HEAT TRANSFER CAPACITY OF CHANNEL HEAT PIPES WITH TRIANGULAR CAPILLARY STRUCTURE

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The flow of a liquid heat carrier is examined in a channel heat pipe. A dependence of the limit thermal fluxes on the regime parameters and the geometry of the capillaries is obtained.

In recent years the domain of heat pipe (HP) application has expanded significantly. The demands imposed on it have also become more diverse. For certain HP the magnitude of the motive forces, including the capillary forces also, can change substantially depending on the geometry of the wick structure, the wetting angle, and the physical properties of the heat carrier.

Channel HP that have axial or spiral capillaries of different cross section as wick can produce considerable capillary pressure and have low hydraulic drag. From this viewpoint, capillaries of triangular cross section are one of the promising wick structures in which the hydrodynamics and heat transfer have been investigated in a number of papers [1-4]. However, until now there have not been either experimental or theoretical data permitting the determination of the operating and limit characteristics of channel HP with triangular capillary profile and the optimization of the channel structure.

At this time one- and two-dimensional models are known by which the characteristics of channel HP can be computed [1-4]. The two-dimensional model proposed in [1] takes into account the change in the drag coefficient as a function of the wetting angle and the aperture half-angle of the capillary. However, the model describes just fully developed flow of the heat carrier under the effect of the gravitational forces. The radius of curvature of the fluid meniscus is constant along the length. A change occurs in the radius of curvature of the meniscus along the HP length in real HP. This circumstance is taken into account in the one-dimensional models proposed in [2-4]. On the basis of the assumption on the equality of gradients of the capillary and hydraulic pressures, the evaporation phase of a channel HP is considered for boundary conditions of the third [2, 3] and second kind [4]. The proposed dependences to determine the maximum heat-flux density are valid only in the absence of gravitation. Moreover, a factor determined from experimental data is contained in the expression for the maximum heat-flux density in [3].

The model proposed in this paper permits computation of HP characteristics with the action of inertial forces and gravitation on the heat carrier taken into account. The method is based on the following assumptions: 1) the capillary cross section has a regular triangular shape with apex angle 2φ; 2) the radius of curvature R of the fluid meniscus is constant over the capillary cross section; 3) the heat elimination and heat delivery are realized for boundary conditions of the second kind ($q = \text{const}$); consequently, the evaporation and condensation rates are constant along the length of the evaporator and condenser and are known in the formulation of the problem.

The whole length of the HP is provisionally divided into two sections: the first section is the angle of contact between the fluid and the solid wall $\psi(z)$ changing from $(\pi/2 - \varphi)$ (i.e., a plane meniscus) to the wetting angle α characteristic for this heat carrier—wall material pair. At the end of this section, the magnitude of the wetted wall r_w equals the capillary wall height r_{w0}. The second section is the angle of contact between the fluid and solid wall and the wetting angle α being equal and constant, while the magnitude of the wetted capillary wall r_w changes along the length of the section. It is assumed in the analysis that the thermophysical properties of the heat carrier are constant along the HP, while the tangential interaction between the fluid and vapor is slight.

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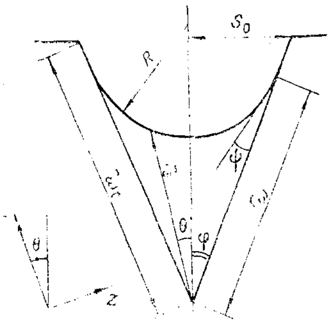

Fig. 1. Geometry of a triangular capillary.

For a fluid volume element in a HP capillary operating in the evaporation mode, the integral equation for momentum and mass conservation in a cylindrical coordinate system (Fig. 1) has the form

$$2\frac{\partial}{\partial \bar{z}}\int_0^{\varphi}\int_0^{\bar{r}_s}\rho_L \bar{W}^2\bar{r}d\bar{r}d\theta = -2\int_0^{\bar{r}_w}\frac{\mu_L}{\bar{r}}\left(\frac{\partial \bar{W}}{\partial\theta}\right)_{\theta=\varphi}d\bar{r} + \rho_L g\sin\gamma\frac{\partial}{\partial\bar{z}}(\bar{z}F) + \sigma\frac{\partial}{\partial\bar{z}}\left(\frac{F}{R}\right); \tag{1}$$

$$\bar{\Phi}(z) = \rho_L\int_0^{\varphi}\int_0^{\bar{r}_s}\bar{W}\bar{r}d\bar{r}d\theta, \tag{2}$$

where $\bar{\Phi}(z) = \rho_V V_e S_0(L_e - \bar{z})$ for the evaporation zone, $\bar{\Phi}(z) = \rho_V V_c S_0 \bar{z}$ for the condensation zone, and $\bar{\Phi}(z) = \rho_V V_c S_0 L_c$ for the adiabatic zone.

The first term of (1) takes account of the change in momentum of a fluid volume element under the effect of inertial forces, the second is the momentum loss because of friction on the capillary wall, the third because of gravitation, and the fourth term takes account of the change in momentum because of the change in capillary pressure.

To reduce (1) and (2) to dimensionless form, we consider that $r_w = r_{w_0}\xi(z)$, where $\xi(z)$ is the dimensionless function of the deepening of the fluid meniscus along the HP length, and that

$$r = \frac{\bar{r}}{S_0},\quad r_w = \frac{\bar{r}_{w0}}{S_0},\quad z = \frac{\bar{z}}{L},\quad W = \frac{\bar{W}}{W_0}. \tag{3}$$

From the geometric relationships, the radius of the fluid meniscus is determined as

$$r_s = r_{w0}\xi(z)A, \tag{4}$$

where the parameter A takes account of the dependence of r_s on the half-angle φ of the capillary and the angle of fluid contact with the wall material $\psi(z)$ and equals

$$A = \frac{\cos\psi\cos\varphi - \sqrt{\sin^2\varphi - \sin^2\theta\cos^2\psi}}{\cos(\psi+\varphi)}. \tag{5}$$

In dimensionless form the integral equations of mass and momentum conservation have the form

$$\Phi(z) = \frac{S_0}{L}\int_0^{\varphi}\int_0^{r_s} rWdrd\theta, \tag{6}$$

$$\frac{\partial}{\partial z}\int_0^{\varphi}\int_0^{r_s} rW^2drd\theta = -\frac{1}{\mathrm{Re}}\int_0^{r_w}\frac{1}{r}\left(\frac{\partial W}{\partial\theta}\right)_{\theta=\varphi}dr + C\,\mathrm{Fr}\frac{\partial}{\partial z}(zr_w^2) + \frac{B}{\mathrm{We}}\frac{\partial}{\partial z}(r_w). \tag{7}$$

Here $B = \cos\varphi\cos(\psi+\varphi) - \frac{\sin\varphi}{\cos(\psi+\varphi)}(\pi/2 - \psi - \varphi) + \sin\varphi\sin(\psi+\varphi)$, $C = B\frac{\sin\varphi}{\cos(\psi+\varphi)}$.

For the different HP sections — evaporation, condensation, adiabatic — the function $\Phi(z)$ in (7) has the following expressions:

$$\Phi(z) = 1 - z; \quad \Phi(z) = z; \quad \Phi(z) = 1. \tag{8}$$

The value of W_0 is determined from (7) as $W_0 = (\rho v/\rho L)V_e$ for the evaporation zone and $W_0 = (\rho v/\rho L)V_c$ for the adiabatic and condensation zones.

The solution of system (6), (7) is sought by the Karman–Pohlhausen method with the longitudinal velocity approximated by the expression

$$W(r,\ \theta,\ z) = r^2\left(\frac{\cos\theta}{\cos\psi} - 1\right) f(z), \tag{9}$$

which satisfies the boundary conditions

$$W(r,\ \theta,\ z)|_{\theta=\varphi} = 0; \quad \left.\frac{\partial W}{\partial\theta}\right|_{\theta=0} = 0. \tag{10}$$

Nonlinear first-order differential equations in the angle of contact $\psi(z)$ and the dimensionless magnitude of the meniscus deepening $\xi(z)$ are obtained from the joint solution of (6), (7), and (9):

$$\xi'(z) = \frac{(N-K)z\xi(z) - D\xi(z)^5}{Nz^2 + 2Dz\xi(z)^4 + E\xi(z)^3}, \tag{11}$$

$$\psi'(z) = \frac{(K-N)z + D}{-Mz^2 + Pz - R} \tag{12}$$

in the condensation zone;

$$\xi'(z) = \frac{K\xi(z) + D\xi(z)^5}{E\xi(z)^3 + 2Dz\xi(z)^4 + N}, \tag{13}$$

$$\psi'(z) = \frac{K+D}{Pz + M + R} \tag{14}$$

in the adiabatic zone:

$$\xi'(z) = \frac{(M+N)(1-z)\xi(z) + P\xi(z)^5}{E\xi(z)^3 + 2Pz\xi(z)^4 + 4N(1-z)^2}, \tag{15}$$

$$\psi'(z) = \frac{(K+N)(1-z) + D}{M(z-1)^2 + Pz + R} \tag{16}$$

in the evaporation zone, where

$$T = \frac{\cos\theta}{\cos\varphi} - 1;$$

$$K = \frac{2\,\mathrm{tg}\,\varphi}{\mathrm{Re}}\left(\frac{L}{S_0 r_{w0}}\right)^2 \frac{1}{\int\limits_0^{\varphi} TA^4 d\theta};$$

$$N = \frac{16}{3}\ \frac{L^2}{S_0^2 r_{w0}^8}\ \frac{\int\limits_0^{\varphi} T^2A^6 d\theta}{\int\limits_0^{\varphi} T^2A^8 d\theta};$$

$$D = \frac{\mathrm{Fr}\, r_{w0}^2}{2}\left(\frac{1}{2}\sin 2\varphi - \sin^2\varphi\left(\frac{\pi/2 - \varphi - \psi}{\cos^2(\varphi+\psi)} - \mathrm{tg}(\varphi+\psi)\right)\right).$$

$$M = \frac{64}{3}\ \frac{L^2}{S_0^2 r_{w0}^8}\ \frac{\int\limits_0^{\varphi} TA^3A' d\theta \int\limits_0^{\varphi} T^2A^6 d\theta}{\int\limits_0^{\varphi} T^3A^{12} d\theta};$$

$$P = \mathrm{Fr}\, r_{w0}\sin^2\varphi\left(\frac{(\pi/2 - \varphi - \psi)\sin(\varphi+\psi)}{\cos^2(\psi+\varphi)} - \frac{2}{\cos(\psi+\varphi)}\right);$$

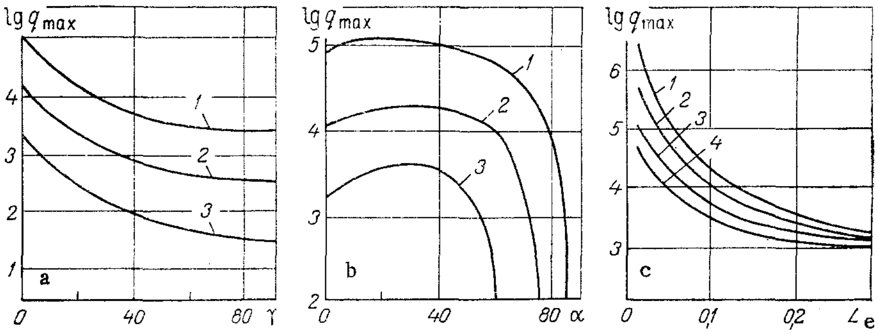

Fig. 2. Dependence of the maximal heat flux density q_{max} (W/m^2) on the slope γ (deg) of the heat pipe (a), the wetting angle α (deg) (b), the evaporation length L_e (m) (c): a) $L_e = L_c = 0.1$ m, $L_a = 0.05$ m, $S_0 = 0.3 \cdot 10^{-3}$ m, $\alpha = 0°$; 1) $\varphi = 5$; 2) 15; 3) 30°; b) $L_e = L_c = 0.1$ m, $L_a = 0.05$ m, $S_0 = 0.3 \cdot 10^{-3}$ m, $\gamma = 0°$; 1) $\varphi = 5$; 2) 15; 3) 30°; c) $S_0 = 0.3 \cdot 10^{-3}$ m, $\alpha = 0°$, $\varphi = 15°$, $L_e = L_c$; 1) $L_a = 0$; 2) 0.05; 3) 0.25; 4) 0.5 m.

$$R = \frac{r_{w0}}{\mathrm{We}}\left(\frac{\sin\varphi}{\cos(\varphi+\psi)} - \frac{\sin\varphi\,(\pi/2-\varphi-\psi)\sin(\varphi+\psi)}{\cos(\psi+\varphi)} - \sin\varphi\right);$$

$$F = \frac{r_{w0}}{\mathrm{We}}\left(\cos\varphi\cos(\varphi+\psi) - \frac{(\pi/2-\varphi-\psi)\sin\varphi}{\cos(\varphi+\psi)} + \sin\varphi\sin(\varphi+\psi)\right).$$

As a result of solving (12), (14), (16) it was obtained that the section of a change in the contact angle $\psi(z)$ for limit heat fluxes and the work against gravity is quite insignificant (about 2-6% of L_c) and it need not be taken into account in the computations. Equations (11), (13), (15) were solved numerically by the method of trial and error under the following boundary conditions: no fluid overflow in the condensation zone, and for $z = 0$, $\xi(z) = 1$, $\psi(z) = \alpha$; there is no "dry wall" regime at the end of the evaporation zone, and for $z = 1$, $\xi(z) = 0.05$.

The results of a computation, presented in Fig. 2a, show the substantial influence of the HP slope on the maximum heat-flux density q_{max} transmitted by the HP: it is substantially nonlinear in nature.

It is ordinarily assumed in theoretical computations that the wetting angle α, equal to zero, is best from the viewpoint of producing the greatest possible capillary pressure. However, computations did not confirm this (see Fig. 2b). The existence of maxima is explained by the fact that as the wetting angle increases there is an optimal relationship between the diminishing capillary pressure and the increasing through section of the capillary, i.e., the diminishing hydraulic drag to the fluid flow. As the capillary half-angle increases, the nature of the maxima becomes more and more definite. This circumstance can be used in selecting the wall material—heat carrier pair for any given geometry of the capillary.

The question of the influence of the length of the adiabatic zone on the HP heat-transmitting capacity has received no attention in the literature until now. Computations show (Fig. 2c) that, for sufficiently long evaporators, the adiabatic zone length has negligible influence on the maximal heat-flux density. The adiabatic zone exerts the most substantial influence on short (about 1 cm) evaporators, since namely there the greatest pressure losses in the fluid appear.

Computations of the velocity profiles showed that on a major portion of the HP, with the exception of the end of the evaporator, $\bar{z} = (0.5\text{-}0.95)L_e$, the mean value of the longitudinal velocity $\tilde{W}$ is 0.2-0.5 m/sec for HP capillaries operating at the maximal heat load, and around 0.1 m/sec for subcritical heat fluxes, i.e., the inertial component is here insignificant and about 4-8% of the moving capillary pressure. At the tip portion of the evaporation zone the mean fluid velocity can reach several tens of meters per second for a HP mode corresponding to the limit. The inertial component here reaches 50-60% of the capillary pressure. In the case of thermal loads less than q_{max}, the change in the mean velocity of fluid motion along the HP length becomes less substantial. Thus, for $\xi(z) = 0.3$ and $\bar{z} = 0.9L_e$ the mean velocity of fluid motion in the evaporator is 1 m/sec. Analysis of the results obtained permits making a deduction about the necessity of taking account of the inertial motion component only in the evaporation zone of the HP, where heat-carrier acceleration and deceleration phases exist.

NOTATION

r, θ, z, coordinates in a cylindrical coordinate system; φ, capillary half-angle; r_w, magnitude of the capillary wall wetted by the fluid; r_s, radius of the fluid meniscus; r_{w0}, height of the capillary wall; S_0, capillar half-width; L, length of the heat pipe zone; R, radius of curvature of the fluid meniscus; F, capillary through section; ψ(z), fluid contact angle with the wall material; α, wetting angle; W_0, axial velocity scale factor; W, axial velocity of heat carrier motion; γ, HP slope; V, evaporation or condensation rate; ξ(z), a dimensionless function of the fluid meniscus deepening; ρ, density; g, free-fall acceleration; σ, surface tension coefficient; μ, dynamic viscosity; q, heat flux density; Fr, Re, We, Froude, Reynolds, and Weber numbers. Subscripts: e, evaporator; a, adiabatic zone, c, condenser; v, vapor; L, fluid; max, maximal.

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CRISIS PHENOMENA UPON EVAPORATION IN GRID CAPILLARY AND POROUS COATINGS AND ARTERIAL STRUCTURES OF HEAT PIPES

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The article presents the results of experimental investigations of critical (limit) heat fluxes upon evaporation on porous coatings, and it substantiates the physical model of the process.

When the density of heat fluxes in porous structures is high, processes of evaporation may be accompanied by abrupt infringements of the conditions of heat transfer, leading to a sudden and substantial increase of the wall temperature. Such phenomena are called crisis phenomena.

According to the conditions of heat liberation and supply of heat carrier, the following situations may be distinguished.

1. Heat is conducted from the wall to the wetted structure: a) with capillary supply of the heat carrier, b) by inundation, c) with combined supply.

2. Heat is conducted from the skeleton to the heat carrier (internal heat liberation): a) under conditions of capillary supply; b) by inundation; c) with supply of liquid under the effect of pressure forces.

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