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A novel evaluation regarding the influence of surface emissivity on radiative and total heat transfer coefficients in radiant heating systems by means of theoretical and numerical methods

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ABSTRACT

In the present study, we investigate by means of theoretical calculations and the program Engineering Equation Solver (EES) total and radiative heat transfer coefficients in an enclosure heated from its one wall at several surface emissivities ($\varepsilon = 0.90$, 0.85, 0.80), with room dimensions ($L \times H \times W = 1.8 \times 2.85 \times 1.8$, $3 \times 2.85 \times 3$, $3 \times 2.85 \times 4$, $4 \times 2.85 \times 3$, $6 \times 2.85 \times 6$ m) and surface temperatures ($T_h = 20$–$40\,^{\circ}$C, $T_c = 5$–$15\,^{\circ}$C, $T_f = 14$–$31\,^{\circ}$C and $T_{ceil} = 10$–$27\,^{\circ}$C). To understand the influence of convection through computational methods in the aforementioned conditions, we performed numerical procedures and found a correlation including the aspect ratio influence (H/L). Afterward, through the tables and figures, we show that the average radiative coefficient lies within the range of 5.4–5.5 W/m^2 K and varies very slightly with room dimensions and temperature differences at practical applications at the surface emissivity value of 0.9, which is a close value to a surface emissivity practically. Furthermore, we found the total heat transfer coefficient to be between 10.2 and 10.8 W/m^2 K for the emissivity value of 0.9, and we observed that it did not significantly vary with room dimensions. Additionally, similar to the experimental literature results, we found that the average proportion of radiation to the total heat transfer was between 64% and 67%.

1. Introduction

Hydronic radiant heating and cooling panels have been used instead of conventional systems with respect to their high comfort and energy efficiency in heat transportation. Many heating and cooling systems, such as solar collectors and heat pumps, which benefit from renewable energy sources that procure low-temperature water, can be operated in association with radiant panels. Using a radiant heating panel allows engineers to select a wide panel at a lower surface temperature. Also, in radiant heating systems the temperature difference between the surface and the room temperature will decrease, and this will lead to improvement in thermal comfort in terms of lowering air movements. Thermal emissivities of the panel surfaces, dimensions of the enclosure and also the thermal boundary conditions of the walls determine the heat transfer that will occur between surfaces of the enclosure. In this context, it should be stated that heat is transferred in the enclosure through radiation and convection, while the outweighing mechanism is radiation. The parameters that carry utmost importance in the calculation and sizing process of radiant heating and cooling systems are heat transfer coefficients. Accurate determination of heat transfer coefficients in enclosures equipped with radiant heating and cooling panels has carried great significance with respect to thermal comfort, heating load evaluations and energy economics.

In the literature, there are many researches, as well as derived correlations over both convective and radiative heat transfer coefficients. Nevertheless, most of these equations have well-defined only the radiative heat transfer coefficient. Due to some uncertainties in the field of convection, the convective and total heat transfer coefficients that develop in enclosures have not been clearly defined yet.

According to many researchers' investigations, diverse usages of convective heat transfer coefficient correlations in heating system simulations result in inaccurate predictions of heating load calculations in buildings. Therefore, the proper usage of convective heat transfer coefficients in real-size rooms has utmost importance not only in terms of thermal comfort but also energy consumption. A few studies in this context are as follows:

ASHRAE Technical Committee [1] have regarded the convective modeling of surfaces in building energy simulation programs as a remarkable investigation issue. Also, Beausoleil-Morrison [2]

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Nomenclature

A	area (m^2)
AUST	average unheated surface temperature (°C)
CFD	computational fluid dynamics
D	hydraulic diameter (m)
E_{bi}	blackbody emissive power of examined surface (W/m^2)
EES	Engineering Equation Solver
F_{i-j}	view factor between the examined and j surfaces
F_{s-j}	view factor between the radiant surface and j surfaces
F_{l-n}	view factor between the examined surface and the surroundings
H	height of the enclosure (m)
h_c	convective heat transfer coefficient (W/m^2 K)
$h_{c,h}$	convective heat transfer coefficient of human body (W/m^2 K)
h_r	radiative heat transfer coefficient (W/m^2 K)
$h_{r,h}$	radiative heat transfer coefficient of human body (W/m^2 K)
h_{tot}	total heat transfer coefficient (W/m^2 K)
J_i	thermal radiosity of examined surface (W/m^2)
J_j	thermal radiosity of j surface (W/m^2)
L	length of one floor (m)
Q_{hr}	radiant heat transfer of heated wall (W)
Q_r	total radiative heat transfer (W)
T_a, T_{ceil}, T_c, T_f, T_h, T_i, T_{op}	temperature of air, ceiling, cold wall, floor, hot wall, inside air, and operative temperature (°C)W: width of the floor

Greek letters

α	thermal diffusivity (m^2/s^2)
β	thermal expansion coefficient (1/K)
γ	kinematic viscosity (m^2/s)
ε	emissivity

indicated that energy demand and consumption could considerably be affected by the choice of convective heat transfer coefficient algorithm and therefore proposed a new method for modelling the surface convection. Furthermore, Le Dreau and Heiselberg [3] detected that the convective heat transfer coefficient in rooms affected on the peak cooling load as well.

Nevertheless, at the beginning, convective heat transfer in buildings was taken into account via the equations which were derived with the assumption that convective heat transfer had a tendency to show similar behavior to free-edge insulated plates. While the number of experimental studies conducted in real-size chambers was increasing, it was understood that air flow at surrounding surfaces, although they were not heated or cooled, affected the flows on adjacent walls. Due to fact that air flows over all surfaces, it affects the flow pattern in the whole enclosure. Thus, correlations derived for free plates cannot be appropriately utilized for natural convection problems in enclosures [4]. The most extensive experimental studies conducted in enclosures that encompass the radiant wall issue are as follows:

In their work, which may be evaluated as the first experimental study for convective heat transfer coefficients in enclosures, Min et al. [5] investigated within the range of Rayleigh number 10^9–10^{11} and with enclosure dimensions 3.60 by 7.35 by 2.40 m, 3.60 by 7.35 by 3.60 m, and 3.60 by 3.60 by 2.40 m presented equations for convective heat transfer coefficients. These equations were proposed for non-ventilated conditions. Unheated surfaces were kept at constant temperature. They determined temperatures of the surfaces and heat fluxes given in the enclosure. They took into account radiation influences as well. Whereas the temperatures of the surfaces that were not heated varied between 4.4 °C and 21.1 °C, temperatures of the floor surfaces varied between 24 °C and 43.3 °C, and the temperature of the ceiling surface varied between 32.2 °C and 65.6 °C. One equation they suggested for walls is seen in Eq. (1).

$$h_c = 1.646\frac{(\Delta T)^{0.32}}{H^{0.05}} \tag{1}$$

Awbi and Hatton [6] investigated free convection in two different enclosures. The enclosures' sizes were 2.78 by 2.30 by 2.78 m and 1.05 by 1.01 by 1.05 m. One wall of the enclosures was used as a "heat sink" via an air conditioner located in a small room beside the large enclosure. Opposite and adjacent walls to the "heat sink" wall were heated with impregnated flexible sheets that had a 200 W/m^2 output. Thermocouples were placed inside and outside of the surfaces. They found the reference air temperature for the wall heating system to be 100 mm from the heated surface and referred to it as the "undisturbed air temperature" the temperature outside the thermal boundary layer, in other words. Consequently, they determined the temperature difference variant in the convective heat transfer coefficient and Nusselt number correlations as the temperature difference between the surface and the undisturbed air temperature. They found the thermal radiation utilizing the measured emissivity of the surfaces, which was derived from the total heat flux. Since they heated the walls partly as well, the characteristic length was calculated as the hydraulic diameter. They interpreted the outcomes presented, which stated that the convective heat transfer coefficient for a heated wall was lower than that found in the small enclosure that had approximately 1 m^3 volume. However, in order to evaluate if the difference was because of the heating plate sizes or enclosure sizes, they performed more experiments with small plates placed on surfaces. They found that there was a close agreement between convective heat transfer coefficients determined with whole wall heated experiments and small plates heated experiments. As a result, they stated that rather than the heated area on a wall, the size of the enclosure considerably influences convective heat transfer coefficients. They did a comparison of their results with those from the equations in open sources, and their data were determined to be in the central section of the curves. The correlation they have derived for heated walls is as follows:

$$h_c = \frac{1.823}{D^{0.121}}(\Delta T)^{0.293} \qquad \text{Heated wall } (2.78\,\text{m} \times 2.30\,\text{m} \times 2.78\,\text{m}) \tag{2}$$

Koca et al. [7] have conducted an experimental study to determine radiative, convective and total heat transfer coefficients in an experimental chamber. Their purpose was to evaluate the coefficients at different location configurations. Three different wall panel arrangements and seven water flow temperatures within the range of 30–42 °C were performed.

To calculate convective heat transfer coefficients for entire surfaces in an enclosure Khalifa and Marshall [8] have set up an experimental chamber that has dimensions parallel with actual size room of a building. 65 aluminum thermistors were placed in order to determine air and surface temperatures in the enclosure. Inner and outer surfaces of the chamber were covered with aluminum. Radiant heat exchanges were not counted in the calculation process. Additionally, an uncertainty analysis was conducted. Temperature measurements, conductivity of the materials and the lack of inclusion of long wave radiation in the low emissivity chamber have been taken into account during uncertainty analysis [4].

In their experimental study, Le Dreau et al. [9] examined convective heat transfer in active chilled beam and radiant wall cooling systems. The convective flow at the cooled wall was found difficult