HEAT EXCHANGE OF A BANK OF TUBES IN THE TRANSVERSE FLOW OF LIQUID COMPLICATED BY THE FORMATION OF THE SOLID PHASE OF THE HEAT CARRIER

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Results of numerical and experimental study of the heat exchange process are considered as applied to the calculation of helical heat exchangers operating under conditions of unavoidable formation of the solid phase of the heat carrier in the intertube space. Based on the developed physical model of uniform icing, the system of differential equations that describes heat exchange in a two-flow apparatus is integrated. Comparison of the results obtained and experimental data shows that there is a limitation on the application of the model of uniform icing.

Heat transfer process in media involving a change in the phase state, which are widespread in cryogenics, power engineering, metallurgy, and other fields, are described by the Stefan class of problems. In a classical definition, i.e., under the assumption of the absence of convection and under the assumption of constant thermophysical properties of liquid, the law of motion of the liquid—ice boundary in time is determined by solving the system of equations of heat conduction with nonlinear boundary conditions. The basis methods of solving such problems are classified and represented most completely in [1]. It should be noted that when constructing a solution, the heat transfer coefficient on the ice—liquid boundary for most practical cases is selected on the basis of known empirical dependences to be equal to the coefficient for a "clean" surface [2, 3]. Whether it is correct to assume that heat exchange processes on the ice—liquid boundaries and "clean" surface—liquid boundaries are adequate requires thorough experimental verification.

Therefore, there exists another side of the known Stefan problem: determining heat transfer on the ice—liquid boundary (or determiing boundary conditions of the third kind in the Stefan problem). Apparently, the given problem can be formulated as a quasi-stationary problem on finite time intervals.

This approach allowed us to develop the technique of calculating spiral tube casing heat exchangers with account of the formation of ice on separating walls of channels [4] based on a physical model of uniform icing. In constructing the model the following main assumptions have been accepted: the thickness of icing is the same over the perimeter of a tube and is uniform over the layers of the heat exchanger; the temperature on the phase boundary is constant and equals T_f; heat transfer on the ice—liquid boundary and the liquid—"clean" surface boundary are adequate.

The given design model is physically justified if we neglect a certain nonuniformity in the distribution of the coefficient of heat transfer over the perimeter of a separate tube.

From the condition that the process is quasistationary, i.e., form the condition of equality of heat flows directed from the cooling agent through the iced wall to the cooled flow over finite time intervals, the thickness of the ice "crust" δ_i in the i-th cross-section is determined as the root of the transcendental equation:

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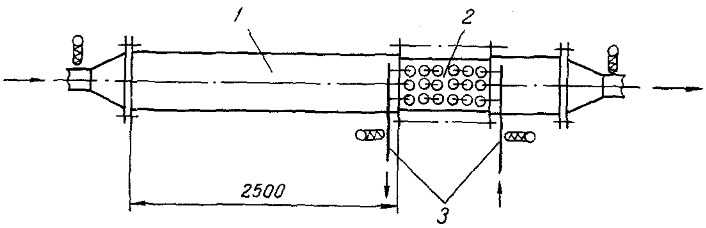

Fig. 1. Outline of the experimental plant: 1) the region of hydrodynamic stabilization; 2) visualization model; 3) collector of high-pressure air.

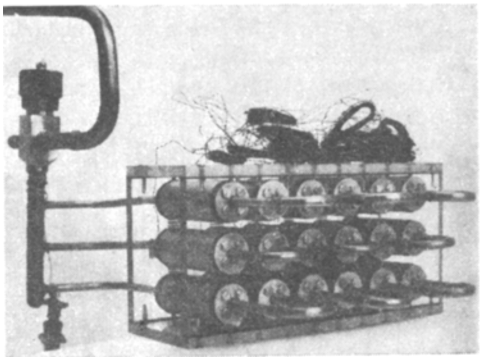

Fig. 2. Visualization model.

$$T_f = \frac{T_{1i} + T_{2i}\left(\dfrac{\alpha_{2i} d_i \ln\left(\dfrac{d_2}{d_1}\right)}{2\lambda_{wi}} + \dfrac{\alpha_{2i} d_i \ln\left(\dfrac{d_i}{d_2}\right)}{2\lambda_{\text{л}i}} + \dfrac{\alpha_{2i} d_i}{\alpha_{1i} d_1}\right)}{\dfrac{\alpha_{2i} d_i \ln\left(\dfrac{d_2}{d_1}\right)}{2\lambda_{wi}} + \dfrac{\alpha_{2i} d_i \ln\left(\dfrac{d_i}{d_2}\right)}{2\lambda_{\text{л}i}} + \dfrac{\alpha_{2i} d_i}{\alpha_{1i} d_1}}. \quad (1)$$

It should be noted that Eq. (1) is a known form of the equation for the surface temperature of a multiple-layer cylindrical wall.

Heat exchange in the spiral heat exchanger is described by the system of ordinary differential equations of the first order

$$\frac{dT_1}{dx} = \frac{1}{c_{p1} G_1} K_s (T_1 - T_2), \quad \frac{dT_2}{dx} = \frac{1}{c_{p2} G_2} K_s (T_2 - T_1), \quad (2)$$

$$\frac{\Delta p_2}{\Delta x} = \frac{\cos(\varphi)}{t} \xi \frac{w_2^2}{\rho_2}.$$

The system was solved by the Runge–Kutta method with the following boundary conditions:

$$T_1(0) = \begin{cases} T_1^{\text{in}} & \text{for } c_{p1} G_1 > 0, \\ T_1^{\text{out}} & \text{for } c_{p1} G_1 < 0, \end{cases} \quad (3)$$

$$T_2(0) = \begin{cases} T_2^{\text{in}} & \text{for } c_{p2} G_2 > 0, \\ T_2^{\text{out}} & \text{for } c_{p2} G_2 < 0, \end{cases} \quad (4)$$

$$\Delta p_2(0) = 0. \quad (5)$$

Integration was considered complete after the value x = H had been reached.

An analysis of the numerical solution has shown that throughout the entire studied range of the Reynolds numbers, heat transfer toward the surface of the tube is lower than that repre-

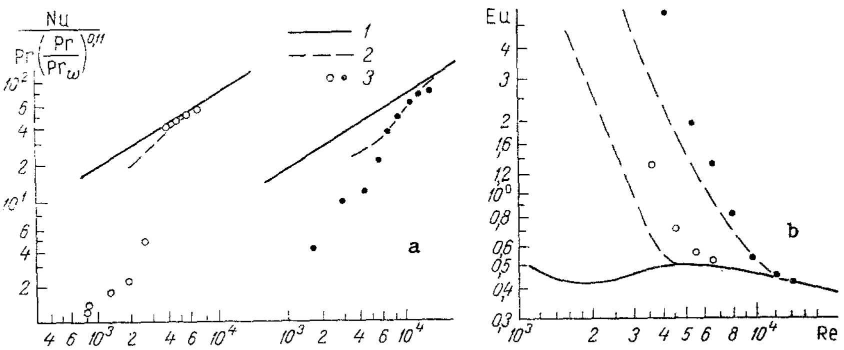

Fig. 3. Comparison of experimental and calculated data on heat transfer (a) and on hydraulic resistance (b): 1) generalized dependence of A. Zhukauskas [5]; 2) calculation from the model of uniform icing; 3) experimental data (open circles, Θ = 0.071-0.091, filled circles, Θ = 0.056-0.064). On the y-axis $\frac{Nu}{Pr^{0.36}(Pr/Pr_w)^{0.11}}$.

sented in [5]. This is due to the additional thermal resistance of the ice "crust." However, the formation of ice results also in an increase of velocity in the intertube space and, consequently, in an increase of heat transfer from the liquid to the ice surface. The interaction of these two factors determined the deviation in the calculated curve [5]. It should be noted that the outer diameter of the "clean" tube was chosen as a characteristic size for the convenience of comparison of Nu and Re criteria. The heat transfer coefficient entering into the Nu criterion was determined from dependence (8), where the calculated value

$$\bar{T}_w = \frac{1}{n} \sum_{i=1}^{n} \frac{1}{2\lambda_{\pi i}} \left(T_f - \alpha_{2i} d_i (T_{2i} - T_f) \ln \left(\frac{d_i}{d_2} \right) \right), \tag{6}$$

was assumed to be an average value of the wall temperature $\bar{T}_w$ in the presence of ice, while the average value of the liquid temperature $\bar{T}_2$ was taken as:

$$\bar{T}_2 = \frac{1}{n} \sum_{i=1}^{n} T_{2i}. \tag{7}$$

In accordance with given assumptions the calculation of the heat transfer coefficient on the ice—liquid boundary α_{2i} is realized as in [5].

The experimental plant represented a hydrodynamic outline of open type, in which circulating water was used as a heat carrier and the air under high pressure (~15 MPa) was first cooled in a bath with liquid nitrogen (Fig. 1). After a horizontal region of hydrodynamic stabilization, the water was directed into the visualization model which represented a rectangular channel bounded by plane walls made of plexiglas and packed with a bank of tube-cylinders (Fig. 2). Altogether, in a channel with a cross-section of 130 × 80 mm there were three layers with 6 cylinders in each layer. The outer cylinder diameter was 36 mm, the inner, 6 mm, the relative interval of spacing 1.2 both in the longitudinal and transverse directions, the distance between the first (last) layer and the cap being equal to half the distance between the cylinders. Tube-cylinders were made of brass, their surface was chromized. The inner caviities were connected as shown in Figs. 1 and 2, forming a flow chart for the heat carriers close to countercurrent. The experimental model, therefore, was a geometric duplicate of the coil region of the spiral heat exchanger. In the upper part of each tube of the middle layer, starting from the end of the cylinder, at a distance 0.5 mm from the outer surface and ~20 mm in depth, holes were drilled with a diameter of 1 mm, in which were placed thermocouples. Temperature was also controlled by using resistance thermometers (Fig. 1). When stable values of temperatures were achieved, the entire plant was considered to be in a stationary regime. Transfer from one regime to another one was realized by changing the water flow rate G_2.

The average heat transfer was determined in the steady-state thermal regime. When processing experimental data, the heat transfer coefficient was calculated from the following formula:

$$\alpha_2 = \frac{c_{p2} G_2 (T_2^{in} - T_2^{out})}{F_c (\bar{T}_w - \bar{T}_2)} . \tag{8}$$

The plant was adjusted when studying heat transfer of the tube bank in the absence of surface icing. Experimental data on heat transfer in "clean" regimes were in satisfactory agreement with the generalized dependence of Zhukauskas [5]. The maximum deviation did not exceed 12% in the range of velocities under investigation, which attested to the reliability of the experimental data obtained. For determining the average heat transfer, the thermophysical properties of water were chosen based on the arithmic mean temperature $\bar{T}_2$ in correspondence with [6].

An analysis of experimental data has shown that there is a possibility for the existence of two principally different groups of regime parameters, determining heat transfer and resistance in the bank of tubes (Fig. 3). Experimental data related to one group of regimes deviate slightly from [5]. Apparently, the formation of a thin crust of ice is followed in this case by a simultaneous increase in velocity, which results in a certain peculiar compensation of losses in the intensity of heat transfer. Experimental data for the other group of regimes shows that there exists a region of parameters with considerably worse heat transfer (a reduction of practically an order of magnitude is possible). Apparently, these regimes correspond to a more complicated picture of the ice formation in the intertube space than for the regimes of the first group. It is likely that with a decrease in the Reynolds numbers the influence of natural convection increases, which leads to the thermal stratification in the liquid flow, and, as a consequence, to the strong nonuniformity of ice formation over the tube layers. The experimental data obtained was compared with the numerical solution of Eqs. (2) in the framework of the presented phsyical model of uniform icing. An analysis of such a comparison shows that there exists a region of good agreement in experimental and calculated data. Deviation from the calculated curve was due to errors of the experiment and did not exceed 15%. Such agreement is characteristic of the experimental data of the first group.

Comparison of the calculated and experimental data for the second group of regime parameters shows that there exists a region where deviation of experimental data from the calculated curve is 80% and more.

Results of the conducted analysis correspond to the data of visual observations. Thus the form of frosted ice close to uniform (both over the perimeter of a separate tube and over the layers of the model) corresponds to the regimes of the first group. The application of the physical model of the uniform icing and the choice of calculated dependences for the intertube heat transfer in correspondence with [5] in this case seems to be justified.

Regimes of the second group were characteristic of a nonuniform formation of ice over the layers and possible closing-up of the lower and middle layers. This can explain the degradation in heat transfer and the sharp increase in the hydraulic resistance. An attempt to describe such a process with the help of the model of uniform icing yields large errors, discrediting accepted assumptions.

Therefore, there is a limitation on the application of the model of uniform icing of the bank of tubes in the transverse flow. Comparison of the experimental and calculated data shows that the region of application is determined by the value $Gr/Re^2 \leq 0.04$ for the studied range of Reynolds number Re = 800-1800 and Gr = 10^5-10^7.

NOTATION

x, coordinate along the length of the heat exchanger (model); H, the length of winding of the heat exchanger (maximal value of x); d_1, inner diameter of the tube; d_2, outer diameter of the "clean" tube; T(x), flow temperature; T_f, phase transition temperature; $\alpha(x)$, heat-transfer coefficient; F_c, "clean" surface of heat exchange for the experimental model; φ, slope of winding with respect to the diametral plane; t, diametral winding pitch; β, volumetric expansion coefficient; $K_s(x) = ((\alpha_1 S_1)^{-1} + \ln(d_2/d_1)(d_2/2S_1)/\lambda_w + \ln(d/d_2)(d_1/2S_1)/\lambda_л + (\alpha_2 S_2)^{-1})^{-1}$, heat-transfer coefficient per unit length; S(x), surface of heat exchange per unit length; w, flow velocity; G, flow rate; $\Delta p(x)$, hydraulic resistance; $\xi(x)$, coefficient of hydraulic resistance; n, number of integration intervals; $c_p(x)$, $\rho(x)$, $\mu(x)$, and $\lambda(x)$, heat capacity, density, viscosity, and heat conduction, respectively; $Gr(x) = 9{,}81\,\beta\rho_2^2(\bar{T}_2 - \bar{T}_w)d_2^3/\mu_2^2$, $Nu(x) = \alpha d/\lambda$, $Re(x) = w\rho d/\mu$, $Eu(x) = 2\Delta p/(\rho w^2)$, and $Pr(x) = c_p\mu/\lambda$, Grashof, Nusselt, Reynolds, Euler, and Prandtl numbers, respectively; $d(x) = d_2 + 2\delta$, outer tube diameter together with the "curst" of ice; $\Theta = T_2^{in} - T_f)/(T_j -$

T_1^{in}); relative temperature head. Indices: 1, tube space; 2, intertube space; w, wall; i, ice; in, out, parameters corresponding to the input and output of the heat exchanger (model).

LITERATURE CITED

1. L. A. Kozdoba, Fiz. Khim. Obrab. Mater., No. 2, 41-59 (1973).
2. N. I. Sokolov, Metrol. Gidrol., No. 3, 44-45 (1954).
3. S. Chuklin, Kholod. Tekh., No. 1, 46-55 (1948).
4. A. V. Vansevich and A. I. Smorodin, "A study of processes in the plants and systems of cryogenic mechanical engineering," in: Collection of Scientific Papers of NPO "Kriogenmash," pp. 106-115, Balashikha (1984).
5. A. Zhukauskas, V. Makaryavichus, and A. Shlanchyauskas, Heat Transfer of Tube Banks in the Transverse Flow of Liquid [in Russian], Vilnius (1968).
6. S. L. Rivkin and A. A. Aleksandrov, Thermophysical Properties of Water and Steam [in Russian], Moscow (1980).

FLOW AND HEAT TRANSFER IN A NONSTEADY JET GENERATED BY LARGE-AMPLITUDE GAS OSCILLATIONS

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The problem of the velocity field in a jet formed by nonlinear oscillations in a tube is solved. Equations are derived that describe heat transfer to a body placed at the jet axis. These relationships are tested experimentally.

Jet flows of an incompressible fluid and a gas comprise one of the best developed fields of boundary-layer science. The importance of this branch of knowledge is due to the wide prevalence of jet flows in nature and engineering. An entire class of jet flows exists, however, that has hardly been investigated up to now, namely, nonsteady jets. Such jets can be produced using a generator of large-amplitude oscillations, which consists of a resonance tube, one end of which is open and communicates with the ambient medium, and at the other end of which a flat piston moves harmonically [1]. Upon the excitation of oscillations in the tube, a nonsteady jet is formed at its open end, with the amplitude of velocity oscillations in the jet reaching 160 m/sec near resonance. Another way of generating a high-velocity nonsteady jet has been described in [2].

The high amplitude of the velocity pulsations makes it possible to use the generated jet to investigate heat transfer between bodies and oscillating flows in a wide range of variation of the oscillatory Reynolds number Re_{osc} and the Strouhal number Sh. Heat transfer between bodies and this jet is also interesting because the velocity oscillations in it are anharmonic: the spectrum of velocity oscillations contains a constant component and a number of harmonics [3, 4].

In this paper we attempt to investigate the velocity field in a nonsteady jet generated at the open end of a pipe during nonlinear gas oscillations in it, as well as heat transfer for bodies (a cylinder, sphere, and disk) placed at the jet axis.

To solve the hydrodynamic part of the problem, we used the law of conservation of momentum [5], which in the case of a nonsteady jet takes the form

$$\frac{\partial}{\partial t}\int_0^\infty ru\,dr+\frac{\partial}{\partial x}\int_0^\infty ru^2dr=0. \tag{1}$$

We assume that the axial velocity u can be represented as a sum $u = u_0 + u_1 + u_2$, where u_0 is the time-averaged component and u_1 and u_2 are the first and second harmonics, with $u_1 \approx u_0$ and $u_2 \ll u_1$. After substituting u into (1) and averaging over time, we obtain

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