

HEAT TRANSFER WITH EVAPORATION AND BOILING OF
LIQUID IN THE CHANNELS OF EVAPORATORS. I.

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The authors consider laws of heat removal with evaporation and boiling of liquid in the channels of evaporators when power is supplied at boundary conditions of the third kind.

Widespread use has been made recently in heat pipes of evaporators in the form of capillary structures with screwlike channels. To calculate the parameters of these heat pipes one must know the maximum heat flux density which the channel can achieve. This maximum value depends on the channel profile, its length, width and depth, and for triangular channels, the vertex angle, and also the material, the cleanliness of fabrication, the orientation of the evaporator relative to the gravitational field, the properties of the working liquid, and also the boundary conditions at the outer surface of the pipe.

Literature [1-3] mainly contains a description of the heat-transfer process in channels when the power is supplied at boundary conditions of the second kind. In [2] the governing equation was given for calculating the average liquid velocity in the channel and it was used to calculate the total heat flux.

In [1], from equating the change in hydraulic and capillary pressures in liquid flow in a triangular channel, an equation was obtained for calculating the maximum heat flux density. The calculation is compared with data obtained from tests of four heat pipes in which the evaporator channels have a different vertex angle. The calculated values being larger than the experimental was explained by a restriction on transport of liquid in the axial direction of the pipe.

In [3] Fel'dman, using this model, improved the computational formula for a triangular channel and derived it for a rectangular channel. He proposed a technique for determining the heat-transfer coefficient, using a finite-difference calculation of the temperature drop through the channel wall and the liquid layer.

In [1, 3] the liquid meniscus in the transverse direction was taken as the moving meniscus in the computational model. It was assumed that full wetting occurred. However, for incomplete wetting, when $\theta = 90 - \alpha$, the transverse meniscus has a planar form and the analytical model loses its meaning. In [2] the so-called equivalent meniscus was taken, having curvature at any wetting angle, in addition to $\theta = 90^\circ$.

We have attempted to determine experimentally the influence of the evaporator channel parameters on the heat removal, with power supplied at boundary conditions of the third kind [4]. This work differs from the previous [1-3] in that the investigations were made in an experimental facility in which one could:

- a) supply heat to the evaporator at boundary conditions of the third kind;
- b) limit heat removal by processes occurring in the evaporator channels;
- c) determine experimentally the influence on the maximum heat removal of the geometric parameters of the channels of triangular and rectangular profile.

1. Experimental Facility. The facility (Fig. 1) comprised a vacuum oven ($D_{int} = 0.45$ m; $h = 0.65$ m), the evaporator element 10, the working liquid supply system, the condenser 3 and the thermal screen 2.

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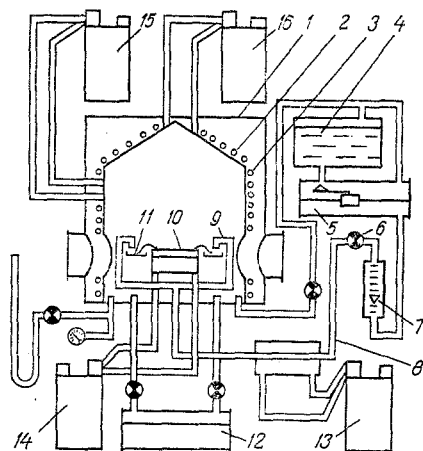


Fig. 1

Fig. 1. Arrangement of the experimental facility.

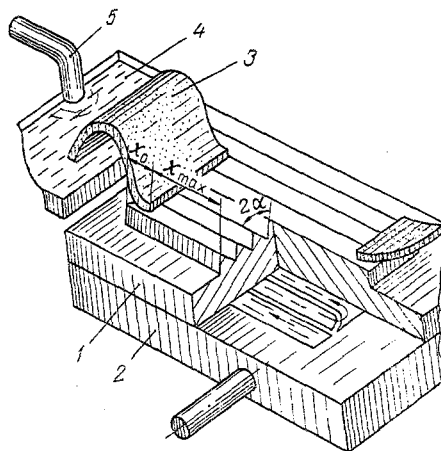


Fig. 2

Fig. 2. The evaporator unit.

The evaporator unit (Fig. 2) consists of the evaporator channel itself 1, the liquid heat exchanger 2, the porous arteries 3, and the two tanks 4 into which the liquid is poured from the branch pipes 5. The length of the evaporator is 0.1 m, and its width is 0.05 m. Its position in space is horizontal. Because of the symmetry in the location of the arteries the length of the evaporator is arbitrarily divided into two equal sections, each of length about 0.05 m. The initial part of the section of length $x_0 = 0.01$ m makes contact with a porous artery (a bundle of stainless steel mesh), and the remaining part $x_{\max} = 0.04$ m has its surface open for evaporation or boiling of liquid.

The condenser 3 (Fig. 1) is a thin-walled copper cylinder carrying coils for the flow of the cooling agent. The thermal screen 2 is cone-shaped, and also has a coil on its outer side.

The working liquid supply system includes the reservoir vessel 4, a volume with a system for maintaining constant liquid head 5, a rotameter 7, a valve 6, a liquid heat exchanger 8 and a tap 9. The liquid reaches the volume 5 from reservoir 4, whence it goes to the tanks via the rotameter 7 which measures the flow rate and the valve 6 which controls it, and the branch pipes 3. With the aid of heat exchanger 8 the temperature of the working liquid is kept equal to that of condensation. From the tanks the liquid is supplied along the porous arteries, under the action of capillary forces, to the evaporator channels, where the process of vaporization or boiling occurs. The vapor is condensed on the surface of the cylinder 3 and flows into the overflow vessel 12 (Fig. 1). The thermal screen 2 does not allow the vapor to condense upwards, in order to avoid drops falling into the evaporator. The heat-transfer agent reaches the liquid heat exchangers from the thermostats 13-16.

Acetone was used as the working liquid. It can be used over quite a wide temperature range (180 to 470°K). In addition, acetone wets metal well ($\cos \theta \approx 1$).

Before the start of the experiment the acetone was degassed by vacuum pumping. In order to minimize the influence of variation of properties of the working liquid on the heat transfer, the saturation pressure was kept practically constant in the experimental facility ($0.14 \cdot 10^5 - 0.145 \cdot 10^5$ N/m²).

As a developed heater surface we used brass plates with a system of triangular channels of different width (from $0.25 \cdot 10^{-3}$ to $0.8 \cdot 10^{-3}$ m), effective length (from $5 \cdot 10^{-3}$ to $40 \cdot 10^{-3}$) and vertex angle (from 15 to 90°), and also rectangular channels with different depth (from $0.2 \cdot 10^{-3}$ to $1.6 \cdot 10^{-3}$ m).

2. Variation of Heat Removal along a Channel. During the experiments we recorded the temperature field along the channel wall. The results of preliminary tests showed that the channel wall temperature increased with increasing distance from the place where the liquid was supplied. This indicates nonuniformity of heat removal along the channel.

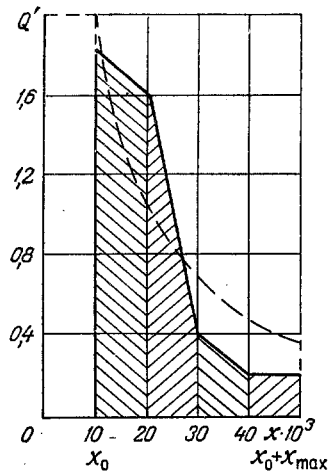


Fig. 3. Variation of the local heat flux along a channel: Q' , W/m; x_{\max} , m.

To determine the law for the variation of heat flux along the evaporator channel we divided its body arbitrarily into elemental sections for which a heat flux balance was written. The local heat flux was calculated by a finite difference method. It was assumed in the calculation that heat removal was only due to vaporization of liquid in the channels, and that there was no heat transfer between the wall and the vapor. The heat transfer at the boundary, the base of the evaporator and the heat-transfer agent in the liquid heat exchanger, is described by the Newton-Rikhman law

$$-\lambda \frac{\partial T(x, y)}{\partial y} \Big|_{y=0} = \alpha_c (T_l - T_w). \quad (1)$$

The results of the calculation are shown in Fig. 3. The area of the quadrilateral, multiplied by a scale factor, equals the total effective heat flux dissipated by the channel at the free line x_{\max} . The variation of heat flux along the channel can be represented with satisfactory accuracy (to 30%) by an elementary hyperbolic function. In this case the effective power is given by

$$Q = \int_{x_0}^{x_0+x_{\max}} Q'_0 \frac{x_0}{x} dx = Q'_0 x_0 \ln \frac{x_0+x_{\max}}{x_0}. \quad (2)$$

By dividing both sides of Eq. (2) by the latent heat of vaporization, we obtain the mass flow rate of working liquid flowing in the channel at section x_0 :

$$\dot{m} = \frac{Q'_0}{r^*} x_0 \ln \frac{x_0+x_{\max}}{x_0}. \quad (3)$$

3. Dimensions of the Liquid Film in the Channel. On the basis of experiments, Bresler [2] concluded that the moving meniscus in the channel is an equivalent liquid meniscus. Our analytical model is also based on this. In this case, for convenience of calculation, the arc of the transverse meniscus is arbitrarily replaced by a straight line, the so-called nominal width $t(x)$, i.e., one considers the cross section that the layer of liquid would have in the case of a planar meniscus.

3.1. Triangular Channel. To find the nominal width of the liquid layer $t(x)$ we need to know the volume of the channel not occupied by liquid. In the case when the supply artery brings considerably more liquid to the channel than the channel vaporizes, we can consider that the nominal width of the liquid layer coincides with the true width (Fig. 4):

$$t(x)_{\max}|_{x=x_0} = t.$$

If the maximum heat flux is achieved, then

$$t(x)_{\min}|_{x=x_0+x_{\max}} = 0.$$

For a hyperbolic type of heat removal, the maximum volume of liquid vaporized in the section from x_0 to x_0+x_{\max} is:

$$V_{\max} = \frac{Q'_0 x_0 \tau}{r^* \rho_{\text{ст}}} \left[\ln \frac{x_0+x_{\max}}{x_0} + \sum_{i=1}^m (m-i) \ln \frac{x_0+x_i}{x_0+x_{i-1}} \right]. \quad (4)$$

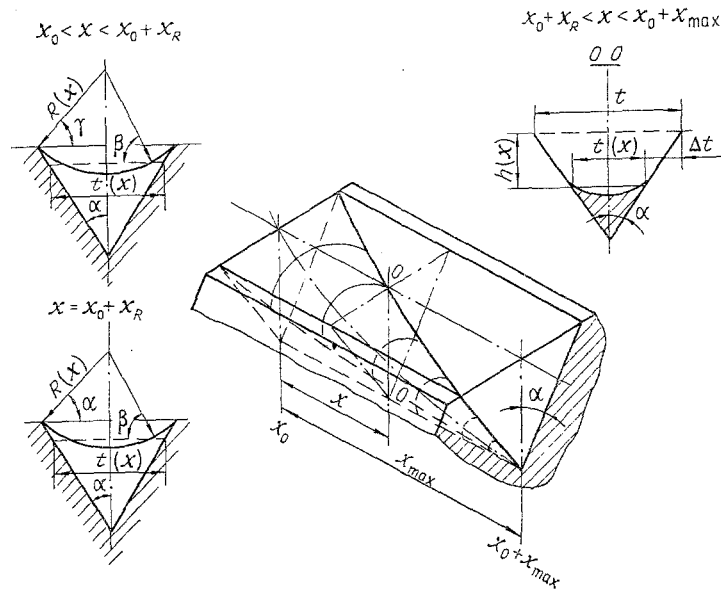


Fig. 4. Diagram to determine the dimensions of the liquid film at any section of a triangular channel.

On the other hand, this volume is

$$V_{\max} = K_p t^2 \operatorname{ctg} \alpha x_{\max}. \quad (5)$$

The volume of liquid vaporized in the section from x_0 to $x_0 + x$

$$V(x) = \frac{Q_0' x_0 \tau}{r^* \rho_l} \left[\ln \frac{x_0 + x}{x_0} + \sum_{i=1}^n (n-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}} \right]. \quad (6)$$

The same volume is

$$V(x) = K_p (t - 2\Delta t)^2 \operatorname{ctg} \alpha x. \quad (7)$$

By equating Eqs. (4) and (5) and (6) and (7) and separating out terms, we obtain

$$\Delta t = \frac{t}{2} \left[1 - \sqrt{\frac{x_{\max}}{x} \left[\frac{\ln \frac{x_0 + x}{x_0} + \sum_{i=1}^n (n-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}}{\ln \frac{x_0 + x_{\max}}{x_0} + \sum_{i=1}^m (m-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}} \right]} \right]. \quad (8)$$

The nominal width of the liquid layer at the section $x_0 + x$ is

$$t(x) = t \sqrt{\frac{x_{\max}}{x} \left[\frac{\ln \frac{x_0 + x}{x_0} + \sum_{i=1}^n (n-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}}{\ln \frac{x_0 + x_{\max}}{x_0} + \sum_{i=1}^m (m-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}} \right]}. \quad (9)$$

As has already been mentioned, when the supply artery brings considerably more liquid to the channel than the channel can vaporize, the transverse meniscus at the start of the channel (section x_0) can be considered planar. With increasing distance from this section the meniscus becomes curved until it becomes tangent to the wall of the channel at its cusps under complete wetting (section $x_0 + x_R$). The radius of this meniscus is

$$R_{\text{tr}}(x_R) = \frac{t}{2 \cos \alpha}. \quad (10)$$

The nominal width of the liquid layer at section $x_0 + x_R$ is

$$t(x) = t \left(\frac{1 - \sin \alpha \sin \beta}{\cos^2 \alpha} \right). \quad (11)$$

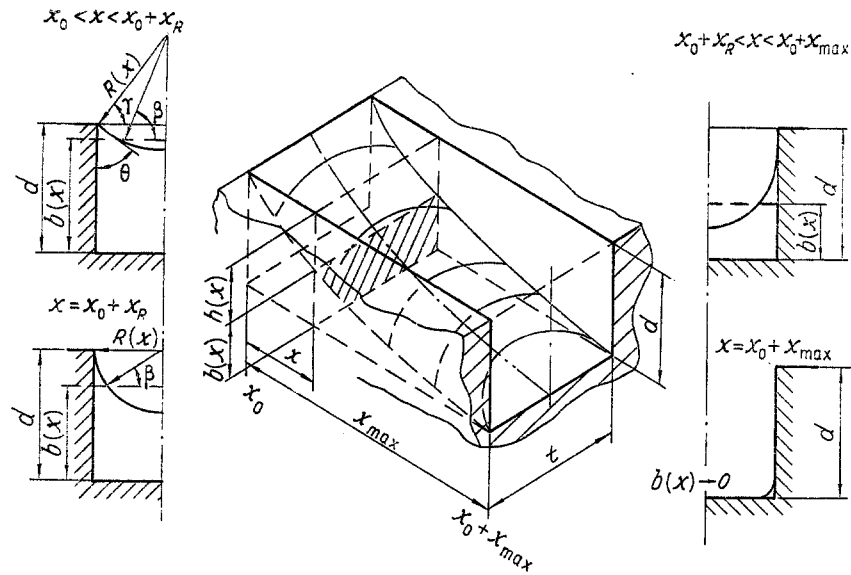


Fig. 5. Diagram for determining the dimensions of the liquid film at any section of a rectangular channel.

The angle β is defined from the equation

$$\frac{\operatorname{tg} \alpha}{\cos^2 \alpha} \sin^2 \beta - 2 \frac{\sin \beta}{\cos^3 \alpha} + \frac{1}{\cos^2 \alpha} \left(\frac{\pi}{2} - \alpha \right) + \operatorname{tg} \alpha + \operatorname{tg}^3 \alpha = 0. \quad (12)$$

By simultaneous solution of Eqs. (9) and (11) we find the value x_R .

The radius of the transverse meniscus at the section from x_0 to $x_0 + x_R$ is

$$R_{tr}(x) = \frac{t}{2 \cos \gamma(x)}. \quad (13)$$

The angle $\gamma(x)$ is determined from the following equations:

$$t(x) = t \left[1 - \frac{\sin \beta(x) \sin \alpha - \sin \gamma(x) \sin \alpha}{\cos \alpha \cos \gamma(x)} \right]; \quad (14)$$

$$\frac{\operatorname{tg} \alpha}{\cos^2 \gamma(x)} \sin^2 \beta(x) + \operatorname{tg} \gamma(x) + \frac{1}{\cos^2 \gamma(x)} \left[\frac{\pi}{2} - \gamma(x) \right] - 2 \sin \beta(x) \left[\frac{\cos \gamma(x) \operatorname{ctg} \alpha + \sin \gamma(x)}{\cos^2 \gamma(x) \operatorname{ctg} \alpha} \right] + \frac{\operatorname{tg}^2 \gamma(x)}{\operatorname{ctg} \alpha} = 0. \quad (15)$$

The wetting angle in the section from x_0 to $x_0 + x_R$

$$\theta(x) = \alpha - \gamma(x). \quad (16)$$

With full wetting in the section from $x_0 + x_R$ to $x_0 + x_{max}$ the arc of the transverse meniscus is tangent to the channel wall. Its radius is given by

$$R_{tr}(x) = \frac{t(x) \cos \alpha}{2(1 - \sin \alpha \sin \beta)}. \quad (17)$$

The capillary equivalent radius of the meniscus at any section of the channel [2], allowing for the curvature of the liquid layer in both the longitudinal and the transverse directions, is:

$$R_e(x) = \frac{A(x)}{\Pi_{wd}(x) - t(x)}; \quad (18)$$

$$R_e(x) = \frac{t(x) \cos \alpha}{2(1 - \sin \alpha) \cos \theta} = \frac{t(x) C_1(\alpha)}{\cos \theta}. \quad (19)$$

At the maximum heat removal, $R_e(x)$ varies from $tC_1(\alpha)$ at section x_0 to a value close to zero at section $x_0 + x_{max}$. Since we can always obtain the necessary drop in capillary pressure, because of this variation in the width of the liquid layer, we consider that the wetting angle for the longitudinal meniscus at the walls is constant ($\theta = \text{const}$).

The transverse flow area for liquid in the channel at section $x_0 + x$ (Fig. 4) is

$$A(x) = t^2(x) \frac{ctg \alpha}{4} = t^2(x) C_2(\alpha). \quad (20)$$

The hydraulic diameter is

$$D_h = \frac{4A(x)}{\Pi(x)}, \quad (21)$$

$$D_h = t(x) \frac{\cos \alpha}{(1 + \sin \alpha)} = t(x) C_3(\alpha). \quad (22)$$

3.2. Rectangular Channel. In a rectangular channel the nominal width of the liquid layer is equal to the channel width, i.e., $t(x) = t$. The distance from the cusp of the channel to the nominal width of liquid at section $x_0 + x$ is found analogously to $t(x)$ for the triangular channel (Fig. 5):

$$h(x) = d \frac{x_{\max}}{x} \left[\frac{\ln \frac{x_0 + x}{x_0} + \sum_{i=1}^n (n-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}}{\ln \frac{x_0 + x_{\max}}{x_0} + \sum_{i=1}^m (m-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}} \right]. \quad (23)$$

With full wetting in the section from x_0 to $x_0 + x_R$ the radius of the transverse meniscus varies from ∞ to $t/2$. We can find the value of $h(x)$ in this section from the equation

$$h(x) = \frac{t}{2 \cos \gamma(x)} [\sin \beta(x) - \sin \gamma(x)], \quad (24)$$

where the angle $\gamma(x)$ is given by:

$$[\sin \beta(x) - \sin \gamma(x)] \cos \gamma(x) - \left[\frac{\pi}{2} - \gamma(x) \right] + \cos \gamma(x) \sin \gamma(x) = 0. \quad (25)$$

At the section $x_0 + x_R$ ($\gamma = 0$; $\beta = 45^\circ$)

$$h(x) = 0.353 t. \quad (26)$$

Substituting this value into Eq. (23), we obtain x_R . The capillary equivalent radius of the meniscus is

$$R_e(x) = \frac{t}{2 \cos \theta}. \quad (27)$$

It can be seen that the required capillary pressure is achieved only in the case when the boundary angle varies. We consider that

$$\theta|_{x=x_0} = 90^\circ, \quad \theta|_{x=x_0+x_{\max}} = 0^\circ.$$

Then we have

$$R_e(x)|_{x=x_0} = \infty; \\ R_e(x)|_{x=x_0+x_{\max}} = \frac{t}{2}.$$

The area of the transverse liquid stream at section $x_0 + x$ (Fig. 5) is

$$A(x) = tb(x). \quad (28)$$

The hydraulic diameter is

$$D_h(x) = \frac{2tb(x)}{t + b(x)}, \quad (29)$$

where

$$b(x) = d \frac{dx_{\max}}{x} \left[\frac{\ln \frac{x_0 + x}{x_0} + \sum_{i=1}^n (n-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}}{\ln \frac{x_0 + x_{\max}}{x_0} + \sum_{i=1}^m (m-i) \ln \frac{x_0 + x_i}{x_0 + x_{i-1}}} \right]. \quad (30)$$

By determining the dimensions of the liquid film at any section of the channel for a given law of heat removal we can set up a physical model of the evaporator and derive a formula for calculating the maximum heat flux density achievable by the channel. By comparing the experimental and computed data, we can evaluate how correct was our choice of analytical model.

NOTATION

$A(x)$, area of transverse liquid layer in the channel at section x , m^2 ; $b(x)$, height of the liquid layer at section x , m ; D , diameter, m ; d , depth of channel, m ; $C_1(\alpha)$, a constant, defining $R(x)$; $C_2(\alpha)$, a constant defining $A(x)$; $h(x)$, height of the free channel surface at section x , m ; h , height of the vacuum volume, m ; K_p , coefficient accounting for the curvature of the vapor volume in the channel; m , the mass flux of liquid in the channel at section x , m^3/sec ; Q , heat flux, W ; Q'^0 , height of the hyperbola at the point x_0 , W/m ; $R(x)$, radius of the meniscus at section x , m ; r^* , latent heat of vaporization, J/kg ; T , temperature, K ; t , channel width, m ; $t(x)$, nominal width of liquid layer at section x ; V , volume of liquid evaporated in the channel, m^3 ; $V(x)$, volume of liquid evaporated in the channel at section x , m^3 ; Π , perimeter, m ; x, y , coordinates; α , half angle at the vertex of a triangular channel, deg ; α_e , heat-transfer coefficient, $W/m^2 \cdot K$; β , angle between the nominal width of the liquid layer and the meniscus radius, deg ; γ , angle between the meniscus radius and the straight line joining the cusps of the channel, deg ; θ , wetting angle, deg ; λ , thermal conductivity, $W/m \cdot K$; ρ , density, kg/m^3 ; τ , time, sec ; Subscripts: h , hydraulic; l , liquid; o , initial; max , maximum; min , minimum; int , internal; e , equivalent; tr , transverse; wd , wetted; x , index of channel section; x_R , index of channel section where the transverse meniscus is tangent to the walls; w , wall.

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