Practical application of natural convective heat exchange of electrical equipment with air

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Abstract. This paper provides the physical and mathematical model of the air flow in a volume, containing electrical equipment with the heat-generating and heat-absorbing surfaces. The model predicts the temperature fields and air flow velocities across the volume. Using the developed model, we calculate the values of heat fluxes in the vicinity of the thermostated electrical equipment for three different cases: natural convective, forced and mixed modes of the airflow. The possibility of beneficial use of natural convective air flows for the transfer of thermal energy is analyzed. The results are applied in an industrial enterprise. Energy consumption for ventilation is significantly reduced.

1. Introduction
Electrical equipment in most of the cases is cooled by the heat exchange with ambient air [1]. Usually, the approach determining the power of the heat load of the electrical equipment does not consider the influence of the natural convective air flows. The velocity distribution of the airflows near the generating sets and the heat flux values on their surface allow assessing the effect of the natural convection.

Modern methods of the mathematical modeling of air motion provide the values of heat fluxes with high accuracy. The subsequent analysis of the natural convective air motion predicts the effect of convection on the heat transfer intensity. In this paper we theoretically investigate the cooling system of an isotope separation plant as a model case of industrial production with high level of electrical equipment [1]. The aim of the research is the ventilation parameters, which ensures the thermostating conditions of the electrical equipment in the building.

2. Problem formulation
The electrical equipment is placed in the limited area in groups of two in five tiers (Fig. 1). The area is a rectangular parallelepiped, which is usually an element of the periodic equipment arrangement. The height of the region is 7.3 m, the depth is 0.2 m, and the width is 1.5 m. Inside this sample area, we investigate the turbulent natural convective flow of the viscous heat-conducting air. The geometric dimensions of the equipment are stylized with the surface of an average generating set (Fig. 2). The surface of each generating set is divided into sections. The temperatures of the sections are constant, but differ. For the sections of the electric motor winding and the cooling jacket, the temperature is set to the characteristic values.

The heat exchange of air with the walls of electrical equipment is investigated during forced and natural convective air flow in the vicinity of the equipment. The mathematical formulation of the problem takes into account the processes of turbulence using a two-parameter k-ε turbulence model. This turbulence...
The presented calculations are estimates. The main objective of the presented results is to substantiate a new ventilation scheme for a production facility with electrical equipment.

3. Mathematical model

The stationary natural-convective flow of the viscous heat-conducting air is described by three-dimensional Navier–Stokes equations written for the incompressible gas taking into account turbulence [3]. We use the k-ε turbulence model to describe the turbulent nature of the airflow. The equation system has the following form:

\[ \nabla \cdot (\rho \vec{V}) = 0, \]  
\[ \nabla \cdot (\rho \vec{V} \vec{V}) + \nabla p = \nabla \cdot \tau_{\text{eff}} - \rho \vec{g} \left( \rho - \rho_0 \right), \]  
\[ \nabla \cdot (\rho H \vec{V}) = \nabla \cdot (\lambda_{\text{eff}} \nabla T) + \nabla \cdot (\tau_{\text{eff}} \cdot \vec{V}) - \rho \vec{g} \left( \rho - \rho_0 \right) \vec{V}, \]  
\[ \nabla \cdot (\rho k \vec{V}) = \nabla \cdot \left[ \left( \mu + \frac{\mu_{\text{turb}}}{\sigma_k} \right) \nabla k \right] + G_k + G_p - \rho \dot{\varepsilon}, \]  
\[ \nabla \cdot (\rho \varepsilon \vec{V}) = \nabla \cdot \left[ \left( \mu + \frac{\mu_{\text{turb}}}{\sigma_\varepsilon} \right) \nabla \varepsilon \right] + C_{1\varepsilon} \frac{\varepsilon}{k} \left( G_k + C_{3\varepsilon} G_p \right) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k}, \]  
\[ p = \rho \frac{R}{M} T, \]

where: \( \vec{V} \) is the gas velocity vector, \( \text{m/s} \); \( H \) is the total enthalpy, \( H = c_p T + \frac{V^2}{2}, \text{J/kg} \); \( E \) is the total energy, \( E = c_v T + \frac{V^2}{2}, \text{J/kg} \); \( k \) is the kinetic energy of the turbulent pulsation, \( \text{m}^2/\text{s}^2 \); \( \varepsilon \) is the dissipation rate of the turbulent kinetic energy, \( \text{m}^2/\text{s}^3 \); \( \tau_{\text{eff}} \) is the effective stress tensor, \( \tau_{\text{turb}} = \tau + \tau_{\text{turb}} \), \( \text{Pa} \); \( \tau_{\text{turb}} \) is the turbulent stress tensor, \( \tau_{\text{turb}} = \mu_{\text{turb}} \left( \nabla \vec{V} + (\nabla \vec{V})^T \right) - \frac{2}{3} \rho k \vec{U} \), \( \text{Pa} \); \( \tau \) is the viscous stress tensor,
\( \tau = \mu \left( \nabla V + \nabla V^T \right) \), Pa; \( \mu_{\text{turb}} \) is the turbulent viscosity, \( \mu_{\text{turb}} = \rho C_\mu \frac{k^2}{\varepsilon} \); \( \lambda_{\text{eff}} \) is the effective heat conductivity coefficient, \( \lambda_{\text{eff}} = \lambda + \lambda_{\text{turb}} \), W/(m·K); \( \lambda \) is the thermal conductivity, W/(m·K); \( \lambda_{\text{turb}} \) is the turbulent thermal conductivity, \( \lambda_{\text{turb}} = \frac{c_p \mu_{\text{turb}}}{\Pr_{\text{turb}}} \), W/(m·K); \( G_k \) is the turbulent kinetic energy generation due to the shearing flow (velocity gradient effect), \( G_k = 2 \mu_{\text{turb}} S : S \), Pa/s; \( S \) is the strain tensor, \( S = \frac{1}{2} (\nabla V + \nabla V^T) \); \( G_b \) is the turbulent kinetic energy generation due to the natural convection (Archimedes force), \( G_b = g \frac{\mu_{\text{turb}} V \rho}{\Pr_{\text{turb}} \rho} ; M \) is the air molecular weight, kg/mol.

The following boundary conditions are set for the equation system (1)–(6): no-slip conditions \( V(x,y,z,t) = 0 \) and conditions for the constant temperature \( T = T_{i,j} \) on the walls of the equipment, where \( i \) corresponds to the number of the section with a constant surface temperature of the generating set, and \( j \) corresponds to the number of the tier. The turbulent parameters of the flow on the wall of the equipment and the near-wall region are determined as follows: \( \frac{\partial \lambda}{\partial n} = 0 \), \( \varepsilon_p = \frac{c_{p3/4} k_{p3/2}}{\kappa y_p} \).

The expression for the rate of the turbulent energy dissipation in a cell near the wall of the equipment is obtained from the expressions for the shear stress on the wall \( \tau_w = \mu_{\text{turb}} \frac{\partial V}{\partial n} \), the turbulent viscosity \( \mu_{\text{turb}} = \rho C_\mu \frac{k^2}{\varepsilon} \) and the logarithmic law of the velocity distribution in the turbulent boundary layer

\[
U^* = \frac{1}{\kappa} \ln \left( E \cdot \tilde{y}^* \right), \quad \text{where} \quad U^* = \frac{C_{\mu}^{1/4} k_{p}^{1/2} U_p}{\tau_w / \rho}, \quad y^* = \frac{\rho C_{\mu}^{1/4} k_{p}^{1/2} y_p}{\mu}, \quad \tilde{y}^* = \max \left( y^*, y^{*\text{lim}} \right), \quad y^{*\text{lim}} = 11.225.
\]

Here \( k_p = 0.4187 \) is the Karman's constant; \( E = 9.793 \) is the empirical constant. The value of \( y^* \) is determined from the equation:

\[
\left( T_w - T_p \right) \frac{\rho C_\mu C_{\mu}^{1/4} k_{p}^{1/2}}{q} = \begin{cases} \Pr \cdot y^*, & y^* < y^*_r, \\ \Pr_{\text{turb}} \left[ \frac{1}{\kappa} \ln \left( E \cdot y^* \right) + P \right], & y^* > y^*_r, \end{cases}
\]

where \( P = 9.24 \left( \frac{\Pr}{\Pr_{\text{turb}}} \right)^{3/4} - 1 \left[ 1 + 0.28 \exp \left( -0.007 \frac{\Pr}{\Pr_{\text{turb}}} \right) \right] \).

The value of \( y^*_r \) is found by solving the equation: \( \Pr \cdot y^* = \Pr_{\text{turb}} \left[ \frac{1}{\kappa} \ln \left( E \cdot y^* \right) + P \right] \).

Symmetry conditions are set on the outer boundaries of the computational domain, being the planes \( \Gamma_2, \Gamma_4, \Gamma_5, \Gamma_6 \). The condition \( V_n = 0 \) is set for the velocity component normal to the boundary, for all other variables \( \frac{\partial \phi}{\partial n} = 0 \). \( \Gamma_1 \) and \( \Gamma_3 \) are the walls. In the case of the forced ventilation of the area, it is assumed that through the border \( \Gamma_1 \) air enters the room and leaves through the border \( \Gamma_3 \). In this case, the boundary conditions are: \( V_z = V_o, \ T = T_0 \). The turbulent flow parameters are determined by the
intensity of the turbulent pulsations and hydraulic diameter $D_{hyd}$:

$$k = \frac{3}{2} (I \cdot V_0)^2; \quad \varepsilon = C_{\mu}^{3/4} \frac{k^{3/2}}{\ell};$$

$$\ell = 0.07 \cdot D_{hyd}.$$ The pressure at the $\Gamma_3$ boundary is equal to the atmospheric one.

4. Solution method and calculation results

The following thermophysical characteristics of the air are used in the calculations [4]:

$c_p = 1006 \, \text{J/(kg·K)}$; $\lambda = 0.0242 \, \text{W/(m·K)}$; $g = 9.81 \, \text{m/s}^2$; $\mu = 1.8 \cdot 10^{-5} \, \text{Pa·s}$; $M = 29 \cdot 10^{-3} \, \text{kg/mol}$; $Pr_{turb} = 0.85$; $C_{1c} = 1.44$; $C_{2c} = 1.92$; $C_{\mu} = 0.09$; $\sigma_i = 1.0$; $\sigma_e = 1.3$.

To solve the equation system (1) - (6), we use the simulation software "Ansys Fluent". The system is solved by the Patankar method using the SIMPLE algorithm. We have carried out the study on the grid convergence. The calculation accuracy is controlled by the balance of the heat and mass fluxes. The computational domain has 960128 cells. The difference grid step size changes from $h_x = h_y = h_z = 3 \cdot 10^{-2} \, \text{m}$ to $h_x = h_y = 3 \cdot 10^{-3} \, \text{m}$ near the boundaries of the heat exchange surface. The chosen difference grid provides the fulfillment of the mass conservation law with the accuracy of 99.7% and the calculation accuracy of the heat fluxes on the surface of the electrical equipment is 96.6%.

Fig. 3 shows the steady-state temperature fields in the vicinity of the electrical equipment. There are two variants of the calculations in the figure: a - natural convective airflow, b - forced air motion. There are two limiting variants of the airflow and heat transfer, presented in Fig. 3.

![Temperature field in the vertical middle section of the computational domain](image)

Figure 3. Temperature field in the vertical middle section of the computational domain

According to the obtained results, the cooling air is stratified in height: colder air is at the bottom and warmer air is at the top (Fig. 3). The greatest temperature gradient in the vertical direction in case of the natural convection is observed in the areas below and above each generating sets (Fig. 3 a). There is a slight change in temperature along the height near the generating sets (Fig. 3a). The air temperature at the top is 290.7 K (Fig. 3a). When the temperature of all the walls of the equipment rises by 2 degrees, the air temperature at the top is 293.47 K. Fig. 3a shows that the upper level of the generating sets is in conditions of the increased air temperature. Overheating of the generating sets is possible in this area.

To prevent this effect, it is possible to use intermittent ventilation. When cold air is supplied through the lower boundary of the region, all tiers of the electrical equipment are cooled (Fig. 3b). The air heats up by about 2°C. Both modes satisfy the thermostating conditions of the generating sets. The amount of energy for forced air motion is known from the cost of the ventilation and air conditioning. The question
arises: are natural convective airflows capable of removing heat energy and what kind of heat fluxes arise in this case?

Natural convective air motion in the vicinity of the equipment has a complex vortex character. There is a qualitative similarity of the air motion from one level of the electrical equipment to another. The air in contact with the cold areas of the generating set surface descends, whilst the air in contact with warm areas rises. The heated air rises and contacts the cold sections of the next tier of the generating sets, gradually cools down, changes the direction of motion, and starts descending. Natural convective vortices occur between the tiers of the generating sets. The turbulent vortical motion is involved in thermostating of the generating sets. The ascending airflows take part of the generated heat from the surfaces of the generating sets with high temperature and transfer it to the generating sets located above, which have areas with low temperatures at the bottom. The calculated heat fluxes on the surfaces of the electrical equipment are presented in Table 1 and illustrated in Fig.4. Reliable calculation accuracy is about 0.01 W.

The diagrams (Fig. 4) of the heat fluxes distribution on the surfaces of the electrical equipment show the effect of the temperature differences and stratification on the direction and values of the heat fluxes.

The dark color corresponds to the right row of the electrical equipment, the lighter one - to the left. The positive values correspond to the heat flux into the air, and negative ones – to that from the air. As can be seen from the diagrams, on the lower (first) tier (Fig. 4a), the upper part emits the heat to the surrounding air, whilst the heat flux to its lower part is insignificant. For the generating sets located above, the intensity of the heat flux into the lower part of the equipment increases (Fig. 4 c-e). The air
velocity reaches the highest values in the area of 2 and 3 tiers (Fig. 4 b, c). The greatest differences in the direction of the heat fluxes are observed here. Near the fifth tier, this difference decreases.

The calculation results of the total positive and negative heat flux to the generating sets are presented in Table 1. The generating sets of the first tier and the third tier give off heat to the ambient air. Other units receive heat from the air. Heat fluxes for two adjacent pieces of the equipment (left and right rows) are different due to the hydrodynamics of the air flow.

Table 1. Total flux absorbed by the surface of the equipment, W.

| Row  | Tier 1   | Tier 2   | Tier 3   | Tier 4   | Tier 5   |
|------|----------|----------|----------|----------|----------|
| Left | 0.516906 | -0.21986 | 0.161542 | -0.15862 | -0.1561  |
| Right| 0.467085 | -0.11478 | -0.13286 | -0.17814 | -0.22543 |

The total heat flux from all electrical units into the air is 1.1455 W, and the total heat flux to all electrical units is 1.1858 W. The balance of the fluxes (the sum of the positive and negative fluxes) is -0.0403 W. This value is less than 4% of the total heat flux coming to the electric units from the air. In the absence of the forced air motion, the resulting natural convective movement provides almost zero balance of the heat fluxes: the heat removed into the air from the warm sections of the electrical units equals the heat entering the cold areas of the electrical units. In fact, the air transfers the heat between the surface areas of the generating sets with different temperatures. This mainly occurs between the surfaces of the units located on different tiers. This phenomenon has a significant effect on the thermostating of the electrical units in the absence of the forced air motion.

The numerical modelling of the natural convective air motion around the electrical equipment shows the possibility of thermostating objects without forced convection. It is possible to achieve a zero total balance of the heat fluxes by creating the conditions for the occurrence of natural convection, when the generating sets are located one above the other.

5. Conclusion

Analysis of the results obtained on the natural convective air motion providing the heat exchange between surfaces of the electrical equipment and air shows the possibility to reduce the energy consumption for thermostating objects. It is possible to achieve a complete closure of the heat fluxes within the system by creating the conditions of intensive natural air motion. The values of the air exchange leading to the disappearance of the natural convective air motion have been determined numerically. The developed mathematical model allows investigating the processes of the heat transfer that occurs in real conditions and carrying out the preliminary estimates for the subsequent practical use of the natural convective air flow for thermostating electrical equipment. The method to determine the dependences of the temperature field formation under the natural convective, forced and mixed modes of air motion in the vicinity of the thermostatically controlled electrical equipment has been successfully tested on the operating industrial enterprise - Siberian Chemical Combine.

References:
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