EPJ Web of Conferences **76**, 01006 (2014)
DOI: 10.1051/epjconf/20147601006

Influence of heatsink from upper boundary on the industrial premises thermal conditions at gas infrared emitter operation

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Abstract. The results of mathematical simulation of the heat transfer processes in the closed domain, which corresponds to production accomodation with the gas infrared emitter operation condition are presented. The system of differential Navier-Stokes equations in the approximation of Boussinesq is solved. The comparative analysis of thermal conditions formation in the closed domain is carried out taking into account heat withdrawal through the upper enclosing construction and under the conditions of its heat insulation. The essential transiency of the analyzed heat transfer process and the influence of heat withdrawal from one of the outer boundaries on the mean temperatures values in large-dimension industrial premises are established.

The distinguishing feature of modern industry becomes the rigid savings of energy consumption, as a rule, for the purpose of reduction in the economic expenditures. Localization of systems and heat supply sources is one of the ways for reduction in total energy expenditures for the heating of large production accomodations (local heating of separate workplaces). As the most acceptable version the system of gas infrared emitters (GIE) [1] can be used. Scale GIE introduction in the production is strongly limited because of the insufficient experimental and theoretical study of convective heat transfer in the large-dimension accomodations with the gas infrared emitters operation conditions.

For the mathematical simulation of the studied process the process description of free convection in air region and thermal conductivity in enclosing constructions (Fig. 1) is necessary. In the problem statement system conjugate heat transfer equations [2–5] was used as a base. All energy from emitter came to bottom medium interface H_{H}.

To analyze the significant factors impact on the thermal modes in the manufacturing conditions of GIE operation two versions of the heat transfer problem statement were considered. The first version: heat insulation conditions are satisfied on all outer boundaries of solution region. The second: on upper boundary of solution region $y = H$ (Fig. 1) heat exchange condition with environment is satisfied. The problem was solved in the dimensionless formulation.

The system of equations describing the heat transfer in the system has the form:

$$\frac{1}{\mathrm{Sh}}\frac{\partial\Omega}{\partial\tau} + U\frac{\partial\Omega}{\partial X} + V\frac{\partial\Omega}{\partial Y} = \frac{\partial^2}{\partial X^2}\left[\left(\frac{1}{\sqrt{\mathrm{Gr}}}\right)\Omega\right] + \frac{\partial^2}{\partial Y^2}\left[\left(\frac{1}{\sqrt{\mathrm{Gr}}}\right)\Omega\right] + \frac{1}{2}\frac{\partial\Theta}{\partial X}, \quad (1)$$

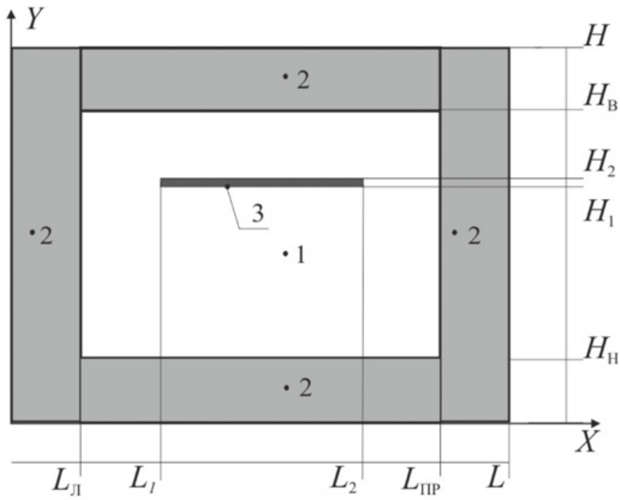

Figure 1. Solution area: 1 – air; 2 – walling construction; 3 – infrared emitter.

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

$$\frac{1}{\mathrm{Sh}}\frac{\partial\Theta}{\partial\tau}+U\frac{\partial\Theta}{\partial X}+V\frac{\partial\Theta}{\partial Y}=\frac{\partial}{\partial X}\left[\left(\frac{1}{\mathrm{Pr}\sqrt{\mathrm{Gr}}}\right)\Theta\right]+\frac{\partial}{\partial X}\left[\left(\frac{1}{\mathrm{Pr}\sqrt{\mathrm{Gr}}}\right)\Theta\right], \tag{3}$$

$$\frac{1}{\mathrm{Fo}}\frac{\partial\Theta}{\partial\tau}=\frac{\partial^2\Theta}{\partial X^2}+\frac{\partial^2\Theta}{\partial Y^2}\,. \tag{4}$$

The initial conditions for the system of equations:

$$\Psi(X,Y,0)=\Omega(X,Y,0)=0, \Theta(X,Y,0)=0. \tag{5}$$

The boundary conditions at the external borders of the field solutions:

$$\frac{\partial\Theta(X,Y,\tau)}{\partial n}=0, \tag{6}$$

The conditions at the boundaries of the solid walls and gas:

$$\frac{\partial\Psi(X,Y,\tau)}{\partial n}=0,\quad \frac{\partial\Theta_1(X,Y,\tau)}{\partial n}=\lambda_{1,2}\frac{\partial\Theta_2(X,Y,\tau)}{\partial n},$$
$$\partial\Theta_1(X,Y,\tau)=\partial\Theta_2(X,Y,\tau). \tag{7}$$

The radiative heat source is accounted on the border of the H_H:

$$\frac{\partial\Psi(X,Y,\tau)}{\partial n}=0,\quad \frac{\partial\Theta_1(X,Y,\tau)}{\partial n}=\lambda_{1,2}\frac{\partial\Theta_2(X,Y,\tau)}{\partial n}+Ki,$$
$$\partial\Theta_1(X,Y,\tau)=\partial\Theta_2(X,Y,\tau). \tag{8}$$

The boundary conditions of the first kind on the surface of the heater:

$$\Theta(X,Y,\tau)=1. \tag{9}$$

where X, Y – dimensionless Cartesian coordinates; Θ – dimensionless temperature; Ω – dimensionless analog vorticity; $\mathrm{Fo}=at_0/L^2$ – Fourier number; $\mathrm{Gr}=g\beta L^3(T_{it}-T_0)/\nu_t^2$ – Grashof number;

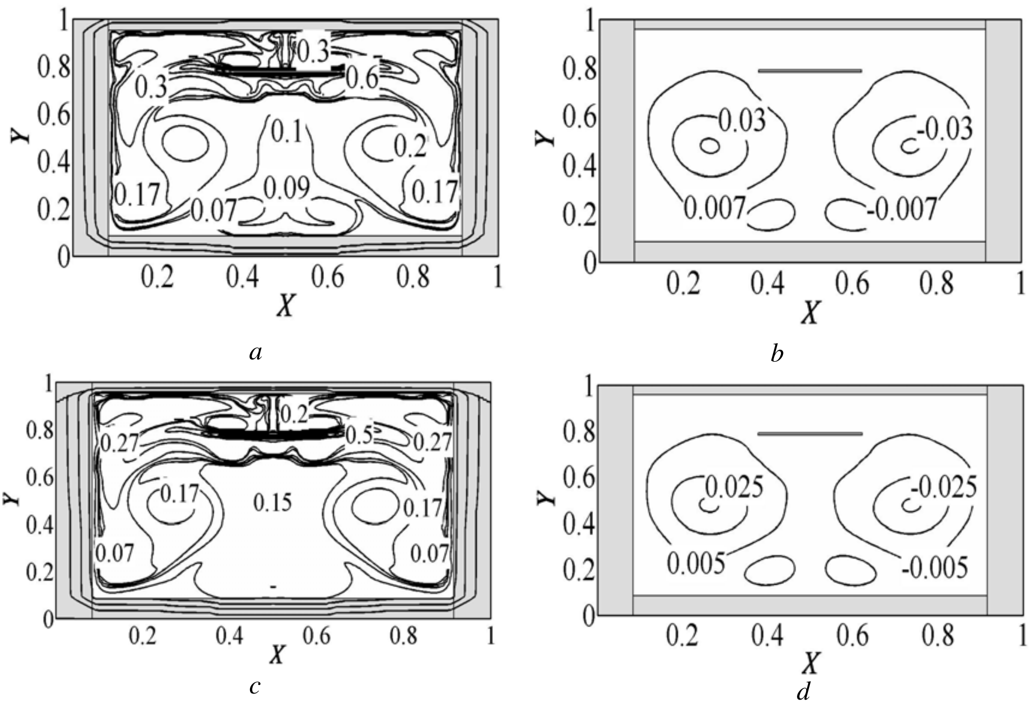

Figure 2. Temperature field (a, c) and isolines of the stream function (b, d) in terms of upper boundary (a, b) thermal insulation and heat sink at the upper boundary (c, d) for time moment $t = 60\,000$.

g – acceleration of gravity, m/s^2; a – thermal diffusivity, m^2/s; β – thermal expansion coefficient, K^{-1}; $\mathrm{Bi} = \alpha L/\lambda$ – Biot number; α – coefficient of heat exchange between the external environment and the area under consideration solutions; $\mathrm{Pr} = \nu_t/a$ – Prandtl number; T_0 – temperature of the gas and solid at the initial time, K; U, V – dimensionless velocity; λ – solid wall thermal conductivity, W/(m·K); $\lambda_{1,2}$ – relative thermal conductivity; ν_t – dynamic molar ratio (turbulent) Viscosity, m^2 / s; τ – dimensionless time; n – vector normal to the surface; Ψ – dimensionless analog current function.

In the second variant, the boundary condition (6) when $y = H$ has the form:

$$\frac{\partial\Theta\,(X, Y, \tau)}{\partial Y} = \mathrm{Bi}\cdot\Theta\,(X, Y, \tau) + \mathrm{Bi}\cdot(\Theta_e)\,, \quad Y = \frac{H}{L},\ \ 0 < x < 1,\ \ 0 < \tau < \frac{t}{t_0}, \tag{10}$$

where Θ_e – dimensionless environment temperature.

In solving the problem (1)–(10) the algorithm [4, 5] developed for the numerical solution of natural convection in closed rectangular regions with local energy sources was used. Accounting for the turbulence effects produced in the approximation of algebraic Prandtl model:

$$\nu_t = l_m^2\left|\frac{\partial\bar{\upsilon}_x}{\partial y}\right|, \quad l_m = k\cdot y,$$

where l_m – the mixing way; x,y – coordinates, m; υ – velocity component, m^2 / s; k – the universal constant of proportionality is independent of the Reynolds number.

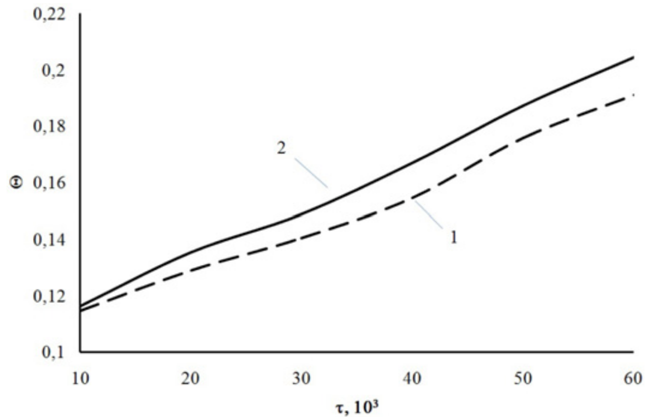

Figure 3. Average air temperature values of the air inside the field solutions for the variant with the upper limit of the heat sink (1) and its insulation (2) for the time t = 60 000.

To accomplish the task the following values of the dimensionless temperature were used for the heater – $\Theta_{it} = 1$, the initial – $\Theta = 0$, for the environment – $\Theta_e = -0.1$.

Figure 2 shows typical results of numerical conjugate heat transfer modeling for two options: taking into account the heat sink from the upper boundary $y = H$ of solution field and insulation provided on this border.

Temperature distributions (Fig. 2) in considered solution area illustrate the intensive circulation of air near the heater surface and bottom enclosing structure heated by a heat flow from GIE. It is important to note that the temperature reaches the maximum values near the heater surface for the two variants of the problem statement.

Due to the heat sink from the top surface of the reinforced concrete slab the air temperature near it is much lower than in the case of upper border thermal insulation. The obtained mean air temperatures values (Fig. 3) in the closed domain indicate a significant heat outflow to the environment (Fig. 2).

Analysis of the data allows to make conclusions about the necessity of such real processes as heat sink from the outer surface walling in determining the thermal conditions in industrial premises when gas infrared radiators are working. This will make possible to select the optimum GIE parameters of and reduce power consumption.

Work performed under the research state assignment «Science» (Code of Federal Target Scientific and Technical Program 2.1321.2014).

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