while by the natural convection transfer, the desired temperature will be reached in 49.40 min.
- The cooling process is supposed to mainly be driven by the conduction transfer mode through the external frontage wall. For an external temperature of 0 °C, the ambient temperature, initially at 22 °C, will take 13.36 min to be lowered to 18 °C by the heat losses.
- The operating duty cycle for the studied system should work as follows. Once the thermal comfort conditions are obtained in the enclosure space, the radiant system will require a duty cycle of 6.46 min alternated by 13.36 min in shutting-down position to assure keeping the ambient temperature within the thermal comfort range of temperature.
- The radiant heating systems are very efficient in terms of energy consumption. For instance, in case of extremely cold-temperature conditions like −45 °C, the studied radiant heating system requires a daily operating load of about 61.2% while other systems—e.g. baseboard heaters—can require up to 95-99% of daily operating load to ensure the same thermal comfort during these cold temperatures.

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References

[1] Koca, A., Gemici, Z., Topacoglu, Y., Cetin, G., Acet, R. C., and Kanbur, B. B. 2014. “Experimental Investigation of Heat Transfer Coefficients between Hydronic Radiant Heated Wall and Room.” Energy and Buildings 82: 211-21.

[2] Chartered Institution of Building Services Engineers. 2005. “Chapter 1: Heating.” In Heating, Ventilating, Air Conditioning and Refrigeration: CIBSE Guide B. London, United Kingdom.

[3] Oxizidis, S., and Papadopoulos, A. M. 2013. “Performance of Radiant Cooling Surfaces with Respect to Energy Consumption and Thermal Comfort.” Energy and Buildings 57: 199-209.

[4] Myhren, J. A., and Holmberg, S. 2008. “Flow Patterns and Thermal Comfort in a Room with Panel, Floor and Wall Heating.” Energy and Buildings 40: 524-36.

[5] ASHRAE. 2016. “Chapter 6: Radiant Heating and Cooling” In ASHRAE Handbook HVAC Systems and Equipment. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, United States.

[6] Versteeg, H. K., and Malalasekera, W. 2007. An Introduction to Computational Fluid Dynamics: The Finite Volume Method. 2nd ed. Pearson.

[7] Buckley, N. A. 1989. “Application of Radiant Heating Saves Energy.” Ashrae Journal 31: 17-26.

[8] Olesen, B. W., Mortensen, E., Thorshauge, J., and Berg-Munch, B. 1980. “Thermal Comfort in a Room Heated by Different Methods.” Ashrae Transactions 86: 34-48.

[9] Karmann, C., Schiavon, S., and Bauman, F. 2017. “Thermal Comfort in Buildings Using Radiant vs. All-Air Systems: A Critical Literature Review.” Building and Environment 111: 123-31.

[10] Atzeri, A. M., Cappelletti, F., Tzempelikos, A., and Gasparella, A. 2016. “Comfort Metrics for an Integrated Evaluation of Buildings Performance.” Energy and Buildings 127: 411-24.

[11] Chen, Q. 1990. “Comfort and Energy Consumption Analysis in Buildings with Radiant Panels.” Energy and Buildings 14: 287-97.

[12] Olesen, B. W. 1995. “International Standards and Ergonomics of the Thermal Environment.” Applied Ergonomics 26: 293-302.

[13] Koca, A., and Cetin, G. 2017. “Experimental Investigation of Heat Transfer Coefficients of Radiant Heating Systems: Walls, Ceiling and Wall-Ceiling Integration.” Energy and Buildings 148: 311-26.

[14] Miriel, J., Serres, L., and Trombe, A. 2002. “Radiant Ceiling Radiant Panel Heating-Cooling Systems: Experimental and Simulated Study of the Performances, Thermal Comfort and Energy Consumptions.” Applied Thermal Engineering 22: 1861-73.

[15] Okamoto, S., Kitora, H., Yamaguchi, H., and Oka, T. 2010. “A Simplified Calculation Method for Estimating Heat Flux from Ceiling Radiant Panels.” Energy and Buildings 42: 29-33.

[16] Jeong, J. W., and Mumma, S. A. 2007. “Practical Cooling Capacity Estimation Model for a Suspended Metal Ceiling Radiant Cooling Panel.” Building Environment 42: 3176-85.

[17] ASHRAE. 2015. “Chapter 54: Radiant Heating and Cooling.” In ASHRAE Handbook HVAC Applications, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, United States.

[18] Causone, F., Corgnati, S. P., Filippi, M., and Olesen, W. B. 2009. “Experimental Evaluation of Heat Transfer Coefficients between Radiant Ceiling and Room, Mixed Convection from Heated Room Surfaces.” Energy and Buildings 41: 622-8.

[19] Zhang, L., Liu, X. H., and Jiang, Y. 2013. “Experimental Evaluation of the Suspended Metal Ceiling Radiant Panel with inclined Fins.” Energy Buildings 62: 522-9.

[20] ASHRAE. 2013. “Chapter 26: Heat, Air, and Moisture Control in Building Assemblies—Material Properties.” In ASHRAE Handbook Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, United States.

[21] Dansk Standard Organization. 2008. EN 15377-1. Heating Systems in Buildings—Design of Embedded Water Based Surface Heating and Cooling Systems—Part 1: Determination of the Design Heating and Cooling Capacity.

[22] ASHRAE. 2013. “Chapter 4: Heat Transfer.” In ASHRAE Handbook Fundamentals, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, United States.

[23] Incropera, F. P., Dewitt, D. P., Bergman, T. L., and Lavine, A. S. 2006. Fundamentals of Heat and Mass Transfer. 6th ed. John Wiley & Sons, 16.

[24] Lefebvre, S., and Desbiens, C. 2002. “Residential Load Modeling for Predicting Distribution Transformer Load Behavior, Feeder Load and Cold Load Pickup.” Electrical Power and Energy Systems 24: 285-93.