NOTATION

ℓ, length of the evacuation channels; U_1, conductance of the chamber-wall−screen channel; U_2, conductance of the screen-object channel; Q_1, intensity of the localized source of gas evolution; Q_2, intensity of gas evolution from the surface of the object being cooled; α, perforation factor of the screen $\alpha = S_a/S$, where S_a is the total area of the openings in the screen and S is the geometric area of the surface of the screen; P_x, pressure in the system at a distance x from the inlet; P_0, pressure in the input section; q, specific gas-evolution flux from the walls of the system; Π, perimeter of the transverse section; x, distance from the inlet to the section under study; Q, gas load at the far end; and U_m, conductance in the molecular regime.

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HEAT PIPE WITH COMBINED CAPILLARY-POROUS STRUCTURE

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The authors present results of an experimental investigation of the heat-transfer characteristics of low-temperature heat pipes with longitudinal capillary grooves coated with a layer of porous metal.

A topic of current interest is to increase the heat transfer in heat pipes, to increase their heat transmission capability. One way to solve this problem is to use capillary-porous structures of combined type, various forms of which were investigated in [1, 2].

In this paper for the first time we examine a heat pipe with a capillary-porous structure in the form of a combination of longitudinal grooves and a fine porous metal layer, deposited in a comparatively simple manner described in [3, 4]. Figure 1, taken with a microcamera, shows a section of an element of such a structure. The porous copper coating was deposited on the inner surface of a copper tube with 24 open longitudinal rectangular grooves of depth D = 0.46 mm and half width W = 0.37 mm, serving as a cathode in the process of cathodic deposition from solution. To avoid secondary effects associated with concentrated polarization, we used copper sulfate electrolyte with triatomic alcohol additive vigorously mixed by means of compressed air. The prepared specimens were washed with water and dried with hot air. The structure thus obtained is a porous skeleton, consisting of spherically shaped particles with a developed surface, having good bonding with the substrate. Its porosity is 45%, the mean pore size is about 140 μm, and here the dimensions of the particles increase as distance from the substrate increases. As can be seen from Fig. 1, the thickness of the porous skeleton varies over the fin surface and is greatest at the fin apex.

The freshly formed trap@ezoidal capillary grooves differ from basic rectanges by having greater depth and less width at the exit to the heat-pipe vapor channel. This must necessarily lead to an increase of the maximum power transmitted by such a heat pipe under conditions where mass forces oppose transport of liquid [5]. In addition, a possible result of depositing the porous layer is to reduce the thermal resistance of the evaporator by increasing the area of liquid evaporation and intensifying the boiling process.

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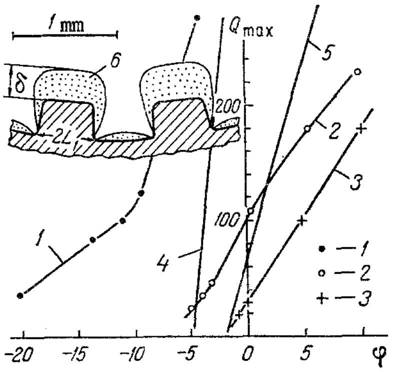

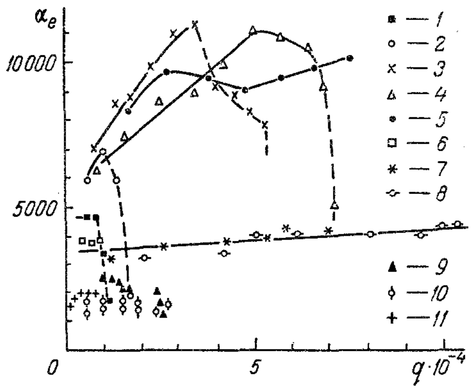

Fig. 1 Fig. 2

Fig. 1. The limiting heat transfer Q_{max}, W, as a function of the slope angle to the horizontal ϕ, deg: 1) heat pipe HP1 with water, M = 3.8 g; 4) calculation using the method of [5]; 2) HP1 with acetone, M = 4.6 g; 3) HP2 with acetone, M = 3 g; 5) calculation using the method of [5]. An element of cross section of the capillary structure; 6) porous layer.

Fig. 2. The heat-transfer coefficient of the evaporator α_e, W/(m^{2}K) as a function of the heat flux density q, W/m^2 at its external surface (the heat transfer agent is acetone): 1) HP1, ϕ = − 1.5°; 2) 0.133; 3) 5.4; 4) 10; 5) 33; 6) HP2, ϕ = 0.1°; 7) 10; 8) 33; 9) data of [1] (W = 0.25 mm; L_1 = 0.24 mm); 10) data of [5] (W = 0.23 mm, L_1 = 0.33 mm); 11) data of [5] (W = 0.12 mm, L_1 = 0.4 mm).

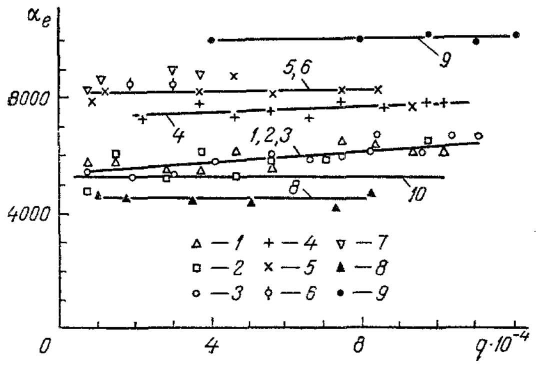

Fig. 3. The evaporator heat-transfer coefficient α_e, W/(m^2·K) as a function of the heat flux density q, W/m^2 on its external surface (the heat-transfer agent is water): 1) HP1, ϕ = 33°; 2) 10; 3) 5; 4) 0.28; 5) − 4.3; 6) −9; 7) − 11; 8) HP2, ϕ = 20°; 9) data of [2] (W = 0.15 mm, L_1 = 0.25 mm); 10) calculated using the method of [5].

Experimental investigations have been conducted, using a conventional technique applicable to determine the steady-state characteristics of the heat pipe. Along a lateral generator of the heat pipe in the wall, we welded 9 Chromel−Copel thermocouples, three in each zone (evaporation, condensation, and transport). As a heat source we used an ohmic heater, and heat removal in the condensation zone was accomplished with the aid of a liquid heat-exchanger through which water at a given temperature was passed. The heat pipe had the following geometrical parameters: R_{ex} = 6.5 mm, L_e = 70 mm, L_c = 75 mm, L_a = 80 mm, minimum wall thickness 0.5 mm. We could solve the model one-dimensional heat-conduction problem for the heat pipe wall by making a number of estimates, having shown that the surface temperature of the adiabatic zone in the middle part is close, with sufficient accuracy, to that of saturated vapor in the heat pipe. Therefore, the heat-transfer coefficients between the external surface of the heat-carrying zones and the saturated vapor (α_e, α_c) were determined as the ratio of the heat-flux densities in the evaporation and condensation zones to the differences of their mean temperatures from that of the center of the adiabatic zone. The limiting heat transfer of the pipe Q_{max} was set by the sharp drop in the heat-transfer coefficient of the evaporator α_e for a stepwise increase of the heat load.

Figures 1-3 show some of the experimental data obtained on two heat pipes − HP1 and HP2, which differed in the thickness of the deposited porous layer. The geometrical parameters of the porous layer in HP1 can be established from Fig. 1; the greatest thickness at the fin apex δ was 0.45 mm. In HP2 the value of δ was 0.08 mm. The heat pipes were tested at various slope angles with heat-transfer agents of acetone and water at an operating temperature of the mean part of the adiabatic zone of 60°C.

Analysis of the data presented in Fig. 1 shows that the limiting heat transfer of the combined structure can be significantly greater compared with the case of a heat pipe with grooves and no coating. This is seen particularly clearly for HP1 with water operating at negative slope angles to the horizontal, $\phi < 0$. The calculated relation $Q_{max}(\phi)$ for this case, obtained using the method of [5] for the original geometry of grooves with no porous coating, is shown as curve 4 on Fig. 1. The increase of the maximum power of the heat pipe as a result of depositing the porous layer can be explained both by the change of geometry of the longitudinal capillary grooves and by the possible transport of liquid phase heat transfer agent along the porous layer. This refers also to the data obtained on HP1 with acetone at $\phi < 0$. We note that the experimental points corresponding to operation of HP2 with acetone are located on Fig. 1 below the calculated curve 5. This is explained by the fact that the thin porous coating on the groove surface practically does not change the groove geometry, but it increases the hydraulic resistance during flow of the liquid heat-transfer agent due to roughness of the walls.

In Fig. 2 we compare values of the heat-transfer coefficient in the evaporators of HP1 and HP2, and also in heat pipes with longitudinal grooves without deposition, operating with acetone. One can clearly see an intensification of heat transfer in the evaporators with a porous coating, which can be explained by the fact that evaporation occurs also from the surface of the fins to which liquid is supplied due to capillary forces. In the case of HP2, where the porous layer is very thin, the apices of the fins are apparently not fully coated with liquid, and therefore the heat-transfer coefficient in the evaporator of HP2 is less than for HP1. The broken lines on Fig. 2 correspond to the start of the hydrodynamic crisis of heat transfer of the heat pipe.

The analogous dependences for the case where water is the heat-transfer agent are shown in Fig. 3, where the solid lines approximate to the experimental data. Here one should note that the values of α_e obtained in [2] on a copper plate with grooves, for correct comparison with the data relating to a heat pipe of circular section, should be decreased by a number roughly equal to the quantity R_{ex}/R_v and then the value of α_e from [2] is about 8500. The solid line 10 on Fig. 3 corresponds to the calculation according to [5] for the original groove dimensions. Thus, in the experiments conducted with water as the heat-transfer agent no substantial intensification of heat transfer in the evaporator was achieved. This may be due to insufficient wetting of the copper porous structure by the water, resulting in the apices of the fins between grooves not being fully moistened with liquid, especially in the HP2 case. Evidently, in such situations one must ensure good wetting of the wick by the heat-transfer agent by special measures, in particular by oxidizing the structure, by a lengthy period of prior operation of the heat pipe before starting the measurements, etc.

The general form of the experimental dependences $\alpha_e(q)$ presented in Figs. 2 and 3 is not characteristic for the developed boiling regime. We can thus postulate that the basic regime of operation of the heat pipes investigated is evaporative. We note that a change of the level of the operating temperature of the heat pipe in these experiments from 20 to 100°C led to an insignificant increase of α_e, by about 20%.

In the experiments we also determined the integral coefficient of heat transfer in the heat-pipe condenser. However, because of the complexity of the heat- and mass-transfer mechanisms during condensation in such capillary-porous structures in a gravity field, a detailed discussion of this matter should be given separately. In the present paper we limit ourselves to a general remark that the thermal resistance of the condensers of HP1 and HP2 is roughly the same or somewhat higher than for a heat pipe with longitudinal grooves with no porous coating. The improvement of heat transfer in this case may stem from the fact that the surface condensation is free from the liquid film which is removed via the porous layer.

Taking into account the relative simplicity of fabricating a heat pipe with the combined capillary porous structure examined here, one can foresee its wide use in various technical enterprises in cases when increased requirements are imposed on the heat pipe (operation in an unfavorable orientation in the field of mass forces, low thermal resistances, mechanical stress). However for this purpose one must conduct a detailed investigation, both experimental and theoretical, of the heat- and mass-transfer processes in such heat pipes.

NOTATION

D, W, depth and half width of the grooves; L_1, half width of the fin between grooves; L, length; α, heat-transfer coefficient; Q_{max}, maximum heat flux; q, heat flux density; R, radius; ϕ, slope angle; δ, maximum thickness of the porous layer. Subscripts: e, evaporator; c, condenser; a, adiabatic; v, vapor; ex, external.

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HEAT-EXCHANGE INTENSITY IN BOILING AT SURFACES WITH POROUS CLADDINGS UNDER CAPILLARY TRANSPORT CONDITIONS

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The results of an experimental investigation of the heat-exchange intensity in boiling at surfaces covered with porous, metal-fiber claddings under capillary transport conditions are described.

Capillary-porous claddings are used on an ever-increasing scale in heat-exchange devices of various types in order to ensure capillary transport of the liquid and also intensify the heat exchange during phase transitions of the coolant. The results obtained in investigating the hydrodynamic and heat-exchange processes occurring in capillary structures have been treated exhaustively in [1-3] and other studies. However, relatively little attention has been paid in these investigations to the heat-exchange intensity in coolant boiling under capillary transport conditions. The results obtained in [4-7] and many other papers are uncoordinated and contradictory. For instance, the heat-exchange intensity is independent of the porous cladding thickness according to data supplied by some authors, while it depends on this thickness according to others; meanwhile, the exponent n of the thermal flux density in the expression $\alpha = f(q)$ varies from 0.12 to 0.7. This line of research does not include comprehensive investigations of the effect of the geometric, thermophysical, and structural characteristics of porous claddings on the heat exchange intensity in boiling.

Our aim was a thorough investigation of the effect of the parameters of porous metal-fiber structures on the heat-exchange intensity in the boiling of water, acetone, and ethyl alcohol under capillary feed conditions at saturation pressures in $(0.5\text{-}0.98)\cdot 10^5$ Pa range.

The experimental investigation was performed by means of the device whose operating section is shown in Fig. 1a, using specimens of capillary metal-fiber structures (Fig. 1b). The characteristics of the specimens are given in Table 1. Each specimen consists of a base-layer 5 with a diameter of 35 mm and a thickness of 3 mm and an oxidized capillary structure 6 with a diameter of 105 mm, sintered onto the base-layer surface. The junctions of six copper-constantan thermocouples 9, coated with a Teflon lacquer, are embedded in grooves 13 of the base layer. The lead-out of the thermocouples from glass cylinder 4 is provided through Teflon sheaths and a hermetic lead-out from flange 7. In order to eliminate the

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