BOILING LIQUID HEAT TRANSFER ON POROUS AND DEVELOPED HEATER SURFACES

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UDC 536.423

The investigation of the process of liquid boiling on porous and developed heater surfaces is an interesting and important development for science and technology. From the physical viewpoint the process has a number of special features, compared with liquid boiling in a large enclosure, or in channels and tubes.

The importance of investigating the liquid boiling process in porous structures for technology is self-evident: At present various branches of industry are beginning to use vapor generators with porous coatings, heat pipes, cryogenic superconducting electrical machinery, LEDs, etc.

Because the parameters of porous structures (the thermal conductivity, porosity, pore size, permeability, capillary potential, electrokinetic effects, etc.) can vary over a wide range, heat transfer with liquid boiling in pores can occur with variable intensity.

One can single out the following directions for investigating the boiling liquid heat transfer process:

1) on a porous surface (immersed or nonimmersed), in a large enclosure, in channels, in tubes;

2) on a developed surface (immersed or nonimmersed);

3) inside porous bodies.

The heat transfer process with liquid boiling on porous and developed heat transfer surfaces has been investigated in very great detail. There has been little study of the heat transfer process with liquid boiling inside porous bodies, particularly with volume heat-generation (electrical heating of a body, radioactive radiation, etc.) [7, 8].

Boiling Liquid Heat Transfer on a Porous Surface. In 1970 Yagov and Labuntsov [1, 19] published a study analyzing conditions for vapor generation of roughened or porous heat transfer surfaces. The vapor generation depends on variation of the thermodynamic potential of the system

$$\Delta\Phi = (f_v - f_l)V\rho_v + \sigma A\left[1 - \frac{A_w}{A}(1 - \cos\theta)\right]. \quad (1)$$

The smaller is the increment of thermodynamic potential $\Delta\Phi$, the easier it is to form a vapor bubble. According to [2], the boiling process can be stabilized and the heat transfer process enhanced by reducing the heater surface area A and increasing the ratio A_w/A. This increase is achieved by going from a smooth surface to a porous or developed one (grooves, slots, etc.) [3, 40], and also by special surface treatment (hydrophobization, etching, etc.). A change in the wetting angle θ has an appreciable influence on the change $\Delta\Phi$. Processing of the heat transfer surface, particularly hydrophobization, can vary the wetting angle θ and considerably enhance the heat transfer [4].

Porous structures have a substantial stabilizing effect on the liquid boiling process, particularly at reduced pressure of the surrounding medium, when they smooth out the onset of critical boiling, and intensify heat transfer at low heat flux density. It was shown in [1] that in the heat flux range up to 250 kW/m^2, with boiling of water and ethanol on a metal surface covered by a Teflon grid with aperture size 1.5-2.5 mm, at a surrounding medium pressure of $16 \cdot 10^4$ N/m^2 the boiling process is stable, and the heat transfer is enhanced relative

A. V. Lykov Institute for Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR, Minsk. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 34, No. 4, pp. 741-761, April, 1978. Original article submitted May 16, 1977.

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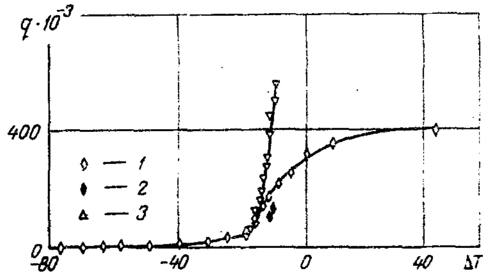

Fig. 1

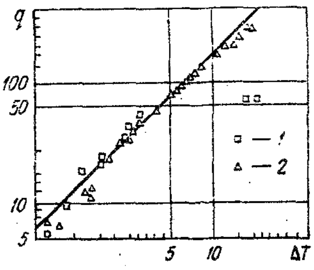

Fig. 2

Fig. 1. The limiting heat flux from the heated wall for water in the pores of a ceramic, sintered nickel powder; 1) surface with two grooves, increase of heat flux; 2) surface with two grooves, reduced heat flux; 3) smooth surface. $q \cdot 10^{-3}$, W/m$^{2\circ}$; ΔT, °K

Fig. 2. The heat flux q, W as a function of ΔT, °K with water boiling on a porous plate made of sintered wires: 1) plate in vertical position; 2) plate horizontal.

to boiling on a smooth surface. Of particular interest is the fact that boiling on a porous surface can occur with very small temperature differences $(T_w - T_{sat}) \approx 0.1$°K, and that it permits high heat flux transfer in thermodynamically convenient conditions.

Enhancement of the heat transfer process with boiling of cryogenic liquids (nitrogen, oxygen, etc.) on a metal heat transfer surface is achieved by a layer of hoarfrost, i.e., a porous coating of ice of higher-boiling point liquid (water, alcohol, etc.). It is noteworthy that metal bodies instantaneously immersed into a cryogenic liquid (nitrogen or oxygen) are cooled faster, if they are coated with a thin layer of a porous polymer or oil [20, 27].

In [21] very interesting results were presented on enhancement of heat transfer with boiling of liquid helium on metal surfaces, partially coated by a porous polymer, a layer of varnish, or a layer of rough particles of diameter 10-25 μm.

One observes an increase both in the maximum and the minimum values (the latter by a factor of 4) of heat flux in bubble boiling, compared with boiling of helium on a smooth surface.

Systematic investigations of the process of liquid boiling in porous media have begun with the appearance of heat pipes, whose evaporators were made of metal-ceramics, felt, mesh, or in the form of grooves [22]. Tests were made of porous metallic and nonmetallic structures (plates and tubes), vertically or horizontally located, and immersed in the liquid partially or totally.

Liquid to surface heat transfer occurred either by means of capillary forces, or by free convection. The test materials were water, organic liquids, and liquid metals.

Porous structures are systems with regular core-formation centers, on which bubble boiling sets in for considerably less heat flux, in comparison with a smooth surface. Since the effective thermal conductivity of most metal porous structures, saturated by nonmetallic liquids, exceeds the thermal conductivity of a layer of liquid of equivalent thickness, the temperature drop is less on the porous structure.

With boiling on a smooth surface, turbulent mixing occurs, while in a porous structure the capillary force field maintains laminar flow of liquid along the pores and capillaries [40]. At large heat flux the process of energy transfer through the porous structure is accomplished by heat conduction [26] and convection of liquid in the pores [24]. At higher heat fluxes core generation arises.

When special vapor-forming grooves are present in a porous body (perforations, grooves, etc.), the vapor bubbles leave the porous structure comparatively easily, which makes it possible to avoid critical boiling at very high heat flux levels (up to $1\text{-}1.3 \cdot 10^3$ kW/m^2 in water). Without vapor-generating channels the vapor bubbles accumulate in the zone between the heater surface and the porous structure and form a vapor film of thickness about equal to the size of the pores or particles, out of which the porous structure is sintered. With further increase in heat flux, heat is transferred by conduction through the vapor film to the saturated liquid in the pores [22, 24, 31]. Thus, with liquid boiling in a porous body the heat flux is not observed to depend on the temperature difference ($T_w - T_{sat}$) in the form of a Nukiyama curve (with two maxima), which is typical for boiling of liquid in a large enclosure over a smooth surface.

The dependence of α_e on ($T_w - T_{sat}$) in liquid boiling in a porous structure has the form of a convex curve with a slight maximum, whose value depends on the properties of the porous structure, and frequently α_e is constant over a wide range of variation of ($T_w - T_{sat}$).

This model of the boiling process is typical for nonmetallic liquids, which have thermal conductivity an order of magnitude less than that of liquid metals. It is known that the liquid must be heated to initiate the boiling. For a given heating of the liquid to temperature T one can create equilibrium bubbles of radius

$$R = \frac{2\sigma}{T - T_0} \cdot \frac{d\,T_{sat}(P)}{dP}. \tag{2}$$

Bubbles of smaller radius will decrease, and bubbles of greater radius will grow. The critical bubble size diminishes almost linearly with increase of the liquid pressure. For metals the critical bubble size is considerably larger than for nonmetallic liquids. Therefore the metal boiling process in a porous body is a more difficult one, compared with nonmetallic liquids.

The process of boiling of a liquid metal in porous structures has a number of special features, and is not considered here. The process of bubble boiling of water on a horizontal surface covered by a layer of Monel metal spheres has been observed by Ferrell et al. [25]. The thickness of the porous body varied from 3 to 25 mm. The liquid layer above the porous body was 7.5 cm. The authors concluded that the process of heat transfer to a saturated liquid occurs by heat conduction through the vapor layer to the junction between the porous body and the heated surface. Marto and Mosteller [22] observed the process of bubble boiling of water on the surface of a horizontal tube encased by four layers of stainless steel mesh. Since the outer container was made of glass, the process could be observed visually. The authors determined the dependence of heat flux on heating rate, and observed that the critical radius of bubbles was 0.013 mm, while the effective radius of the pores was 0.6 mm. A curious fact was that no perturbations could be seen with liquid boiling of the surface of a porous structure. Costello and Fry [28] investigated heat transfer with boiling of liquid in porous media and determined the critical heat fluxes. They concluded that the critical heat flux in a porous structure can exceed the critical heat flux on a smooth surface by 20%.

Kuntz et al. [29] investigated heat transfer with boiling of water and Freon 113 on metal-ceramic and felt made of nickel and stainless steel. The porous body had the form of plates of dimension 85 × 16 × 25 mm, attached to the heater, also made of nickel and stainless steel. The heater was covered electrolytically with Al_2O_3 powder by plasma deposition, which provided high thermal conductivity of the coating. The liquid was supplied to the porous plate through its side surface by capillary forces. The plate could be rotated around its axis.

The heat transfer process with boiling of liquid on a porous flat plate has been investigated with vapor-release grooves and without the grooves.

Figure 1 shows the relationship $q = f(\Delta T)$ for flat plates without grooves (400 kW/m^2) and with grooves (570 kW/m^2). The authors recommend determination of the critical heat flux as a function of the liquid properties using the formula

$$\frac{q_{crit}}{\rho_l r^*} = 143\, g^{1/4} \left(\frac{\rho_l - \rho_v}{\rho_l}\right)^{0,6} \text{m/sec}. \tag{3}$$

This formula describes the process of boiling of a liquid on a smooth surface in a large enclosure, but it may be used to estimate the efficiency of using different liquids.

Moss and Kelly [30] were probably the first to propose a model for heat transfer with liquid boiling in a porous body, where the heater surface is separated from the liquid by a thin film of vapor, of thickness about the size of the pores. They investigated the process of water boiling on a porous stainless steel flat plate of thickness 1 mm. They used a

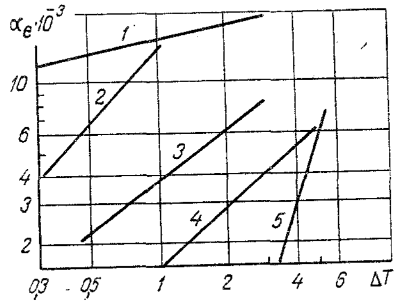

Fig. 3. Dependence of α_e on ΔT on various heater surfaces. The liquid is Freon-22: 1) coating obtained electrochemically; 2) sprayed powder; 3) baked powder; 4) finned tube; 5) smooth tube. $\alpha_e \cdot 10^{-3}$, $W/m^2 \cdot °K$; ΔT, °K.

television camera to measure the water film thickness. The pictures were taken by means of a flux of neutrons penetrating through the porous flat plate. Korman and Volmit [31] concluded that the heat transfer coefficient in liquid boiling does not depend on the thickness of the porous body, and that the liquid—vapor interface lies inside the porous body. Actually, the authors did not give data on the thickness of the porous plate and the typical pore size. A similar conclusion was drawn by Basiulus and Filler [32].

Systematic investigations of heat transfer with liquid boiling in porous media have been conducted at the University of South Carolina (USA) under the leadership of Ferrell [24-26].

Ferrell [23] carried out very interesting investigations of heat transfer with boiling of water and potassium in porous plates made of sintered Monel metal powder, mesh, and also metallic felt. Figure 2 shows the relationship between q and ΔT for water boiling in a porous stainless steel plate, sintered from wire ($\delta^* = 3.2$ mm; $\varepsilon = 0.58$; $K' = 0.55 \cdot 10^{10}$ m^2; $h_{max} = 0.26$ m).

The main conclusion of the study is as follows.

The heat transfer coefficient for liquid boiling in a porous body is a conservative quantity which depends very little on temperature and heat flux (below the critical level). The heat transfer process is distinguished by its stability, and for the same heat flux conditions the temperature drop ($T_w - T_{sat}$) in porous bodies is much less than for liquid boiling in a large enclosure. If one adopts the hypothesis that the vapor film above the heater surface is constant, then one can conclude that it is uniform, does not depend on the heat flux, and is determined by the action of capillary forces. The thickness of the vapor film is approximately that of the characteristic pore size.

By combining the equations of mass, energy, and momentum in a porous body, Ferrell et al. [23] obtained an equation for calculating the critical heat flux giving the best agreement with experimental data. The equation takes account of friction when the vapor passes out through the pores.

Direct measurements of the electrical resistance of a porous body saturated with water, with heat flux supplied and without it, has confirmed the hypothesis that there is a thin film of vapor between the heater surface and the porous structure. When a boiling liquid is present in a porous body the electrical potential is constant along the entire section, apart from a very thin layer near the surface where the porous body makes contact with the metal heater surface. With no boiling (q = 0) the electrical potential decreases linearly as a function of the porous body thickness.

The critical heat flux and the heat transfer coefficient with liquid boiling in a porous body which has a wide spectrum of pores of various sizes was a minimum in the case where the large pores contacted the heater surface, and a maximum where they contacted the small pores. Therefore, the size of the pores in the contact zone determines the thickness of the vapor film, and the heat transfer in porous bodies saturated by nonmetallic boiling liquids is carried by conduction through the thin vapor layer near the heater surface and by ejection of vapor through the large pores into the surrounding medium.

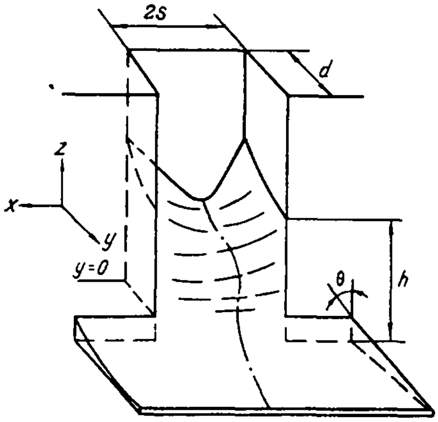

Fig. 4. The liquid—vapor interface z(x, y) in a capillary groove with square cross section.

The main limiting factor in boiling heat transfer is the limited capacity of the capillary force field to supply liquid to the heated zone:

$$q_{\text{crit}} = \frac{jr^*}{A_{\text{E}}}, \tag{4}$$

where A_e is the evaporator area.

For a flat heat transfer surface and a homogeneous porous wick Ferrell [23] has proposed a formula for determining q_{crit}:

$$q_{\text{crit}} = \frac{g\left[h_{\max}(T_1)\frac{\sigma_l}{\sigma_l\ (T_1)}\ \rho_l\ (T_1) - \rho_l\ L\sin\gamma\right]}{\frac{L_{\text{E}}\mu_l}{\rho_l\ K'r^*\varepsilon b}\left(\frac{L_{\text{E}}}{2} + L_{\text{a}}\right)}, \tag{5}$$

where L_E, L_a are the length of the evaporator and the adiabatic zone, and $h_{max}(T_1)$ is the maximum height of rise of the liquid at a given temperature.

Allowing for the thermal expansion of the porous body, Eq. (5) takes the form

$$q_{\text{crit}} = \frac{g\left[h_{\max}(T_1)\rho_l\ (T_1)\frac{\sigma_l}{\sigma_l\ (T_1)} - \rho_l\ L\sin\gamma\left[1 + \beta'\Delta T\right)\right]}{\frac{L_{\text{E}}\mu_l}{r^*\rho_l\ K'\varepsilon b(1 + \beta'\Delta T)}\left(\frac{L_{\text{E}}}{2} + L_{\text{a}}\right)}, \tag{6}$$

where $\Delta T = T - T_1$.

This equation agrees well with experimental data for water, right up to q_{crit} = 130 kW/m², and for potassium, q_{crit} = 315 kW/m².

For larger heat fluxes the limiting factors are such parameters as the reactive pressure of the vapor issuing from the pores, and so on.

The mechanism for the process of liquid boiling in porous bodies was examined theoretically in [6]. The authors considered the fact that, because the pores contain cavities with phase interfaces, the evaporation there will proceed with considerably less heating than for evaporation in a vapor bubble cavity. The high heat flux for small temperature differences during boiling in porous structures is explained by three factors:

1. the presence within the layer of a phase boundary which lowers the heating required for vapor generation;
2. a high coefficient of convective heat transfer in laminar flow of a liquid in the capillary channel;
3. the developed surface of the capillary structure.

The authors postulate that during boiling of a liquid on a porous surface there is a phase transition zone near the heater surface. The vapor moves away along the large pores, while the liquid is sucked towards the phase transition zone along the small pores by means of capillary forces. Thus, one vapor-removal channel requires m liquid channels.

The final formula for determining the relation $q = f(\Delta T)$ has the form

$$q = \frac{64.8(1-\varepsilon)}{D}\left[\left(\sqrt{\frac{1}{\varepsilon}}-1\right)\lambda_l\lambda_l\right]^{0.5}(\Delta T-\Delta T^*)\frac{m}{m+1};$$

$$m = 1.41\cdot10^{-3}\,\frac{\sigma r^*\rho_v\cos\theta}{\mu_v q_c}\left(\frac{D}{h}\right)^2\left(\frac{\varepsilon}{1-\varepsilon}\right)^3-\Delta m; \tag{7}$$

$$\Delta m = \frac{\mu_l\rho_v}{\mu_v\rho_l};\qquad q_c = \frac{\delta^*}{h}\lambda_{1r}\sqrt{\frac{2\alpha_e}{\delta^*\lambda_{1r}}}\Delta T_0;$$

$$\Delta T_0 = \Delta T-\Delta T^*;\qquad \Delta T^* = 9.75\,\frac{\sigma T_{sat}}{r^*\rho_v D}.$$

In the formula obtained by the authors there are some uncertainties. In particular, the quantity λ_{1r} can have values differing by an order of magnitude, depending on the nature of the grain contact, and friction is not taken into account in the motion of the vapor inside the porous body. In contrast with the investigation of heat transfer with liquid boiling inside porous structures in [7, 8], Danilova et al. [10] at the Leningrad Technological Institute of the Refrigeration Industry conducted analogous experiments, but with a layer of liquid present (Freon 11, 12, 22) above the porous surface (with immersed porous structure). The experiments encompassed the regions of evaporation, undeveloped boiling, and the beginning of developed boiling.

The porous coatings were formed by electrochemical deposition, spraying, and welding on the surface of copper and steel tubes (D = 20-25 mm). The volume porosity of the sprayed coating was ε = 16-60%, and the layer thickness was δ^*_{1r} = 75-580 μm. The coatings obtained by the welding method were laid down on a type 1Kh18N10T stainless steel tube (D = 20 mm), and were formed from powder of the same material with D = 63-250 μm (ε = 45-50%; δ^*_{lr} = 0.3-1 mm). The range of heat flux was q = 500-30,000 W/m². Visual observations showed that there was stable bubble boiling in the tubes with coatings, even for very small values of ΔT for which the free convection regime is observed on smooth and finned tubes.

Heat transfer in boiling in tubes with coatings is only slightly affected by heat flux density and pressure.

Figure 3 shows the dependence of α_e on ΔT for various heater surfaces. With a temperature drop of 1-2°K the investigators were able to obtain α_e = 16,500-183,000 W/m² on a tube with a porous coating, which is larger by about a factor of 6.5-10 than for a smooth tube or a finned tube. It should be noted that the relation $\alpha_e = f(\Delta T)$ in this case differs appreciably from α_e on a nonimmersed porous surface [23, 33, 34], where the heat transfer coefficient with liquid boiling in a porous structure behaves extremely conservatively and is almost constant over a wide range of variation of q and ΔT. A modification of the liquid boiling process in a porous structure above a heated surface is boiling of a liquid in the dispersed layer of solid particles flooded with the liquid [11]. This boiling model is intermediate between liquid boiling in a large enclosure above a smooth surface and liquid boiling in an immersed capillary-porous structure.

Under the influence of the vapor and liquid fluxes the solid particles are set in motion (thermal fluidization), which affects the hydrodynamics of the process and the heat transfer. Test data have been obtained in the range $q = 10^3$-$2\cdot10^5$ W/m², and pressure $(0.035-1)\cdot10^5$ N/m² for various heater surface materials (Kh18N10T, Nichrome, and copper) and particles (glass, aluminum silicate, mullite, and corundum), heater surface shape (plate and tube), and kind of liquid (water, ethanol, NaCl solution), disperse layer characteristics (particle diameter D = 0.05-10 mm, layer height h = 0-200 mm).

The experiments were conducted with particles whose thermal conductivity is close to that of the liquid.

This model of the process is interesting from the viewpoint of avoiding scale formation. Two boiling regimes were observed. For layer heights $h < h_{lim} = 6$ to $8\,(\sigma/g\rho_l)^{1/2}$ the layer

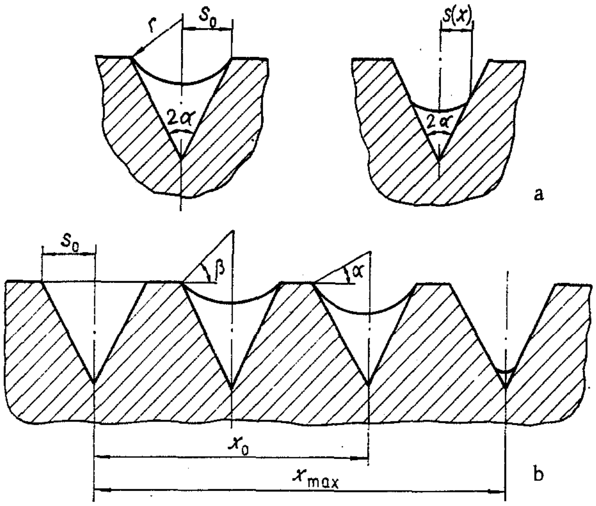

Fig. 5. Profile of the meniscus in a triangular groove: a) with $Q' = Q'_{max}$; b) with $Q' \ll Q'_{max}$.

particles are in a fixed state, and the vapor bubbles formed filter through the layer of porous material. This model is close to that considered by Danilova et al. [10]. For $h > h_{lim}$, $D \leq 2(\sigma/g\rho_l)^{1/2}$ at the heated surface, independent of q and P, according to the authors, a vapor–liquid sublayer is formed, from which the vapor passes away along individual vapor channels. This model is close to the boiling liquid model in a large enclosure above a smooth surface. For $D/(\sigma/g\rho_l)^{1/2} > 1$ the dispersed layer is fixed, and for $D/(\sigma/g\rho_l)^{1/2} < 0.2$ it is fully withdrawn from the heated surface. It should be noted that the influence of the geometric dimensions of the heater was not taken into account in [11], nor the density of the solid phase. Eight empirical constants are used in the formulas obtained. It can be assumed that the intensity of boiling heat transfer of an immersed layer of dispersed particles in the above thermal fluidization regime is greater than for liquid boiling in a large enclosure, but is lower than for boiling on a metallic capillary-porous surface. These results are from the experiments of [11].

Heat Transfer with Liquid Boiling on a Developed Surface. Boiling of liquid on a developed heat transfer surface (capillary grooves, a screw-type slot, fine-scale fins, slots) can take place both on an immersed and a nonimmersed surface (the liquid is in the groove). The latter case is widely employed in heat pipes and vapor chambers, when the liquid is supplied from the condenser to the evaporator along longitudinal grooves, or porous arteries, and flows along a screw-type groove under the influence of capillary forces. The heat transfer in the wetted grooves of the evaporator occurs either by evaporation of liquid from the film surface, or by boiling in the grooves. These systems are called thin-film evaporators.

The literature contains comparatively little information on methods for calculating thin-film evaporators and experimental investigation of their parameters. However, the available data [12-13] indicate that in these the heat transfer intensity can be as high as 10^3-10^4 W/m^2•°K for $T_w - T_{sat} < 1$°K.

The process of evaporation from thin films has been studied by Lastenader [35]. The thin films were formed on the inner surface of a vertical copper cylinder by means of spring-loaded carbon brushes mounted on a rotating bath. The working liquid was water. The experimentally obtained heat transfer coefficient was as high as $4.7 \cdot 10^4$ W/m^2•°K.

By solving the equations of continuity, momentum, and energy, and assuming a parabolic velocity profile for laminar flow, Van Sant [36] determined the heat transfer coefficient for evaporation of thin liquid films. He came to the conclusion that a high heat flux may be obtained with a low temperature drop during the process of evaporation from thin films.

Bresler and White [13] investigated the process of wetting of the surface by means of capillary grooves. They studied the influence of the size and shape of the grooves on the evaporation process. The authors suggested that in the evaporation process the heat is transferred through the liquid film by conduction and there is no bubble formation. The flow regime is laminar. They assumed that the solid body–liquid interface and the liquid–vapor

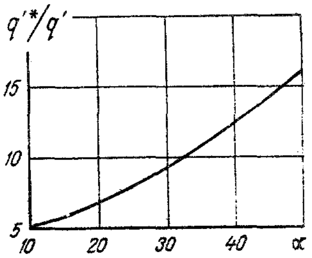

Fig. 6. Comparison of the maximum heat flux according to Eq. (22). The quantity α is in degrees.

interface are isothermal (Fig. 4). On the basis of these arguments the authors wrote the governing equation in the form

$$\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} = -(g(\rho_l - \rho_v)/\mu)(z - z')/z'. \tag{8}$$

All the numerical solutions of this equation were presented in the form of a function of two dimensionless parameters Λ and Ω, which were defined as follows:

$$\Lambda = \frac{C\sigma\mu\lambda\cos\theta\,(T_W - T_{sat})}{\rho_l\,(\rho_l - \rho_v)^2\, g^2 r^* d^5}, \tag{9}$$

$$\Omega = D'\mu v(\rho_l - \rho_v)\, g d^2, \tag{10}$$

where $C = 2(\csc^3\alpha - \csc^2\alpha)$, $D' = \csc\alpha$ for triangular grooves; $C = 8(\pi - 2)/\pi^2$; $D' = 2$ for semicircular grooves; $C = 2$; $D' = 2$ for square grooves.

Bresler and White determined the optimal angle at the apex of a triangular groove 2α. For a constant depth the maximum heat flux was produced by a groove with an apex angle of about 30°. They obtained an expression for the groove depth d_0 for which the heat flux reaches a maximum:

$$d_0 = B\left[\frac{\sigma\mu\lambda\cos\theta\,(T_W - T_{sat})}{\rho_l\,(\rho_l - \rho_v)^2\, g^2 r^*}\right]^{1/5}, \tag{11}$$

where B = 5.11, 1.97, 2.67, respectively, for a triangular groove with apex angle 30°, a semicircular groove, and a square groove.

For the optimal groove depth the maximum heat power is

$$Q = E\psi(T_W - T_{sat})^{3/5}\, t, \tag{12}$$

where E is a constant whose values are, respectively, 0.269, 0.283, and 0.438 for a triangular groove with apex angle 30°, a semicircular groove, and a square groove; and,

$$\psi = \left[\frac{\rho_l^2\, r^{*2}\sigma^3\lambda^3\cos^3\theta}{g\mu^2(\rho_l - \rho_v)}\right]^{1/5}. \tag{13}$$

Morits [12] has investigated arterial heat pipes with groove-type thin-film evaporators in the case where power is supplied under boundary conditions of the second kind. The author proposed a technique for calculating the maximum heat flux, proceeding from hydrodynamic analysis of laminar flow of the liquid, and assuming that the heat transfer is uniform along the groove length, for two orders of magnitude of the axial heat flux Q'.

1. $Q' = Q'_{max}$. In this case it is assumed that the meniscus at the beginning of the groove is concave (Fig. 5a). By equating the pressure drop due to friction of the liquid (the Hagen–Poiseuille equation)

$$dP_l = \frac{32\dot{m}(x)\,\mu_l}{A_l\,(x)\, D_h^2(x)\,\rho_l} \tag{14}$$

with the increment of capillary pressure

$$dP_{cap} = \sigma\cos\theta\, d\left(\frac{1}{r}\right) \tag{15}$$

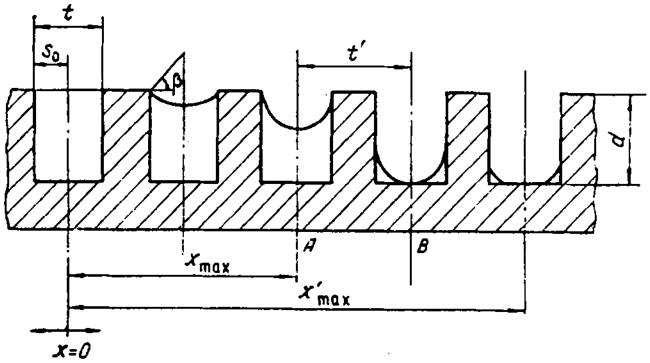

Fig. 7. Meniscus shape in a rectangular channel with $Q' \ll Q'_{max}$.

and carrying out a transformation, the author obtained an expression for the maximum heat flux density

$$q' = \left(\frac{s_0}{x_{max}}\right)^2 N_l \cos\theta K_1(\alpha). \tag{16}$$

Here

$$K_1(\alpha) = 5.21 \cdot 10^{-3} \left(\frac{\mathrm{tg}^2\alpha}{\sin^3\alpha}\right) [2\cos^2\alpha - \mathrm{tg}\,\alpha\,(\pi - 2\alpha - \sin 2\alpha)]^3; \tag{17}$$

and N_l is a characteristic of the working liquid,

$$N_l = \frac{\sigma r^* \rho_l}{\mu}. \tag{18}$$

2. $Q' \ll Q'_{max}$. The author assumed that here one reaches the limiting case where the meniscus at the beginning of the groove (x = 0) is flat (Fig. 5b). The heat flux density is determined from the following formula:

$$q'^* = \frac{s_0^2}{x_0(x_{max} - 1/2x_0)} N_l \cos\theta K_2(\alpha), \tag{19}$$

where the constant is

$$K_2(\alpha) = 9.765 \cdot 10^{-4} \left(\frac{\mathrm{tg}^2\alpha}{\sin^3\alpha}\right) [4\cos^2\alpha - \mathrm{tg}\,\alpha\,(\pi - 2\alpha - \sin 2\alpha)]^3. \tag{20}$$

Introducing the quantity $x^*_{max} = x_{max} - x_0$ into Eq. (16) and relating Eq. (19) to Eq. (16), the author obtained

$$\frac{(x_{max} - x_0)^2}{x_0(x_{max} - 1/2x_0)} = \frac{K_1(\alpha)}{K_2(\alpha)} \tag{21}$$

or

$$\frac{q'^*}{q'} = \left(\frac{x_{max}}{x_{max} - x_0}\right)^2. \tag{22}$$

The solution of this equation is presented graphically in Fig. 6. However, because heat pipes have evaporators with considerably distorted groove profiles, the author was not able to compare the experimental and theoretical results with sufficient accuracy.

Fel'dman and Berger [37] analyzed the evaporation process in thin-film evaporators for heat pipes in the case where the power is supplied under boundary conditions of the second kind. The authors improved the formula for calculating the maximum heat flux when $Q' \ll Q'_{max}$. The improved equation (19) has the form

$$q_1' = N_l \cos\theta \left(\frac{s_0}{x_{max}}\right)^2 \frac{K_3(\alpha)}{\frac{x_0}{x_{max}}\left(1 - \frac{1}{2}\frac{x_0}{x_{max}}\right)}, \tag{23}$$

where

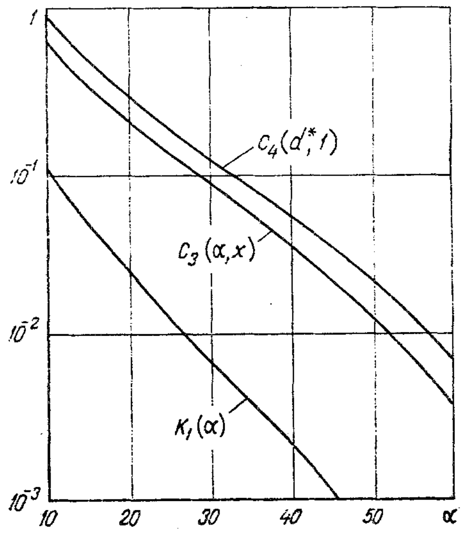

Fig. 8

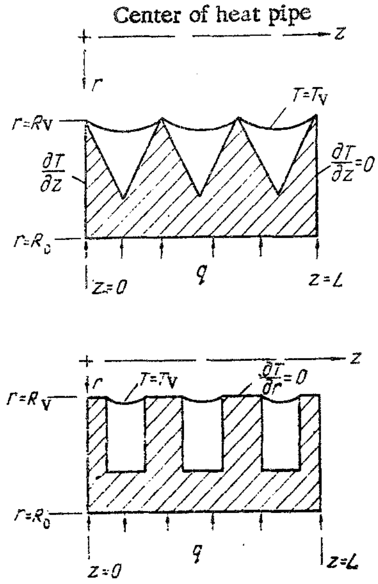

Fig. 9

Fig. 8. The constants $C_4(d^*, 1)$, $C_3(\alpha, x)$, and $K_1(\alpha)$ as a function of the semivertex angle of the groove. α, deg, $\tilde{\alpha}_{pr}$ = arctan s_0/h.

Fig. 9. Boundary conditions for calculating temperature drop over the liquid layer in the groove.

$$K_3(\alpha) = \frac{\sin^2\alpha}{16} \int\limits_{\alpha}^{\frac{\pi}{2}} \left[\operatorname{ctg}\alpha - \frac{\pi - 2\beta - \sin 2\beta}{2\operatorname{cosec}^2\beta}\right] \sin\beta d\beta. \tag{24}$$

The authors postulated that Eqs. (16), (19), and (23) could be written in the following form:

$$q' = N_l \cos\theta \left(\frac{s_0}{x_{max}}\right)^2 C_1(\alpha, x), \tag{25}$$

$$q'^* = N_l \cos\theta \left(\frac{s_0}{x_{max}}\right)^2 C_2(\alpha, x), \tag{26}$$

$$q_1' = N_l \cos\theta \left(\frac{s_0}{x_{max}}\right)^2 C_3(\alpha, x), \tag{27}$$

where $x = x_0/x_{max}$, and where

$$C_1(\alpha, x) = \frac{K_1(\alpha)}{\left(1 - \frac{x_0}{x_{max}}\right)}, \tag{28}$$

$$C_2(\alpha, x) = \frac{K_2(\alpha)}{\frac{x_0}{x_{max}}\left(1 - \frac{1}{2}\frac{x_0}{x_{max}}\right)}, \tag{29}$$

$$C_3(\alpha, x) = \frac{K_3(\alpha)}{\frac{x_0}{x_{max}}\left(1 - \frac{1}{2}\frac{x_0}{x_{max}}\right)}. \tag{30}$$

Fel'dman and Berger derived an equation for the heat flux in evaporation of a liquid from a rectangular groove for the case when $Q' \ll Q'_{max}$. The assumptions in deriving the formula are the same as for triangular grooves (Fig. 7):

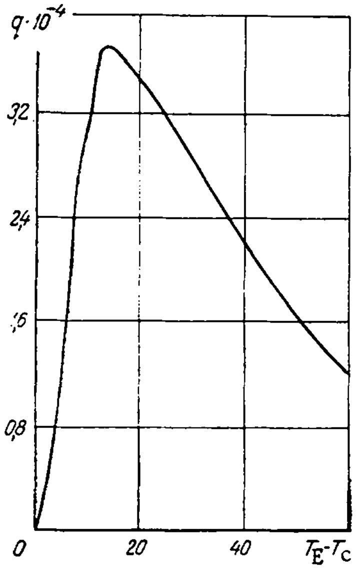

Fig. 10. The heat flux transmitted by a heat pipe as a function of the temperature difference between the evaporator and the condenser. $q \cdot 10^{-4}$, W/m^2; $(T_E - T_C)$, °K.

$$q' = N_l \cos\theta \left(\frac{s_0}{x_{\max}}\right)^2 C_4(d^*, \delta), \tag{31}$$

where

$$C_4(d^*, \delta) = \frac{J(d^*)}{16(1+\delta)(1+d^*)^2} \tag{32}$$

and

$$J(d^*) = \int_{\delta}^{\frac{\pi}{2}} [2d^* - C(\beta)]^3 \sin\beta d\beta, \tag{33}$$

where

$$d^* = d/s_0 \tag{34}$$

and

$$\delta = \frac{t' - t}{t} . \tag{35}$$

Equations (16), (23), and (31) differ with respect to the quantities $K_1(\alpha)$, $C_3(\alpha, x)$, and $C_4(d^*, \delta)$. Figure 8 shows these quantities graphically. It can be seen that the rectangular grooves remove the largest heat flux.

Using a finite-difference method Fel'dman and Berger calculated the temperature drop over the layer of liquid evaporating in the groove. Here they assumed that the heat transfer along the liquid layer in the groove occurred only by heat conduction. At the wall–liquid interface the boundary conditions are of the fourth kind, and the meniscus temperature is the same as that of the vapor (Fig. 9). The heat transfer level for which bubble-type boiling begins in the groove was determined from the Rohsenow formula [39].

For a heat pipe operating in the evaporation condition, a formula was assumed for calculating the power transmitted by an evaporator with a triangular groove [14]:

$$q = \frac{3.26 \cdot 10^{-2} t^2 \cos\alpha \cos\theta}{(\delta + 1)\pi^2 R^2} \frac{\sigma \rho_l r^*}{\mu} C(\alpha), \tag{36}$$

$$C(\alpha) = \left[\operatorname{tg}\alpha - \operatorname{ctg}\alpha - \frac{(\pi/2 - \alpha)}{\cos^2\alpha}\right]^3 \cos^2\alpha. \tag{37}$$

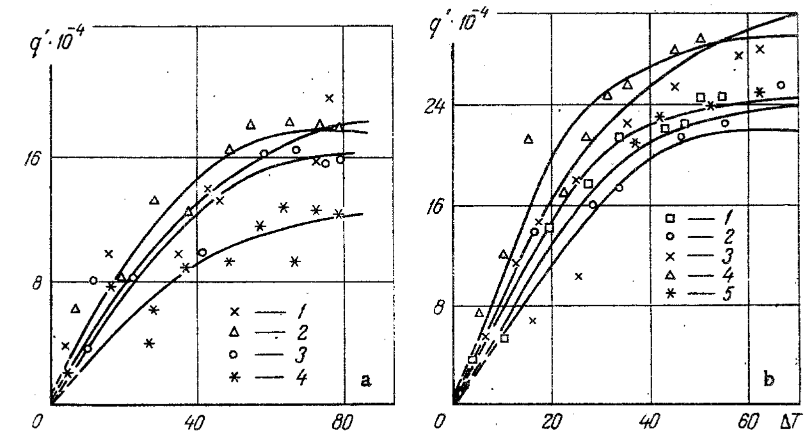

Fig. 11. The heat flux as a function of the wall overheat: a) for triangular groove [1) $2\alpha = 15°$; 2) 30; 3) 45; 4) 60]; b) for rectangular grooves of width 0.4 mm [1) d = 0.4 mm; 2) 0.6; 3) 0.8; 4) 1.2; 5) 1.6]. $q' \cdot 10^{-4}$, W/m²; ΔT, °K.

An analytical model has been generated in a nonarterial heat pipe [15] to determine the maximum evaporator power; the model describes the hydrodynamic conditions for mass transfer. Here the following basic assumptions were made:

1. There is no gravitational force, g = 0.
2. The vapor pressure above the liquid is constant along the channel.
3. The specific heat flux is constant along the evaporation zone.
4. The preheating of the liquid in the evaporator is small compared with the phase transition heat.
5. The liquid flow is laminar.
6. There is no friction between the vapor and the liquid.

The equation of motion of a viscous liquid in a groove due to surface tension forces

$$\frac{dP_h}{dx} = -\frac{dP_{cap}}{dx} \quad (38)$$

after transformations in accordance with the assumptions adopted, takes the form

$$\sigma \frac{d}{dx}\left(\frac{1}{R_{cap}}\right) = \frac{32\mu_l\, q_E C_0 L_E}{A_l D_h^2 \rho_l r^*}. \quad (39)$$

After integration, one obtains an expression for the maximum heat flux:

$$q_{max} = \frac{s_0^2 N_l\, \Phi_W K_\Omega}{2(\delta+1)(L_E + L_c + 2L_a) L_E}. \quad (40)$$

In the ITMO low-temperature laboratory of the Academy of Sciences of the Belorussian SSR investigations have been made of the heat transfer process during evaporation and boiling of a liquid in groove-type evaporators. Tests of an arterial heat pipe with a thin-film evaporator [9] have shown that the heat power transmitted depends on the temperature difference between the evaporator and the condenser (Fig. 10). With an increase in this difference (the evaporator temperature remains constant), the power transmitted by the pipe begins to grow, reaches a maximum, and then drops. This is explained by the fact that when a certain level of overheat of the wall above the saturation temperature is reached, the process of film boiling in the evaporator grooves begins. The heat pipe was filled with methanol and had an evaporator with triangular grooves of width 0.2 mm and vertex angle 30°.

In this laboratory tests were made of thin-film evaporators with triangular and rectangular grooves. The influence of the geometric parameters of the grooves on the heat removal was determined. The thermal power was supplied under boundary conditions of the third kind. During the experiments the total thermal power taken away by the evaporator was determined, and also the variation of the groove temperature along the moving liquid [34].

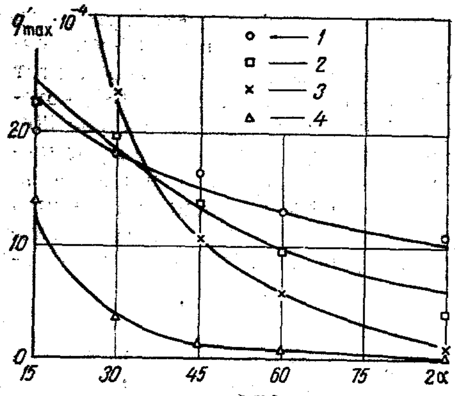

Fig. 12

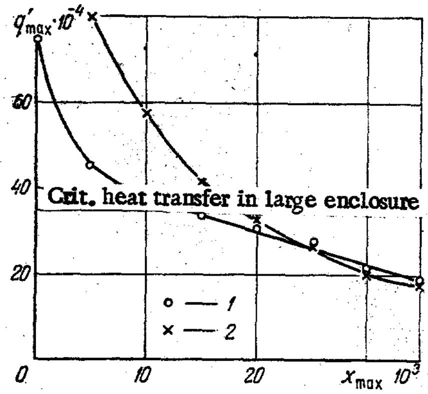

Fig. 13

Fig. 12. The maximum heat flux as a function of the groove vertex angle. x_{max} = 40 mm, t = 0.4 mm: 1) experiment; 2) Eq. (44); 3) the Morits formula, Eq. (19); 4) the Morits formula, Eq. (16). α, deg.

Fig. 13. The maximum heat flux as a function of groove length. t = 0.4 mm, 2α = 30°: 1) experiment; 2) Eq. (44). $x_{max} \cdot 10^3$, m.

The experiments were carried out with acetone in the temperature range 283-363°K.

The optimal vertex angle of a triangular groove was calculated. For equilibrium of forces at the phase interface the hydraulic drag increment is equal to the capillary drop

$$-\xi \frac{\mu v}{2D_h^2}\, dx = \frac{\sigma \cos\theta}{R_{cap}}. \tag{41}$$

For a constant groove width (t = const) the rate of motion of the liquid along the groove as a function of vertex angle is

$$v = -K \frac{\cos\alpha\,(1-\sin\alpha)}{1+\sin\alpha}. \tag{42}$$

For a constant depth (d = const) we have

$$v = -K \frac{(1-\sin\alpha)\sin\alpha}{(1+\sin\alpha)^2}. \tag{43}$$

For t = const the maximum velocity is achieved with a groove whose vertex angle tends to zero (i.e., the depth tends to infinity). For d = const the maximum velocity occurs for 2α = 38°. This result agrees well with the data of Bresler and White [13], who obtained an optimal value of 2α equal to 30°, experimentally.

The experiments have shown that, when power is supplied under boundary conditions of the third kind, the heat transfer is nonuniform along the grooves. The thermal power is reduced in inverse proportion to the distance where the liquid is supplied. On this basis, an equation was derived for calculating the maximum heat flux taken away in the working length of the groove [16]:

$$q'_{max} = \frac{t^2 N l \cos\theta\, C(\alpha)\, K(\alpha)}{x_{max}\left[x_{max} - (x_{max} + x_0)\left(1 - \dfrac{1}{\ln \dfrac{x_0 + x_{max}}{x_0}}\right)\right]}, \tag{44}$$

where

$$C(\alpha) = [\cos^2\alpha/(1+\sin\alpha)^2]\,\frac{(1-\sin\alpha)}{\sin\alpha}, \tag{45}$$

and x_0 is the length of groove section occupied by the supply artery.

The coefficient K(α) takes account of the influence of boiling in the groove on the heat transfer process. For acetone it was determined experimentally that $K(\alpha) = 0.0535\,(2\alpha)^{1.55}$.

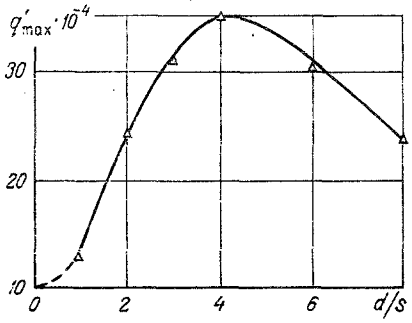

Fig. 14. The maximum heat flux as a function of the depth-to-width ratio for a rectangular groove. t = 0.4 mm.

The heat flux is shown in Fig. 11a and b as a function of the temperature drop $\Delta T = T_w - T_{sat}$. It can be seen that there are three characteristic sections on the curve $q' = f(\Delta T)$. The first is the region of evaporation of saturated liquid. The heat is transmitted by natural convection from the heated surface along the layer of liquid. The second region is the bubble boiling region. The heat flux increases in proportion to ΔT. The third region is the region of critical boiling heat transfer. In this region the heat flux reaches a maximum, and an increase in the overheat of the wall does not produce further increase.

The limiting heat flux depends on the transport properties of the groove. Figure 12 shows q'_{max} for grooves with vertex angles of $2\alpha = 15, 30, 45, 60, 90°$. It can be seen that a tendency for an increase in heat flux is caused by the increase in liquid flow rate along the groove. The largest value of q'_{max} is reached for a groove having $2\alpha = 15°$. The experimental results agreed well with the theory of Eq. (16). Only for the case of $2\alpha = 90°$ does the error reach 50%, since in this case the heat transfer is affected mainly by the process of sputtering of liquid during boiling, in view of the smallness of the capillary drop. For comparison the results of calculations using Eqs. (16) and (44) are shown.

The influence of the length of movement of the liquid on the heat transfer is shown in Fig. 13. Tests were conducted on evaporators with triangular grooves, having $2\alpha = 30°$. The length of movement x_{max} varied from 40 to 5 mm. With decrease of x_{max} an increase in heat flux is observed. The heat transfer limit at large heat flux is the onset of the film boiling regime. It is characterized by formation of a stable vapor sublayer at the heater surface, which prevents liquid reaching there.

Zuber [38], by analyzing Helmholtz instability, derived an equation for the critical heat flux:

$$q_{crit} = \frac{\pi}{24} \rho_v r^* \left[\frac{\sigma(\rho_l - \rho_v) g}{\rho_v^2}\right]^{1/4} \left[\frac{\rho_l}{\rho_l + \rho_v}\right]^{1/2}. \quad (46)$$

According to this formula, the critical heat flux for the acetone is ~40 W/cm². Analysis of the limiting heat flux values shows that for boiling in grooves the heat transfer is affected predominantly by the rate of flow of liquid along the groove. For small motion of the liquid (x_{max} = 5 mm) q'_{max} = 75 W/cm² exceeds the critical value for a large enclosure. It is probable that the vapor bubble bursts in the groove, without attaining its separation diameter, because of the small thickness of liquid. Therefore, a stable vapor film cannot form at the heater surface. This is confirmed by tests of evaporators with rectangular grooves (Fig. 14). In the limit of values of depth up to 8 mm, an increase in depth leads to an increase in heat flux. In a deeper groove favorable conditions exist for clustering of bubbles, and therefore the heat transfer is decreased for such grooves.

The main conclusion of this review is that groove-type thin-film evaporators allow thermal power to be removed, which is close to, and in some cases greater than, that obtained with liquid boiling in a large enclosure.

However, it should be pointed out that the liquid boiling process in a groove is affected by many factors: the cleanliness in preparation of the grooves, the wetting, the position of the relative gravitational field, the physical properties of the liquid and the vapor, the geometrical size of the heater surface, and so on. The difficulty of a numerical evaluation of this influence is an obstacle to a detailed analytical description of heat transfer with boiling of a liquid moving in a groove.

Heat Transfer with Boiling on an Immersed Developed Surface. At present methods of cooling high-power low-volume heat sources (laser lamps, klystrons, etc.) are finding increasingly wide use, based on boiling on developed (finned) heater surfaces, when the heat flux density at the surface carrying the fin or the groove is several times larger than the value q_{crit} typical for a smooth surface [5]. However, there are not enough numerical data in the literature pertaining to boiling on immersed developed heater surfaces washed by a flow of saturated or nonheated liquid. It is known that there is a vapotron effect, which has been experimentally confirmed in [21]. With boiling of a liquid on a developed surface q_{crit} is determined by the conditions affecting movement of the liquid in the grooves or between the fins.

The investigations of [18] have shown that the descent of liquid into the groove depends on the nature of the liquid flow outside it. In the case of two-phase flow having a large mixing rate, the distribution of vapor and liquid over the channel section is nonuniform, which leads to local overheat and to pressure fluctuations over the channel perimeter. According to [18], the critical heat flux for an entire finned wall is given by the liquid flow conditions at the entrance to the channel (θ_{ent}, v):

$$\bar{q} = f(\theta_{ent}\, v) = q_{W crit}/q_{k1}, \tag{47}$$

$$\bar{q} = q_0(k) + \bar{q}_1(x,\ v,\ k), \tag{48}$$

where

$$q_0(k) = \frac{q_{0crit}}{q_{crit}} = \frac{7.5k}{k+6.5};\quad \varphi(k) = -\frac{k+10}{k+6.5}\ x_0(k) = 0.0037k - 0.05;$$

and k is a fin coefficient, equal to the ratio of the heat-removal surface area to the surface area carrying the fins. Experiments have shown that with unheated water heat fluxes up to 15 mW/m² can be removed from the finned surface, with comparatively low flow velocities and underheat.

This area holds out great promise for the construction of closed evaporator-condensation devices (heat pipes or vapor chambers) used in a gravity field, when one observes motion of the cooled liquid over the developed heater surface from top to bottom.

Heat transfer with boiling of liquid in horizontal slots was studied by Smirnov et al. [17], who correlated the test data in terms of limiting heat flux, from the viewpoint of analysis of stability of the two-phase boundary layer in bubble boiling, when: 1) continuity of the liquid phase and discreteness of the vapor phase were observed; 2) the removal of the vapor phase showed fluctuations; and 3) for q_{crit} all the heat supplied to the liquid is converted to enthalpy of vapor formation within the two-phase boundary layer.

For a horizontal two-dimensional slot the expression

$$\frac{q_{crit}}{q_{0\,crit}} \leqslant \frac{1}{\sqrt{1 + K\dfrac{\rho_v}{\rho_l}\left(\dfrac{w}{l}\right)^3}} \tag{49}$$

was obtained, where q_{0crit} is the critical heat flux for liquid boiling in a large enclosure.

Thus, it can be stated that in evaporation and boiling of liquids in capillary-porous bodies, and also on developed surfaces, coated with a mesh of capillary channels, the heat transfer process has a number of special features, compared with boiling of a liquid in a large enclosure, over a smooth surface, and also in channels and tubes with smooth walls. The porous structure and the capillary grooves enchance the heat transfer process over a wide range of heat flux and allow a smooth transition from the evaporation regime to the boiling regime without sound effects and mechanical vibrations. The process of approaching critical boiling is smoothed out, and the dependence of q on ΔT has the form of a smooth curve without clearly pronounced maxima. Coating the porous structures or grooves of the heat transfer surface to enhance the process is particularly effective for cryogenic liquids (helium, hydrogen, etc.), and also in low-temperature heat pipes and thermosiphons. It is of great practical importance to investigate the heat transfer process in filtration transfer of liquid within porous bodies and boiling of liquid in the pores with discharge of two-phase flow or superheated vapor. Unfortunately, there are almost no results of such investigations in the literature, although the importance of this method of cooling heat-generating porous elements is self-evident.

NOTATION

A, area; b, wick thickness; C, constant; D, diameter; D_h, hydraulic diameter; d, groove depth; d*, ratio of groove depth to one-half of its width; f, specific thermodynamic potential; g, acceleration of gravity; h, height of liquid rise; j, liquid flow; K, a constant; K', permeability; L, length; l, slot gap; m, mass flow rate of liquid; P, pressure; Q, amount of heat; q, heat flux density; Q', power removed by one groove of the evaporator; q', heat flux taken away by one evaporator groove; R, radius; R_{cap}, capillary radius; r^*, latent heat of vaporization; s_0, one-half groove width; T, temperature; t, groove width; V, volume; v, mixing rate of liquid; w, slot gap; x_{max}, maximum length of liquid motion in the groove; x, y, z, coordinates; α, half-angle at vertex of a triangular fin; α_e, coefficient of convective heat transfer; β, angle between radius of curvature of meniscus and the plane of the groove; β', coefficient of linear expansion; γ, slope angle of wick; δ, ratio of length of protuberance to its width; δ*, thickness of liquid film; ε, porosity; θ, wetting angle; λ, thermal conductivity; μ, dynamic viscosity; ξ, drag coefficient; ρ, mass density; σ, surface tension. Subscripts: l, liquid; w, wall; v, vapor; sat, saturation; lr, layer; max, maximum; min, minimum.

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