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HEAT TRANSFER WITH CONDENSATION ON MESH CAPILLARY STRUCTURES OF HEAT PIPES

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The authors present results of an experimental investigation of heat transfer with condensation of water vapor on the capillary structures of heat pipes. They compare the test data with theory.

The thermal resistance of low-temperature heat pipes depends on the intensity of heat transfer in the condensation section more strongly, the less is the heat removal area compared with the surface area of the heat supply zone. The need to reduce the mass and size of heat exchange equipment has led to the situation that in heat pipes with minimum possible length of the condensation section the total thermal resistance of the pipe is roughly the same as the thermal resistance of the heat removal section. Therefore for this type of heat pipe it is especially important to have: 1) well-founded physical ideas as to the heat removal mechanism for different conditions of condensation on the capillary surfaces of the heat pipe; 2) reliable knowledge of the dependence of the heat removal intensity on the basic independent factors.

The presently available experimental data on heat removal with condensation in capillary structures has been obtained from tests on heat pipes (e.g., [1-5]). In most cases the test results suggest a thermal resistance with condensation in the form

$$R_c = \delta_\phi/\lambda_e. \quad (1)$$

It is noted in [1] that on the surface of a metal fiber capillary structure with condensation there is a liquid film of thickness on the order of 10 μm. It is suggested that the thickness of this film is constant and does not depend on the main regime and geometrical parameters.

Reference [2] suggested a method of calculating R_c, using the heat transfer coefficient with condensation α_c, determined from the correlation equations. But here no basis was given for choice of the correlation or the structure of the correlation equations.

It should be noted that in tests on heat pipes one can have too little and too much heat transfer agent, leading to a deviation of the heat transfer coefficients with condensation from the nominal value.

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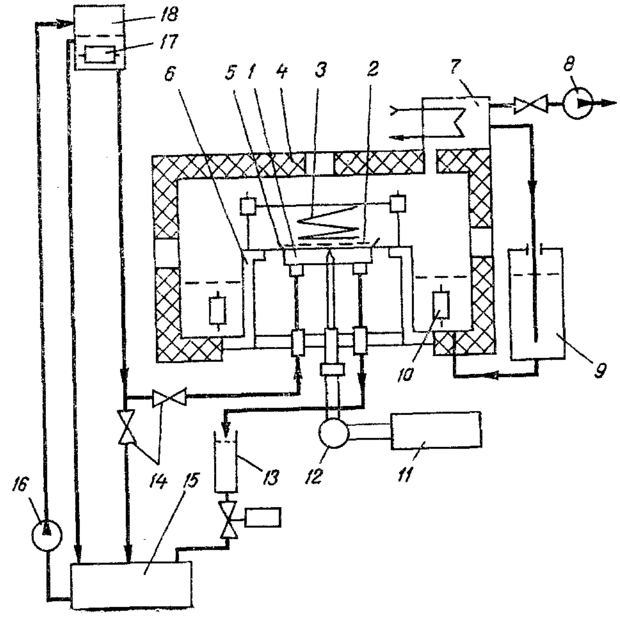

Fig. 1. Diagram of the experimental stand: 1) cooled plate; 2) test specimen of capillary structure; 3) clamp device; 4) chamber; 5) protective screen; 6) body of the working section; 7) main condenser; 8) vacuum pump; 9) supplementary condenser; 10, 17) electric heaters; 11) microvoltmeter; 12) thermocouple switch; 13) flowmeter; 14) control valve; 15) thermostat; 16) pump; 18) constant level tank.

In cylindrical heat pipes one cannot avoid this kind of distribution of liquid around the perimeter of the body, and then the test results depend on the position of the temperature sensors and the method of averaging their readings. The reliability of experimental data on heat transfer with condensation is also affected by heat leakage, the dependence of pressure on heat flux density, the degree of contact between the capillary structure and the body, the possible presence of noncondensible gas, and so on.

To obtain complete information on condensation on capillary porous structures we built an experimental stand as shown diagrammatically in Fig. 1. The main element is the working section, being a sealed cylinder in the top part of which is mounted a cooled nickel plate, using diaphragms of stainless steel foil. Above the plate we used a clamp arrangement to mount specimens of capillary structure. The surface temperature of the cooled plate was measured by ten welded copper-constantan thermocouples. The working section was located in a thermally insulated sealed chamber partially filled with heat transfer agent, which was heated to the saturation temperature.

Figures 2 and 3 show the results of experiments on heat removal with condensation of water vapor on the stainless sections of normal mesh for a slope angle to the horizontal of $\phi = 10°$. It can be seen that the mesh capillary structure on the heat transfer surface reduces the heat transfer coefficients in condensation compared with a surface without a capillary structure. The reduction of the values of α_c is proportional to the thickness of the structure used. The values of α_c calculated from Eq. (1) (see Figs. 2 and 3) do not agree either in nature or in value with the direct experimental data. This suggests that the mechanism of the process depends not only on the equivalent thermal conductivity of the wick but also on other factors.

In order to generalize the results of the experimental investigations of heat removal with condensation on an inclinced flat surface with superimposed mesh capillary structure we formulate a physical model of the process (Fig. 4). The following assumptions are made:

1) the dry saturated vapor at rest is condensed;

2) the physical properties of the condensate are constant;

3) the thermal resistance of the phase transition can be neglected, and the heat removal intensity is determined by the heat conduction from the phase change boundary to the cooled surface;

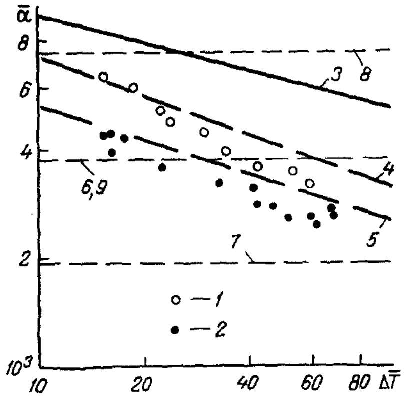

Fig. 2

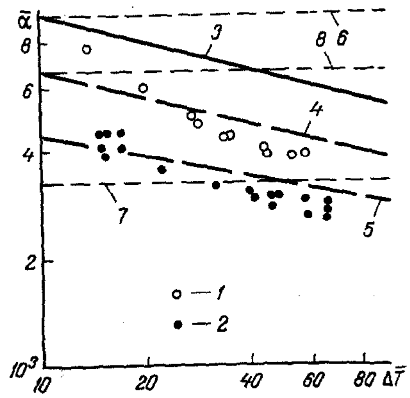

Fig. 3

Fig. 2. Heat removal during condensation of water on mesh capillary structures of stainless steel with cell dimension $\underline{a}$ = 180 μm, T_s = 100°C: 1) one mesh layer; 2) two mesh layers; 3) calculation for a plate with no capillary structure; 4, 5) calculation from Eq. (17) for one and two mesh layers, respectively, for λ_e = 2; 6, 7) calculation from Eq. (1) for one and two mesh layers with λ_e = 1; 8, 9) calculation from Eq. (1) for one and two mesh layers with λ_e = 2. The quantity $\bar{\alpha}$ is in W/(m²·K); $\Delta\bar{T}$ is in K.

Fig. 3. Heat removal during condensation of water on mesh capillary structures of stainless steel with cell size $\underline{a}$ = 71 μm, T_s = 100°C: 1) one mesh layer; 2) three mesh layers; 3) calculation for a plate with no capillary structure; 4, 5) calculation from Eq. (17) for one and three mesh layers, respectively, for λ_e = 2; 6, 7) calculation from Eq. (1) for one and three mesh layers with λ_e = 1; 8) calculation for three mesh layers from Eq. (1) with λ_e = 2 (for one layer with λ_e we have α = 19,200 W/(m²·K)).

4) in general the condensate can move in the capillary structure and simultaneously in the capillary structure and the liquid film covering it;

5) the motion of the liquid in the capillary structure and in the film can be regarded independently, i.e., the flow in the capillary structure obeys Darcy's law in the form

$$\frac{dP}{dx}=\frac{\mu' G_\Phi}{K_\Phi \rho' F_\Phi}, \tag{2}$$

and the motion in the film corresponds to the law for planar laminar flow

$$G_f=\frac{(\rho'-\rho'')\,g\sin\varphi}{3\nu'}\,\delta^3\Pi. \tag{3}$$

Thus, in the proposed approximate physical model we neglect inertia effects, which, as is known, is allowable for low speeds of film flow and flow in the porous structure;

6) the length of the condensation section is much greater than the thickness of the structure, and therefore we can neglect edge effects and can regard the specific thermal resistance in the condensation section as the sum of the thermal resistance of the film and the wetted capillary structure, accounting for variation of the thickness of the film of condensate δ as a function of the condensation conditions and the ratios between the flow rates of liquid in the film, in the structure, and the total of the condensate;

7) the liquid moves under the action of forces of gravity and capillary forces for which the dynamic head is formed outside the condensation section. In our special case we can neglect this dynamic head for an inclined position of a sufficiently long section. Then the pressure gradient is associated with the hydraulic resistance of the capillary structure by the relation governing the flow rate of liquid in the fully saturated capillary structure:

$$(\rho'-\rho'')\,g\sin\varphi=\frac{\nu' G_\Phi}{K_\Phi F_\Phi}. \tag{4}$$

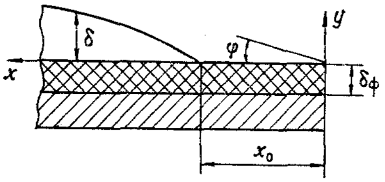

Fig. 4. Diagram of flow of condensate over the heat exchanger surface.

The total flow rate of condensate is $G_c = G_\phi + G_f$. On the other hand, the ambient local value of total flow rate of condensate is determined by the amount of heat transferred

$$Q=\int_0^x \frac{(t_s-t_{ca})\Pi}{\frac{1}{\alpha_{rem}}+\frac{\delta_w}{\lambda_w}+\frac{\delta_\Phi}{\lambda_e}+\frac{\delta}{\lambda'}}dx. \tag{5}$$

Thus, for certain x, when $x < x_0$, the inequality $G_\phi > G_c$ holds. This means that in general flow occurs only inside the capillary structure on a part of the surface of the porous structure of the condensation section. If the length of the condensation section is large enough we have $L > x_0$, and then we have a section with simultaneous flow of liquid in the capillary structure and in the film.

Since the actual wicks of heat pipes are quite thin, in the section $0 \leq x \leq x_0$, when the flow occurs in the capillary structure, on the basis of the approximate model we can neglect variation of the radius of curvature of the phase boundary and can calculate the specific thermal resistance as δ_ϕ/λ_e. For the section $x_0 < x \leq L$ we start with the following relation: $G_f = G_c - G_\phi$. If we neglect the supercooled condensate, then we have

$$G_c=\frac{(t_s-t_{ca})\Pi}{r}\int_0^x \frac{dx}{\frac{1}{\alpha_{rem}}+\frac{\delta_w}{\lambda_w}+\frac{\delta_\Phi}{\lambda_e}+\frac{\delta}{\lambda'}}. \tag{6}$$

Considering Eqs. (3), (4) and (6) simultaneously, we obtain

$$\frac{(\rho'-\rho'')\Pi g\sin\varphi}{3\nu'}\delta^3=\frac{\Pi(t_s-t_{ca})}{r}\int_0^x\frac{dx}{\Sigma R_i+\frac{\delta}{\lambda'}}-\frac{K_\Phi F_\Phi(\rho'-\rho'')g\sin\varphi}{\nu'}. \tag{7}$$

The rate of growth of the flowing condensate film is

$$\frac{dx}{\Sigma R_i+\frac{\delta}{\lambda'}}=\frac{r}{t_s-t_{ca}}\,\frac{(\rho'-\rho'')g\sin\varphi}{\nu'}\,\delta^2 d\delta. \tag{8}$$

To find how the film thickness δ depends on the coordinate we integrate Eq. (8) and determine the constant of integration from the condition $\delta = 0$ for $x = x_0$. We obtain

$$\frac{\delta^4}{4\lambda'}+\frac{\delta^3}{3}\Sigma R_i=\frac{t_s-t_{ca}}{r}\,\frac{\nu'}{(\rho'-\rho'')g\sin\varphi}x+C, \tag{9}$$

$$C=-\frac{t_s-t_{ca}}{r}\,\frac{\nu'}{(\rho'-\rho'')g\sin\varphi}x_0. \tag{10}$$

Substituting expression (10) into Eq. (9), we have

$$\frac{\delta^4}{4\lambda'}+\frac{\delta^3}{3}\Sigma R_i=\frac{t_s-t_{ca}}{r}\,\frac{\nu'}{(\rho'-\rho'')\,g\sin\varphi}\,(x-x_0). \tag{11}$$

The quantity x_0 is the coordinate of the point of departure of the film to the surface of the capillary structure, determined from the condition of equality of G_c and G_ϕ:

$$x_0=\frac{K_\phi F_\phi(\rho'-\rho'')\,g\sin\varphi}{\nu'}\,\frac{r\Sigma R_i}{(t_s-t_{ca})\,\Pi}. \tag{12}$$

Having obtained the dependence $\delta = f(x)$ from Eq. (11) and taking account of Eq. (12), we can determine the amount of heat transferred from Eq. (5). The average heat removal coefficient over the plate is found from the equation

$$\frac{(t_s-t_{ca})\,F}{Q}-\frac{1}{\alpha_{rem}}-\frac{\delta}{\lambda_w}=\frac{1}{\bar{\alpha}_c}. \tag{13}$$

If we take into account that the condensation section of actual heat pipes is practically isothermal, we can simplify the problem substantially, by considering only the process of heat removal with condensation of vapor in conditions of constant temperature of the cooled wall. Then we rewrite Eq. (8) as

$$\frac{(\rho'-\rho'')\,g\sin\varphi}{\nu'}\,\delta^2 d\delta=\frac{1}{r}\,\frac{\Delta t}{\dfrac{\delta_\phi}{\lambda_e}+\dfrac{\delta}{\lambda'}}\,dx, \tag{14}$$

and Eqs. (10) and (11) as

$$\frac{\delta^4}{4\lambda'}+\frac{\delta^3}{3\lambda_e}\,\delta_\phi=\frac{\Delta t}{B}\,(x-x_0), \tag{15}$$

$$x_0=\frac{K_\phi\delta_\phi^2 B}{\lambda_e\Delta t}, \tag{16}$$

where

$$B\equiv\frac{r(\rho'-\rho'')\,g\sin\varphi}{\nu'};\quad \Delta t\equiv t_s-t_w.$$

With $\delta_\phi = 0$ Eq. (15) converts to the known Nusselt relation for the condensate film thickness. An analogous relation for the thickness of the flowing film was obtained in [6]. By solving Eq. (15) and determining the relation $\delta = f(x)$, we find the value of the average heat removal coefficient for condensation on a capillary structure under the assumptions made:

$$\bar{\alpha}_c=\frac{1}{L}\left[\frac{\lambda_e}{\delta_\phi}\,x_0+\frac{L-x_0}{\dfrac{\delta_\phi}{\lambda_e}+\dfrac{\delta}{\lambda'}}\right]. \tag{17}$$

Figures 2 and 3 compare the calculated average heat removal coefficients from Eqs. (15) and (17) with the direct experimental data. It can be seen that there is satisfactory agreement between the calculated and experimental values (allowing for the approximation of the assumed model). The existing discrepancies are due to the large uncertainty in the values of λ_e and K_ϕ (according to the data of [1], λ_e = 0.8-4.5 for mesh capillary structures of stainless steel, and values of K_ϕ differ by a factor of two), possible depression of the liquid meniscus in the section $x < x_0$, failure to account for the heat of supercooling of the condensate and for the dependence of the thermophysical properties of the liquid on temperature.

NOTATION

R_c, thermal resistance of the condensate, $m^2 \cdot K/W$; δ_ϕ, thickness of the capillary structure, m; λ_e, equivalent thermal conductivity of the capillary structure, saturated with liquid, $W/(m \cdot K)$; μ', dynamic viscosity, $N/m^2 \cdot sec$; K_ϕ, permeability of the capillary structure, m^2; ρ', density of the liquid, kg/m^3; F_ϕ, area of cross section of the capillary structure, m^2; G_ϕ, flow rate of condensate, flowing in the capillary structure, kg/sec; G_f, flow rate of condensate in the flowing film, kg/sec; ϕ, slope angle of the heat-transfer surface to the horizontal, deg; δ, thickness of the flowing condensate film, m; ν', kinematic viscosity, m^2/sec; G_c, flow rate of condensate, kg/sec; t_s, saturation temperature of the heat-transfer agent, C; t_{ca}, temperature of the cooling agent, C; α_{rem}, coefficient of heat removal from the heat exchange surface to the cooling agent, $W/(m^2 \cdot K)$; δ_w, thickness of the heat transfer surface wall, m; λ_w, thermal conductivity of the wall material, $W/(m \cdot K)$; λ', thermal conductivity of the condensate, $W/(m \cdot K)$; x, longitudinal coordinate, m; x_0, coordinate of exit of the condensate film to the surface of the capillary structure, m; r, latent heat of vaporization, J/kg; Q, heat flux, W; L, length of the condensation section, m; Π, width of the condensation zone, m; ρ'', vapor density, kg/m^3.

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