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Mathematical modelling of conjugate heat transfer and fluid flow inside a domain with a radiant heating system

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ABSTRACT

This study deals with the numerical investigation of combined heat transfer by conduction, turbulent natural convection, and surface thermal radiation in a closed square air-filled cavity with a local radiant heater. The turbulent flow was computed within the quasi (pseudo) direct numerical simulation approach. The governing equations were solved by means of the finite difference method. Developed numerical code was validated by comparison of temperature profiles obtained experimentally and numerically. The effect of time, buoyancy force, walls emissivity, and emitter height on local and mean heat transfer characteristics was studied. For the first time it was found that the mean convective Nusselt number at the bottom solid-fluid interface was slightly altered in the cavity with the local radiant heater when varying the governing parameters. An increase in the Rayleigh number led to a significant rise in the overall temperature. The isotherms and streamlines were significantly altered with time. However, the mean radiative Nusselt number was slightly changed over the time.

1. Introduction

In order to provide a high thermal comfort of indoor environment for working personnel, various heating systems are used in industrial large-scale premises. Unit heaters, electrical air heaters, and radiant heating systems can be highlighted as the main. It was found [1] that the radiant panel heating, within the heat transfer by radiation is more than 50%, has the greatest potential for energy savings. Such energy supply systems are widely applied in Europe and East Asia [2,3] for heating not only industrial premises but residential buildings too. Radiant heating systems are more comfortable in terms of the smaller vertical temperature gradients, air movement, and dust transport in comparison with the unit and electrical air heaters [4]. Moreover, usage of radiant panels does not assume air heating in the entire domain. For example, only locally-located working areas are heated in large-scale industrial premises. This factor leads to a significant savings of energy resources [5]. Additionally, radiant heating systems can maintain the indoor temperature at 15^0C, while the same thermal comfort as in the case of the unit heaters [6] is achieved.

Investigations of thermal regimes of domains with radiant heating systems conducted both experimentally [7–10] and numerically [11–18]. Literature review showed that currently mathematical modelling results of conjugate turbulent heat transfer under conditions of the intensive energy supply from infrared emitters are poorly described. The closest articles related to this study are presented below.

Sh. Seyam et al. [7]. conducted both numerical and experimental research of hydrodynamics and heat transfer in the model room with a radiant heating system. In order to describe the fluid flow in the domain, three-dimensional model of turbulent natural convection was applied. It was found that an increase in the size of the radiant heater led to a decrease in the heat flux under the same emitter surface temperature. Emitter location affected the temperature distribution. However, the mean air temperature (35.5 °C) inside the room was approximately the same. It should be noted that the conjugate heat exchange was not taken into account when investigating thermal regimes.

M. Tye-Gingras and L. Gosselin [15] presented numerical results of thermal comfort analysis and energy consumption of radiant ceiling panels. Stationary equations of mass, momentum, and energy conservation were solved by means of the control volume method in the commercial software ANSYS Fluent. It was found that the thermal comfort was more sensitive to variation of radiant panel location in comparison with the energy consumption. It should be noted that finite thickness walls were not taken into account when conducting numerical research. Along with that a possibility of heat accumulation in enclosures was not considered.

M.D. Ahanj et al. [16]. presented both experimental and numerical results of combustion and radiation heat transfer under operating

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conditions of radiant tube heater. Complex heat transfer was simulated in the commercial software FLUENT6.2. According to the results of this study, it was found that heater efficiency significantly depended on the temperature of the inlet air to the burner.

Numerical studies [7,15] are devoted to low-temperature radiant panels applied for heating of residential buildings. On the other hand, it is well-known [19] that large-scale industrial premises with locally used working spaces have the greatest potential for energy savings. In order to study thermal regimes of these radiant systems, an approach [17,20] based on the solution of the Reynolds-averaged Navier–Stokes (RANS) equations for air and conduction equation for enclosures was suggested. However, investigations [17,20] did not take into account radiative heat transfer.

Based on the results of literature review, it was found that there is a significant lack in numerical modelling of heat transfer and fluid flow in the radiant systems. The real motivation of this research is to develop a mathematical model of conjugate heat transfer for thermal regimes analysis of domains heated by local radiant heaters.

2. Problem formulation

2.1. Geometrical and physical models

Generally, typical industrial premises have a shape of parallelepipeds. If the characteristic size of one of the coordinates is much greater than others, two-dimensional problem can be considered. In these cases, solution domain is presented as a rectangular cavity (Fig. 1) filled with air and bounded by heat-conducting walls. The radiant energy source was horizontally fixed under the top solid-fluid interface. The heat insulation conditions were set at the external boundaries of the solution domain. The equalities of heat fluxes and temperatures were set at the solid-fluid interfaces. We assumed that the radiant energy source had a constant surface temperature during the operating process. In addition, mounting a reflector to the back side of the infrared emitter was contributed to directional energy distribution. In this case, radiant fluxes were supplied only to the bottom and vertical solid-fluid interfaces.

Problem formulation assumed that the air was radiatively non-participating medium. We considered turbulent natural convective flow by means of the quasi (pseudo) direct numerical simulation method [21]. The thermophysical properties of the air, radiant heater, and solid walls were assumed as temperature-independent, since the temperature difference was relatively small. Air was viscous incompressible fluid satisfying Boussinesq approximation. The effects of energy viscous dissipation assumed neglectable small.

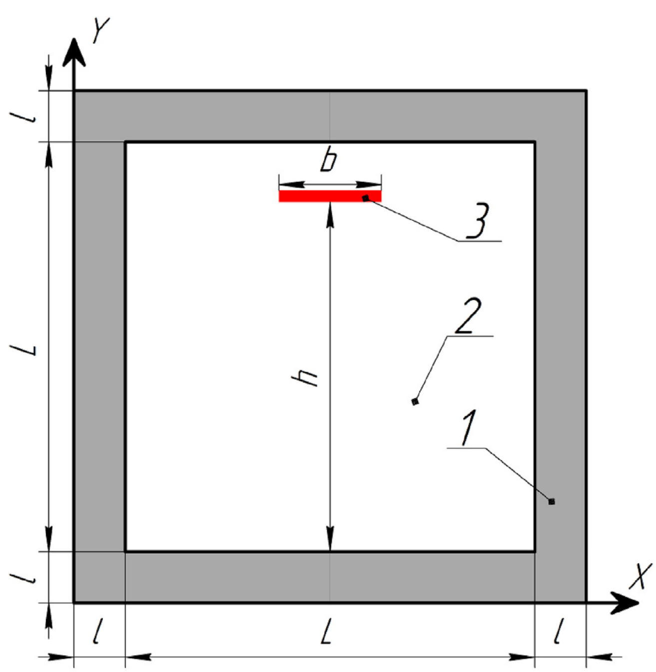

Fig. 1. Solution domain: 1 – heat-conducting walls; 2 – air; 3 – radiant energy source.

2.2. Mathematical model

Equations of mass, momentum, and energy conservation are presented in terms of the vorticity – stream function – temperature dimensionless variables. We used a length of the air cavity as the scale of the distance. In order to reduce the system of equations to the dimensionless form, the following relationships were used:

$$X = \frac{x}{L};\ Y = \frac{y}{L};$$

$$D = \frac{b}{L};\ J = \frac{q'}{\sigma\cdot T_h^4};\ Q_r = \frac{q''}{\sigma\cdot T_h^4};\ M = \frac{l}{L};\ N = \frac{h}{L};\ \tau = \frac{t}{t_0};\ U = \frac{u}{V_{nc}};\ V = \frac{v}{V_{nc}};\ \Theta = \frac{T - T_0}{T_h - T_0};\ \Psi = \frac{\psi}{\psi_0};\ \Omega = \frac{\omega}{\omega_0};$$

$$V_{nc} = \sqrt{g\cdot\beta\cdot(T_h - T_0)\cdot L};\ t_0 = L/\sqrt{g\cdot\beta\cdot(T_h - T_0)\cdot L};\ \psi_0 = V_{nc}L;\ \omega_0 = \frac{V_{nc}}{L}.$$

The stream function and vorticity are defined by the relations:

$$u = \frac{\partial\psi}{\partial y};\quad v = -\frac{\partial\psi}{\partial x};\quad \omega = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}.$$

Dimensionless conduction equation for enclosures is as follows:

$$\frac{\partial\Theta_1}{\partial\tau} = Fo_1\left(\frac{\partial^2\Theta_1}{\partial X^2} + \frac{\partial^2\Theta_1}{\partial Y^2}\right). \tag{1}$$

Dimensionless vorticity transfer, Poisson, and energy equations for air are as follows:

$$\frac{\partial\Omega}{\partial\tau} + U\frac{\partial\Omega}{\partial X} + V\frac{\partial\Omega}{\partial Y} = \sqrt{\frac{\Pr}{\mathrm{Ra}}}\cdot\left(\frac{\partial^2\Omega}{\partial X^2} + \frac{\partial^2\Omega}{\partial Y^2}\right) + \frac{\partial\Theta_1}{\partial X}, \tag{2}$$

$$\frac{\partial^2\Psi_2}{\partial X^2} + \frac{\partial^2\Psi_2}{\partial Y^2} = -\Omega, \tag{3}$$

$$\frac{\partial\Theta_1}{\partial\tau} + U\frac{\partial\Theta_1}{\partial X} + V\frac{\partial\Theta_1}{\partial Y} = \frac{1}{\sqrt{\mathrm{Ra}\cdot\Pr}}\cdot\left(\frac{\partial^2\Theta_1}{\partial X^2} + \frac{\partial^2\Theta_1}{\partial Y^2}\right), \tag{4}$$

Dimensionless conduction equation for radiant energy source is as follows:

$$\frac{\partial\Theta_3}{\partial\tau} = Fo_3\left(\frac{\partial^2\Theta_3}{\partial X^2} + \frac{\partial^2\Theta_3}{\partial Y^2}\right). \tag{5}$$

The initial conditions for equations (1)–(5) are as follows:

$$\Psi(X, Y, 0) = 0;\quad \Omega(X, Y, 0) = 0; \tag{6}$$

$$\Theta_1(X, Y, 0) = \Theta_2(X, Y, 0) = \Theta_3(X, Y, 0) = 0. \tag{7}$$

The boundary conditions for equations (1)–(5) are as follows:
at the surface of the radiant energy source:

$$\Theta_3 = 1. \tag{8}$$

at the external boundaries of the solution domain:

$$\frac{\partial\Theta_2(X, Y, \tau)}{\partial n} = 0. \tag{9}$$

at the solid-fluid interfaces:

$$\Psi = 0,\quad \frac{\partial\Psi}{\partial n} = 0,\quad \begin{cases}\Theta_i = \Theta_j,\\ \frac{\partial\Theta_i}{\partial n} = \frac{\lambda_j}{\lambda_i}\cdot\frac{\partial\Theta_j}{\partial n} + N_r\cdot Q_r,\end{cases}\quad \text{where}\ \left|\begin{array}{l} i = \overline{1,2} \\ j = \overline{1,2}\end{array}\right.. \tag{10}$$