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HEAT EXCHANGE IN TWO-PHASE FLOW ABOUT A SURFACE LOCATED IN A RECTANGULAR CHANNEL

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Similitude equations are obtained on the basis of the principle of superposition of separate effects to calculate heat exchange between surfaces with complex-shaped cross sections located in a rectangular channel during their cooling by a two-phase flow.

The design of equipment characterized by high levels of heat release and which satisfies the requirement of maintaining the thermal state of the heat-exchanging surfaces within a specified temperature range involves the use of efficient methods of cooling. Mass-exchange cooling has come into wide use in this regard. One form of mass-exchange cooling is cooling with a two-phase flow of a gas and a finely dispersed liquid. The main advantages of this method compared to convective cooling are: a reduction in the temperature of the surface, with the direct contact of the liquid droplets with the wall; a reduction in the temperature of the coolant as a result of evaporation of moisture in the gas flow; an increase in the specific heat of the coolant (the gas—vapor—liquid mixture); an increase in the heat-transfer coefficient as a result of additional agitation of the flow by the liquid particles. In some cases, it proves best to employ a combination method — convective cooling together with the use of a two-phase flow during the operation of units under "peak" loads. Here, it is necessary to know the limits of applicability of the convective cooling and the consumption of liquid required in relation to the service conditions of the unit (the heat flux, time of operation under "peak" load, number of startups of the water-supply system, etc.).

However, the data available in the literature pertains mainly either to the flow of two-phase mixtures in channels or to the interaction of individual droplets with a hot surface. Furthermore, the final results cannot be generalized so as to yield a suitable design of cooling system for hot surfaces of complex shape located in a channel admitting a two-phase flow.

The present work presents results of a study of heat exchange and the thermal state of electrically heated metal surfaces (resistors) of length L, width H, and thickness 2δ when the inequalities $L \gg 2\delta$, $H \gg 2\delta$, $L \gg H$ are observed and the resistor is cooled on both sides. Here the temperature field of the resistor depends only on the coordinate y and, assuming that $\lambda = \text{const}$, the mathematical description of the heat-conduction process for steady-state conditions has the form

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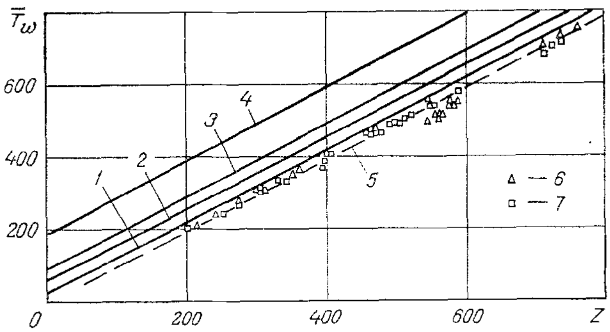

Fig. 1. Diagram for comparative evaluation of the thermal state of a surface. Calculation. 1) T_f = 30°C; 2) 70; 3) 100; 4) 200; 5) T_f = 20°C; Experiment: T_f = 20-25°C; 6) single-phase flow; 7) two-phase flow.

$$\frac{d^2T}{dy^2}+\frac{q_v}{\lambda}=0,$$

$$y=\delta,\ \alpha(T_w-T_f)=-\lambda\left(\frac{dT}{dy}\right)_{y=\delta}, \tag{1}$$

$$y=-\delta,\ \alpha(T_w-T_f)=\lambda\left(\frac{dT}{dy}\right)_{y=-\delta}.$$

Here $q_V = I^2R/V$ is the output of the internal heat sources.

Considering the dependence of the resistance of the heated material on the temperature $R = R_0(1 + aT)$ and using the substitution

$$T^* = bT + c,$$

where $b = aq_{V_0}/\lambda; c = q_{V_0}/\lambda$, we obtain an expression for calculating the temperature of the surface on the assumption that the temperature field is symmetrical relative to the origin:

$$T_w=\frac{T_f+\dfrac{c\lambda}{\alpha\sqrt{b}}\operatorname{tg}(\sqrt{b}\,\delta)}{1-\dfrac{\lambda\sqrt{b}}{\alpha}\operatorname{tg}(\sqrt{b}\,\delta)}. \tag{2}$$

Estimates show that, assuming the initial inequalities are observed, even at very high heat outputs ($q_V \geqslant 2\cdot10^5$ kW/m^3) the value of $\sqrt{b}\delta$ does not exceed 0.05. In this case, we may assume that $\tan(\sqrt{b}\delta) \simeq \sqrt{b}\delta$. Taking this into account, the expression for T_w has the form

$$T_w=\frac{T_f+\dfrac{q_v\delta}{\alpha}(1+aT_w)^{-1}}{1-a\dfrac{q_v\delta}{\alpha}(1+aT_w)^{-1}}. \tag{3}$$

It follows from (3) that the parameter $Z = q_V\delta/\alpha$ and its dimensionless analog $Z^* = aZ$ unambiguously determine T_w and can be used to estimate the thermal state of the surface given different combinations of its dimensions, the level of heat output, cooling rate, and initial conditions.

The agreement of Eq. (3) with measurements of the temperature of the surface of a model enveloped by both single- and two-phase flows (Fig. 1, line 5) confirms the correctness of the original assumption on the uniformity of the temperature field in the resistor and demonstrates the universality of the parameters Z and Z* in evaluating the thermal state of complex-shaped surfaces. Curves allowing the use of values of Z or Z* to determine the rate of heat transfer required to ensure the necessary temperature for the surface under different operating regimes of the latter prove very useful in engineering calculations (Fig. 1).

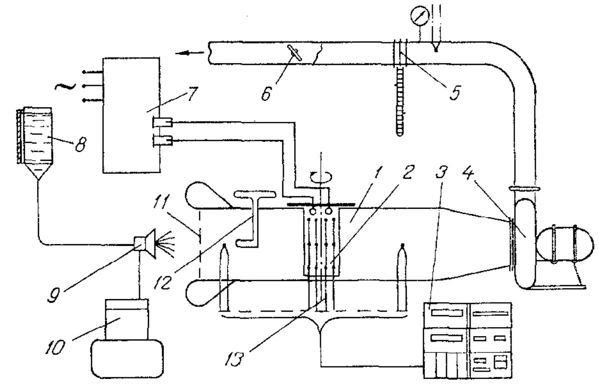

Fig. 2. Experimental unit: 1) working channel; 2) test model; 3) data-measurement system; 4) fan; 5) measurement diaphragm; 6) butterfly valve; 7) rectifier; 8) measuring vessel; 9) nozzle; 10) compressor; 11) netted distributor; 12) pitot tube; 13) thermocouples.

It is particularly important in applying the results of studies of heat transfer on a model cooled by a two-phase flow to other objects that the condition Z^* = idem be observed, since the character of interaction of the droplets contacting with the hot surface changes in relation to the temperature of the material (subspheroidal or spheroidal drops [1, 2]) and the heat transfer rate.

It follows from (3) that evaluation of the thermal state of the surface from the parameters Z and Z^* requires that we have data permitting us to determine the heat-transfer coefficient as a function of the form and orientation of the surface, as well as the regimes of flow of the coolant and the amount of sprayed moisture.

The average rate of heat transfer was determined empirically on a unit which took the form of an open wind tunnel (Fig. 2). Forced air movement was accomplished with a centrifugal fan. Air flow rate was regulated with a butterfly valve. The working section was a rectangular channel 26 × 130 mm in cross section and 600 mm long. The inlet device, designed from a Vitoshinskii curve, ensured a uniform velocity profile at the inlet to the channel. Heat losses were reduced by thermally insulating the entire channel.

The test models were made of strips of alloy Kh13Yu4 in the form of two parallel surfaces 112 mm long and 1.2 mm thick, located 10 mm from each other. One of the models (12 mm wide) was flat, while the other (H = 48 mm) has swirl vanes turned to one side. The height-to-width ratio of the vanes was 0.33. The models were secured to a device which rotated them about their vertical axis. The rotation device was mounted on the removable top cover of the working section. At zero rotation, the plane of the model was parallel to the broad side of the channel.

The models were heated by a direct current from the VGK-100 rectifier, the design of which made it possible to vary the heat flux by changing the number of turns and the voltage on the secondary winding of the transformer. The temperature distribution on both sides of the heated surfaces and the temperature of the coolant and channel walls were measured with 0.3-mm-diameter Chromel—Copel thermocouples connected to a K-200/4 data-measurement system. The measured quantities were recorded on punch tape at a channel interrogation frequency of 1 Hz.

Water was delivered to the channel inlet through a regulable air nozzle. Water flow rate was determined by the volume method. A uniform distribution of the liquid phase at the inlet was ensured by a fine-mesh capron distributing screen.

The tests were conducted within the range of values of the Reynolds number $\overline{Re}_f = (0.33\text{-}6.1)\cdot 10^4$, temperature factor $\psi = 1.05\text{-}3.53$, and relative weight content of liquid phase X = 0-14 g H_2O/kg air. The volumetric heat flow rate in the heating of the model was $(0.8\text{-}3.2)\cdot 10^5$ W/m^3, and the angle of attack was changed from 0 to 15°.

The mean heat-transfer coefficient was determined by the calorimetric method with allowance for heat loss. For the latter, we conducted heat-exchange experiments involving flow

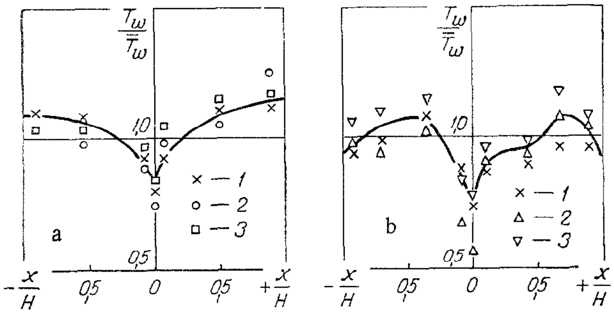

Fig. 3. Temperature distribution on both sides of the surface: a) flat surface (1 — single-phase flow; 2 — X = 4.9; 3 — 19.9 g/kg); b) surface with vanes (1 — single-phase flow; 2 — X = 7.8; 3 — 3.2 g/kg).

about a smooth, heated plate ($\psi > 1$). Here, we obtained the dependence of the heat losses that could not be modeled on the number Re_f and the power supplied to the plate.

The results were generalized in the form of similitude equations using the principle of superposition of individual factors — the two-phase nature of the flow, the angle of attack, the shape of the surface (the presence of the vanes). As the determining quantities, we took the width of the model and temperature of the incoming flow.

The theoretical formulas for determining average heat transfer have the form:

for the flat surface

$$\overline{\mathrm{Nu}}_f = \overline{\mathrm{Nu}}_{f_0}\varepsilon_m\varepsilon_\varphi; \tag{4}$$

for the surface with swirl vanes

$$\overline{\mathrm{Nu}}_f = \overline{\mathrm{Nu}}_{f_0}\varepsilon_m^*\varepsilon_\varphi^*\varepsilon_s^*. \tag{5}$$

The relative functions ε_m and ε_m^* in Eqs. (4) and (5) consider the effect of the two-phase nature of the flow on heat transfer, while ε_φ and ε_φ^* account for the angle of attack and ε_s^* accounts for the presence of the vanes. The value of $\overline{\mathrm{Nu}}_{f_0}$ is determined from the heat-transfer equations for a flat plate.

The transition to turbulent flow in experiments conducted with a single-phase flow under appreciably nonisothermal conditions is observed at $\overline{\mathrm{Re}}_{f\,cr} \simeq 10^4$, which qualitatively and approximately quantitatively agrees with the data in [3]. The selection of a theoretical formula for $\overline{\mathrm{Nu}}_{f_0}$ was therefore dictated by the value of the number $\overline{\mathrm{Re}}_f$. At $\overline{\mathrm{Re}}_f < 10^{-4}$, we used the relation for laminar flow. At $\mathrm{Re}_f > 10^4$, we used the relation for turbulent flow.

The study results are shown in Figs. 3 and 4. Analysis of the temperature distribution on both sides of the surfaces of the models shows that there is similitude of the temperature profiles for each type of surface at any $X \geqslant 0$ (Fig. 3). The temperature profile for cooling of the flat model is symmetrical (Fig. 3a). The presence of the vanes makes the temperature field of the model nonsymmetrical (Fig. 3b) due to the fact that the flow must pass over a surface of complex shape with sections of different curvature. The quantity $T_w^{max}/\overline{T}_w$, characterizing the noniniformity of the temperature field, is 1.18-1.20 for all regimes. However, even the disruption of the temperature field symmetry by the vanes does not affect the character of the function $T_w(Z)$ (see line 5 in Fig. 1).

Analysis of the test results allowed us to obtain theoretical relations to determine the individual functions:

$$\varepsilon_m = 1 + 0.26X^{0.71} \text{ (Fig. 4a, } 1), \tag{6}$$

$$\varepsilon_s^* = 1 + 0.057X \text{ (Fig. 4a, } 2), \tag{7}$$

$$\varepsilon_s^* = 0.045\,\overline{\mathrm{Re}}_f^{\,0.36} \quad (\overline{\mathrm{Re}}_f > 10^4) \text{ (Fig. 4b)}. \tag{8}$$

For a single-phase flow, $\varepsilon_\varphi = 0.8$, $\varepsilon_\varphi^* = 1.15\text{-}1.20$; for a two-phase flow, $\varepsilon_\varphi = \varepsilon_\varphi^* = 1$.

Thus, the experiment showed that the principle of superposition of separate effects may be used to calculate the rate of heat exchange between a heated surface and a two-phase flow.

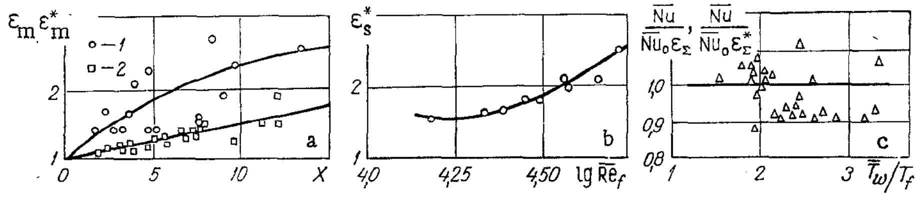

Fig. 4. Results of study of surface heat transfer: a) relative effect of two-phase nature of flow on heat transfer (1 — flat surface; 2 — surface with vanes); b) relative effect of shape of surface; c) generalization of heat-transfer data.

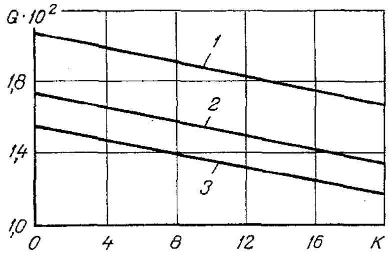

Fig. 5. Effect of frequency of startup of water spray system (K) on the unit water-flow rate: 1, 2, 3) different total times of operation (τ) of water system, $\tau_1:\tau_2:\tau_3 = 4:2:1$. $G \cdot 10^2$, g/kJ.

The results obtained show that the selection of a method of cooling depends significantly on the shape of the surface. Swirl vanes intensify the process 1.5-2.5 times (Fig. 4b). The presence of a two-phase (versus single-phase) flow increases the rate of heat exchange with the flat surface 1.5-2 times (Fig. 4a). If both factors — swirl vanes and two-phase flow — are acting, the relative effect of the two-phase flow is less than for the flat surface ($\varepsilon_m^* < \varepsilon_m$).

A change in the angle of attack from zero in the two-phase flow has no effect of surface heat exchange; in the single-phase flow, a change in orientation reduces heat exchange with the flat surface ($\varepsilon_\varphi < 1$) and increases heat exchange in the presence of the vanes ($\varepsilon^* > 1$). This is due to the formation of zones with different local conditions for flow about the body when the latter is rotated within the narrow rectangular channel.

The results obtained here in each specific case allow us to determine the feasibility and limits of single-phase cooling and the water flow rates necessary to ensure the required temperature of the heated surface. The water flow rate per unit of heat released during heating depends on the heat flux, rate of heating of the surface, total time of operation of the unit on regimes requiring the use of two-phase cooling, and the number of startups of the water spray system. An increase in the number of startups of the system and a shortening of the periods of its operation lead to a reduction in the required water flow rate (Fig. 5).

NOTATION

T, q, temperature and heat flux; $\overline{T}_w$, mean surface temperature; I, R, current and electrical resistance; V, volume of the material; α, λ, and a, heat-transfer coefficients, thermal conductivity, and linear expansion of the material; ε, relative functions; $\varepsilon_\Sigma = \varepsilon_m \varepsilon_\varphi$; $\varepsilon_\Sigma^* = \varepsilon_m^* \varepsilon_\varphi^* \varepsilon_s^*$; ψ, temperature factor; X, relative weight content of liquid phase. Indices: w, surface; f, incoming flow; v, volume; m, two-phase flow; φ, angle of attack; s, shape of surface; *, pertains to surface with swirl vanes.

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