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Numerical simulation of heat transfer in a large room with a working gas infrared emitter

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Abstract. The article presents a new approach to determining the main characteristics of the thermal regime in the buildings and structures heated by the gas infrared emitters, based on the analysis of convective heat transfer in the air and thermal conductivity in the enclosing structures. The authors have considered a mathematical model of turbulent heat transfer in the framework of the standard k-ε model under radiant heating conditions. The simulation was carried out for a large room with an approximation to the real operating conditions of an industrial facility. The options for heating an empty room and the room with an object were considered.

1. Introduction

In the modern world, much attention is paid to reducing energy consumption, most of which falls on the industrial facilities (both the production process itself and the heating of the industrial complexes that occupy large areas) [1-3]. The consumption of energy resources is often uneconomical and therefore several regulations on energy efficiency have been adopted recently [1]. Attention is paid to the measures aimed at reducing costs of radiant heating systems for residential and, to a greater extent, industrial premises, because the efficiency of such heating systems in comparison with standard air and water heating systems is confirmed constantly [4]. The assessment of radiant heating and cooling systems for floors and wall panels is carried out in sufficient detail in the literature [5-7]; however, a less significant role is assigned to the gas infrared emitters (GIE). Nevertheless, they can become an economical alternative for heating small local areas in the industrial premises [8-12]. In most cases, a heated room contains equipment with different geometric characteristics, which can and most likely affects the thermal conditions of the local working areas. To date, no assessment has been made of this impact. The present work aims to conduct mathematical modeling of heat transfer processes in a large-sized production room heated by a gas infrared emitter.

2. Problem statement and solution methods

When formulating the problem, a rectangular area was considered with dimensions 4.5 × 5 m, shown in Figure 1. The simulation was carried out for a large room with an approximation to the real operating conditions of an industrial facility: heat dissipation into the enclosing heat-conducting walls 10 cm thick was taken into account. The dimensions of the emitter were 10 × 40 cm. Studies were carried out with a pinewood table 1.2 m wide, 0.040 m thick The heated surface of the table was located at a height of 0.755 m from the floor, and the middle of the tabletop, like the middle of the GIE, were located at a distance of 1.600 from the left wall. The distance from the floor to the lower edge of the GIE was 2.975 m, and the center of the emitter was situated at a distance X = 1.6 m.

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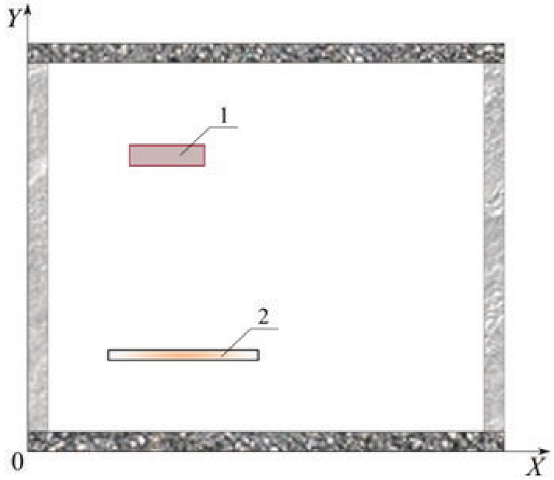

Figure 1. Study area: 1 – GIE, 2 – wood table.

It was assumed that heat transfer from the GIE was carried out through all three mechanisms: heat conduction, convection, and radiation. Convective-conductive heat transfer was simulated by the equation:

$$\rho c_p \frac{\partial T}{\partial t} + \rho c_p \left(\vec{u} \cdot \nabla\right) T = \nabla \cdot \left(\kappa \nabla T\right), \tag{1}$$

where t, ρ, T, c_P, k – are, respectively, the time, temperature-dependent density, absolute temperature, specific isobaric heat capacity, and thermal conductivity. The vector velocities field was described by a system of motion equations of an incompressible gas in the Boussinesq approximation:

$$\rho \frac{\partial \vec{u}}{\partial t} + \rho\left(\vec{u} \cdot \nabla\right)\vec{u} = \nabla \cdot \left[-p\vec{I} + \vec{K}\right] + \left(\rho - \rho_0\right)\vec{g}\ , \tag{2}$$

$$\rho \nabla \cdot \left(\vec{u}\right) = 0\ , \tag{3}$$

where $p, \vec{I}$ is the pressure and unit tensor; $\rho_0, \vec{g}$ are the initial density and gravitational acceleration; $\vec{K} = \left(\mu + \mu_T\right)\left(\nabla \cdot \vec{u} + \left(\nabla \cdot \vec{u}\right)^T\right)$ is the stress tensor of viscous friction which takes into account the turbulent (index «T») component, μ is the dynamic viscosity coefficient, $\mu_T = \rho C_\mu k^2 / \varepsilon$ is the turbulent viscosity coefficient.

When simulating turbulence, a k-ε model was used, in which the turbulence kinetic energy (k) and the turbulence dissipation rate (ε) were determined by the equations:

$$\rho \frac{\partial k}{\partial t} + \rho\left(\vec{u} \cdot \nabla\right)k = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_k}\right)\left(\nabla \cdot k\right)\right] + P_k - \rho\varepsilon\ , \tag{4}$$

$$\rho \frac{\partial \varepsilon}{\partial t} + \rho\left(\vec{u} \cdot \nabla\right)\varepsilon = \nabla \cdot \left[\left(\mu + \frac{\mu_T}{\sigma_\varepsilon}\right)\left(\nabla \cdot k\right)\varepsilon\right] + C_{\varepsilon 1}\frac{\varepsilon}{k}P_k + C_{\varepsilon 2}\rho\frac{\varepsilon^2}{k}\ . \tag{5}$$

In equations (4) - (5), the ratio $P_k = \mu_T\left[\nabla \cdot \vec{u} : \left(\nabla \cdot \vec{u} + \left(\nabla \cdot \vec{u}\right)^T\right)\right]$ as well as the following constants were used: $C_\mu = 0.09$; $C_{\varepsilon 1} = 1.44$; $C_{\varepsilon 2} = 1.92$; $\sigma_\varepsilon = 1.3$; and $\sigma_\kappa = 1$.

The system of equations (2) - (5) determines the atmospheric air velocity flowing into the room, while all other elements of the concerned area in Figure 1 were considered solid, thus flow rates in them were assumed to be zero. In this case, equation (1) described the heat fluxes taking place only by thermal conductivity.

When simulating heat transfer by radiation, the air was assumed to be a diathermic medium, and the surfaces of walls, floor, ceiling, GIE, and table were opaque gray. The determination of heat fluxes corresponded to the zone model and was implemented by direct integration of fluxes between all surfaces («Surface-to-Surface Radiation») of a closed system of surfaces taking into account the mean angular coefficients determined within this system.

As the initial conditions, we used constant values of the main thermodynamic parameters (T_0, P_0, ρ_0) and zero values of the velocity components throughout the entire region. The exception was the downward-facing working surface of the GIE, on which the working temperature of the radiating surface T_{GIE}, constant for the entire simulation time, was set.

The conditions of adiabaticity on the outer surfaces of the simulation domain were used as the boundary conditions for equation (1), since it was believed that, with a limited simulation time, the enclosing structures did not have time to warm up over the entire thickness. As already mentioned above, on the working surface of the GIE, the boundary condition of the first kind was set to maintain a constant temperature T_{GIE}. For all other surfaces, boundary conditions were not required, since the heat transfer over all components of the simulation domain was described by the same equation (1).

The system of equations (1-5) was solved using the finite element method implemented in the COMSOL Multiphysics software system with $k - \varepsilon$, Turbulent Flow, Heat Transfer in Fluids, and Surface-to-Surface Radiation modules.

3. The results

Figure 2 shows the results of numerical simulation of convective-conductive heat transfer in the room under consideration of a heating system based on the GIE for the time instant τ corresponding to 1 hour of physical time.

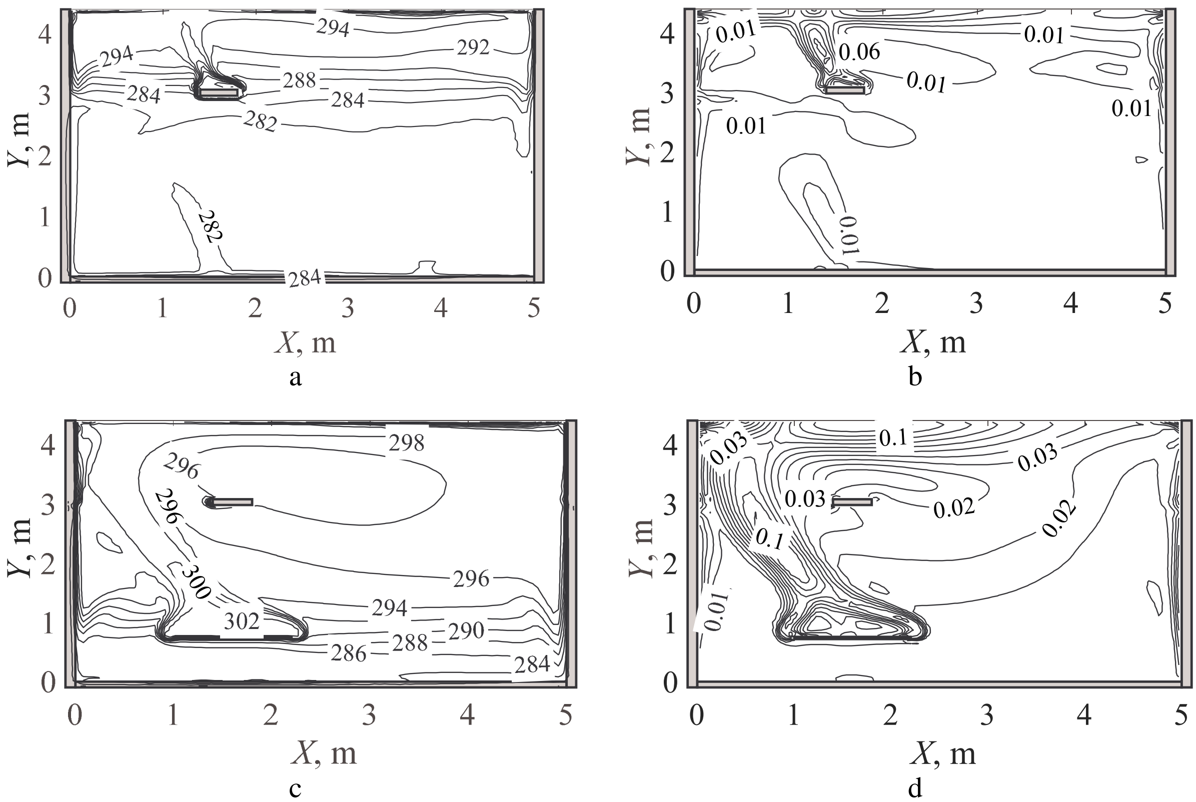

Figure 2. Temperature fields (a, c) and current function isolines (b, d).

For small-time values, on the T distribution curves, one can distinguish a zone of high temperatures near the radiator oriented in the X direction (Figure 2a). It can be seen that in the considered area, two separate circulation zones are formed: under the GIE and above it (Figure 2 b).

The presence of a table in the investigated area significantly affects the thermal regime in the room (Figure 2 c, d). The surface of the table intensifies the heating of the air near it, which leads to intensive movement of air masses. Warm air rises to the upper area and moves down along the walls to the floor. A stable circulating air current is formed around the GIE.

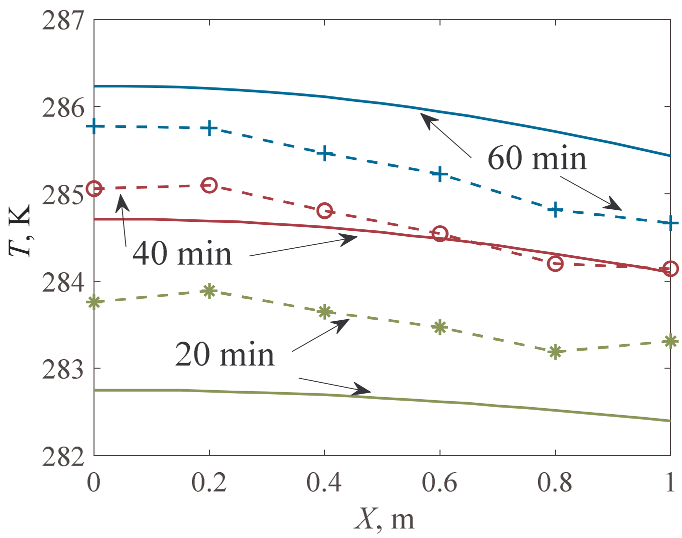

Figure 3. Distributions of the floor surface temperature in the *X* direction obtained experimentally (dashed line) and as a result of numerical simulation (a solid line) at different times: 1 – 20 min, 2 – 40 min, 3 – 60 min.

Comparing the temperatures of the floor surface obtained as a result of a numerical simulation with those obtained experimentally (Figure 3), one can see their good correspondence in the entire range of time variation.

4. Conclusion

Based on the results of the theoretical studies of the heat transfer processes in the large-sized industrial premises with the radiant heating systems using the gas infrared emitters, it can be concluded that it is possible to describe the temperature fields of such an object by a mathematical model based on the Navier-Stokes and energy equations for air, and heat conduction equations for the enclosing structures. The obtained distributions of the main characteristics of the process under study (temperature and velocity) illustrate the leading role of thermogravitational convection processes in the establishment of the thermal regime of objects with the radiant sources of heating, as well as the significant effect of the equipment located in a heated room on the structure of the convective flows and, accordingly, on the thermal regime of a local workplace.

The regularities and characteristics of the studied heat transfer processes in the local working zones of the large-sized industrial premises heated by the radiant heating systems revealed in the experiments can be used to construct the models of such processes and develop significantly the theory of the microclimate formation processes in the industrial premises.

Acknowledgments

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