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The dynamics of thermal regime changes of a local working zone in conditions of its heating by gas infrared radiators

A Nee
Tomsk Polytechnic University, Tomsk, Russia

E-mail: nee_alexander@mail.ru

Abstract. Mathematical modeling of unsteady heat transfer in a closed rectangular area with a local heat supply object in a conjugate formulation in working conditions of radiation source of energy is passed. Fields of temperatures and stream functions, illustrating the influence of a local typical object on thermal regime are received. The effect of Grashof number on dimensionless heat transfer coefficient - Nusselt number is investigated. The influence of nonconducted heat supply object on heat transfer rate in solution domain is showed.

1.Introduction

For modeling of temperature fields of local heat supply objects located in large industrial premises and heated by gas infrared radiators (GIR) was developed an approach [1], which is based on natural convection model in air-filled cavity with solid enclosing walls of finite thickness [2]. Heat sink to the enclosure structures and accumulation of heat in them was allowed at problem formulation [1], but was assumed that the radiation coming from GIR evenly distributed only on the lower horizontal boundary of the heating region. In addition, specific working area (heat supply object) was not considered as an obstruction for the movement from the lower boundary of the heated air in formulation problem [1].

It should be noted that the results of experimental researches analogically considered in [1] of conjugate convective – conductive heat transfer processes still haven't been published. Also the results of theoretical studies of fundamental mechanisms of heat conduction, convection and radiation processes jointly progressing with the work of gas infrared radiators haven't been published as well.

The aim of this study is the numerical simulation of unsteady heating process of a typical manufacturing object in the gas cavity of a closed rectangular area in conjugate formulation.

2.Problem formulation

The boundary value problem of convective – conductive heat transfer in a closed rectangular area consisting of six rectangular subdomains (Fig. 1) was solved. Heat insulation conditions were adopted at its external borders. At internal borders "air – enclosure structures" are the fourth type boundary conditions.

Assumptions that thermal properties of air and enclosure structures don't depend on temperature are introduced. Flow regime is laminar. The air is considered as a Newtonian fluid, incompressible, satisfies the Boussinesq approximation and absolutely transparent for thermal radiation. Total radiant flux coming from the GIR can be represented as the sum of heat fluxes values of which were determined by zonal method as shown in [3]. Investigated heat transfer process is described by the unsteady Navier - Stokes equations and the energy for the air and the heat conduction equation for enclosure structures within the adopted model. Dimensionless equations of Navier - Stokes and energy in variables "vorticity - stream function - temperature " are as follows [1, 2]:

$$\frac{\partial\Omega}{\partial\tau}+U\frac{\partial\Omega}{\partial X}+V\frac{\partial\Omega}{\partial Y}=\frac{1}{\sqrt{\mathrm{Gr}}}\cdot\nabla^2\Omega+\frac{1}{2}\frac{\partial\Theta_1}{\partial X}, \quad (1)$$

$$\nabla^2\Psi=-2\cdot\Omega, \quad (2)$$

$$\frac{\partial\Theta_1}{\partial\tau}+U\frac{\partial\Theta_1}{\partial X}+V\frac{\partial\Theta_1}{\partial Y}=\frac{1}{\mathrm{Pr}\sqrt{\mathrm{Gr}}}\cdot\nabla^2\Theta_1, \quad (3)$$

$$\frac{\partial\Theta_2}{\partial Fo_2}=\nabla^2\Theta_2, \quad (4)$$

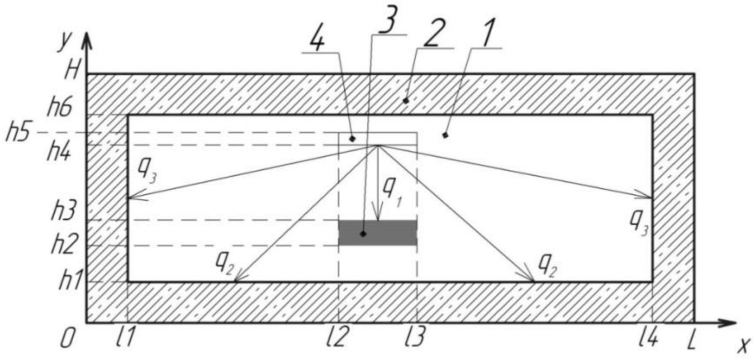

Figure 1. Solution domain: 1) air; 2) enclosure structures; 3) heat supply object; 4) gas infrared radiator (symbolic notation).

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

$$\Psi(X,Y,0)=0,\ \Omega(X,Y,0)=0,\ U(X,Y,0)=0,\ V(X,Y,0)=0,$$

$$\Theta_1(X,Y,0)=\Theta_2(X,Y,0)=0.$$

The boundary conditions at the outer boundaries of the solution domain are as follows:

$$X=0,\ X=1,\ 0<Y\le 1:\ \frac{\partial\Theta_2(X,Y,\tau)}{\partial X}=0,$$

$$Y=0,\ Y=1,\ 0<X\le 1:\ \frac{\partial\Theta_2(X,Y,\tau)}{\partial Y}=0.$$

at internal interfaces “heat supply object – air”, “solid wall – air”, parallel to the axis OX:

$$\Psi=0,\ \frac{\partial\Psi}{\partial Y}=0,\ \begin{cases}\Theta_i=\Theta_j,\\ \dfrac{\partial\Theta_i}{\partial Y}=\dfrac{\lambda_j}{\lambda_i}\cdot\dfrac{\partial\Theta_i}{\partial Y}+Ki_{q_k},\end{cases}\ \text{где}\ \left|\begin{array}{l}i=\overline{1,3}\\ j=\overline{1,3}\\ k=1,2.\end{array}\right.$$

at internal interfaces “heat supply object – air”, "solid wall - air", parallel to the axis OY:

$$\Psi=0,\ \frac{\partial\Psi}{\partial X}=0,\ \begin{cases}\Theta_i=\Theta_j,\\ \dfrac{\partial\Theta_i}{\partial X}=\dfrac{\lambda_j}{\lambda_i}\cdot\dfrac{\partial\Theta_i}{\partial X}+Ki_{q_3},\end{cases}\ \text{где}\ \left|\begin{array}{l}i=\overline{1,3}\\ j=\overline{1,3}\end{array}\right.$$

where $Fo-$ Fourier number; $Gr-$ Grashof number; $Ki-$, Kirpichev number; $\mathrm{Pr}-$Prandtl number; X,Y – dimensionless Cartesian coordinates; U, V– dimensionless velocities along X,Y directions

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correspondingly; $\nabla -$ dimensionless Laplace operator. Indexes: 1,2,3 - design elements; for Ki: - heat flow values. Indexes: 1, 2, 3 – element number; for Ki: q_1, q_2, q_3 – heat flux values.

Equations (1)–(4) with the corresponding initial and boundary conditions were solved by applying the finite difference method, as in [4]. Locally one scheme of A.A. Samarskiy was used for approximation of equations (1), (3) – (4) [5]. Approximation of Poisson's equation was done by the scheme of variable directions [5]. Woods condition [5] was used to determine the boundary conditions for the vortex velocity. One - dimensional difference analogues were solved by the sweep method [5]. To evaluate the reliability of results of the computational modeling, the conservatism of the difference scheme was checked analogously to [6].

3.The results of the numerical simulation

Numerical investigation was performed for the following values of dimensionless criteria: $Gr=10^7$, $Pr=0,71$, $Ki_1=42$, $Ki_2=25, Ki_3=5$. The results of solving boundary value problem for three possible variants of the heated air flow are shown in Figures 2, 3 and 4.

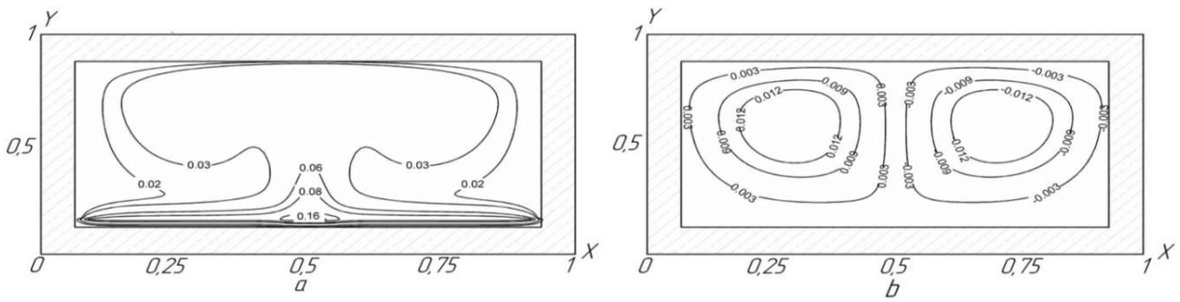

Figure 2. Fields of temperature (a) and the stream function (b) (τ = 3600) in the absence of an heat supply object in the gas cavity.

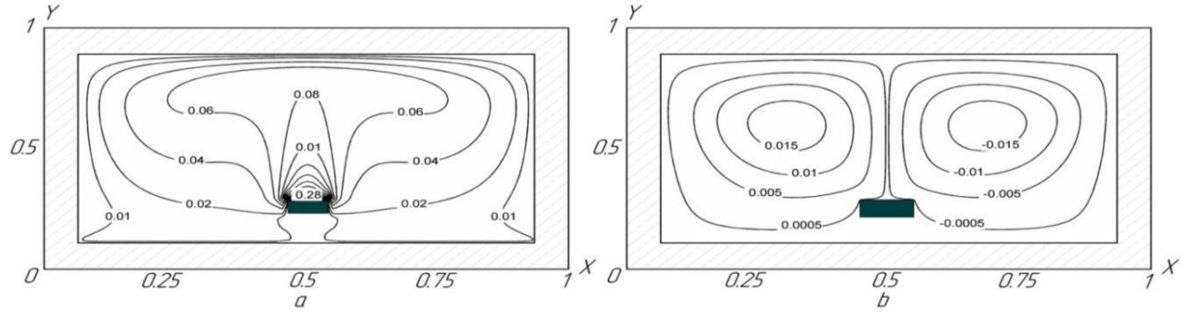

Figure 3. Fields of temperature (a) and the stream function (b) (= 3600) in the presence of an nonconducted heat supply object in the gas cavity (symmetric variant).

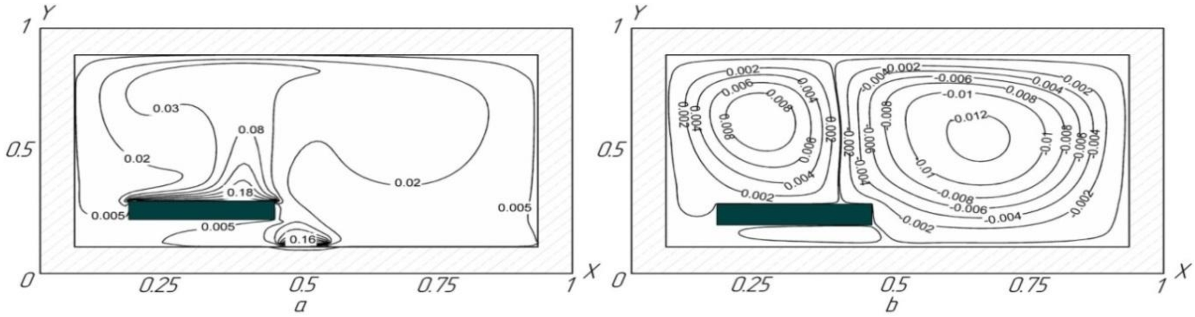

Figure 4. Fields of temperature (a) and the stream function (b) (= 3600) in the presence of an nonconducted heat supply object in the gas cavity (asymmetric variant).

Comparing the results of numerical simulations, it can be concluded that the nonconducted object in the gas cavity significantly effects the character of heat transfer in this solution domain. If the heat supply object is located under GIR (Fig. 3) and coincides with it in size, a significant proportion of the energy comes to the border in y= h3, l2 <x<l3, what is well illustrated by the position of the isotherms. Heated air flows around the object and rises in the result of natural convection then cools down due to heat sink to the enclosure structures and descends along the vertical walls. As the result a symmetrical circulation flow is formed relatively to the section X = 0.5, which clearly demonstrates the field of the stream function (Fig. 3 b). Movement along the stream lines with the sign "-" is oriented clockwise, with the "+" counterclockwise.

For the purpose of analyzing the possible asymmetric flow, the variant with offset of nonconducted object larger lateral dimension to the left is considered (Fig. 4). In this case, an intense air heating occurs at the border = h1, l2 <x <l3. In the section X = 0.39 of the solution domain above the heat supply object an ascending heated air is formed. Besides the two main vortices additional one (under the object) is formed.

By analogy with [7, 8] the influence of the Grashof number at the average Nusselt number was analyzed. At the interface "enclosure structures - air» (y = h1, l1 <x <l4) average dimensionless heat transfer coefficient is defined [8] as:

$$\overline{Nu} = \frac{1}{L}\int_{0,12}^{0,88}\left(\frac{\partial \Theta}{\partial Y}\right)_{Y=0,12} dX.$$

Dependences $\overline{Nu}$ from Gr are presented in Figure 5.

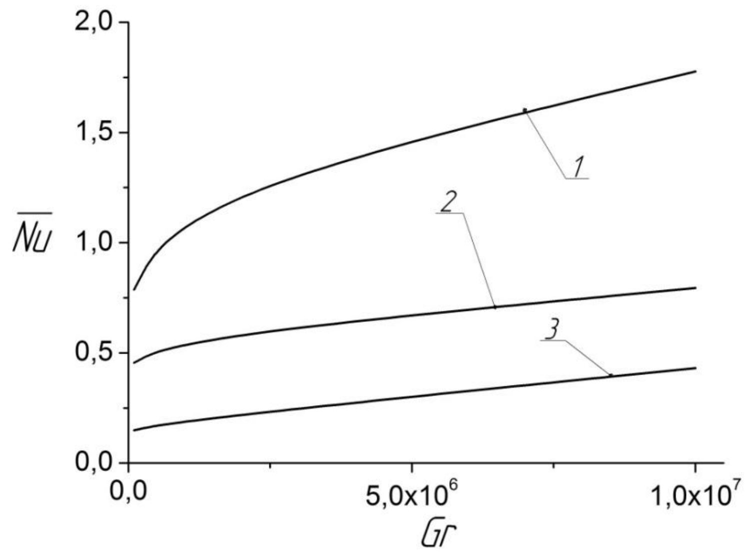

Figure 5. Dependences from Gr (= 3600): 1 – heat supply object is absent; 2 – nonconducted object in the gas cavity (asymmetric variant); 3 – nonconducted object in the gas cavity (symmetric variant).

Figure 5 shows that is increased with rising Gr. Also nonconducted object in the gas cavity (and its location) significantly effects the intensity of convective heat transfer. If the heat supply object is absent, the average dimensionless heat transfer coefficient in section Y = 0,12 of the solution domain 4.5 times greater compared with the case of its presence. It can be concluded that with placement of the GIR the location of the workplace in order to optimize heat transfer conditions must be allowed.

4.Conclusion

Numerical simulation results show that the presence of the heat supply objects in the gas cavity significantly changes the temperature field and the intensity of the air movement in the working area of industrial premises.

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