HEAT OUTPUT AND MAXIMUM HEAT TRANSFER OF HEAT PIPES WITH CONTINUOUS CORRUGATED WICKS

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From experimental and analytical studies the authors conclude that corrugated structures are attractive as heat pipe wicks.

The main element of a heap pipe (HP) is its capillary-porous structure. Known efficient heat pipe capillary structures are mesh, metal fiber, sintered, and other types. In spite of this diversity, the search continues for new structures that are simpler to fabricate.

It was shown in [1] that with a heat pipe with a capillary structure made of continuous corrugated metal foil one can obtain high heat-transfer characteristics. The fabrication process for this kind of wick is simple, within existing technology, does not require special instruments and equipment, and ensures good reproducability of the parameters.

For these structures the experimental data are extremely limited, and there are no physical models and the necessary design relations. Individual studies [2, 3] are known where results were given of heat-pipe investigations with corrugated mesh wicks. In the experiments the vapor temperature was not measured, and it is therefore not possible to estimate the heat-transfer coefficients in the evaporation and condensation sections, as well as other important characteristics of the heat pipe.

In accordance with what has been said, special experiments were set up to study heat transfer with vapor formation in corrugated metal foil wicks. The heat transfer in the evaporation zone of the heat pipe was modeled using an approved technique [4]. The structure specimens were attached to the test surface of a wedge heater of diameter 30 mm, located in a sealed chamber. The experiments were conducted under conditions of capillary level makeup and wetting with heat-transfer agent (water) at atmospheric pressure in the heat flux range 10^4-10^6 W/m^2.

The corrugated structure was fabricated by rolling metal foil between toothed wheels of a certain modulus. The foil was made of copper and stainless steel, first cut into a strip of width b = 5-40 mm. Table 1 shows the geometrical parameters of the corrugated structures.

Before the test the structure was cleaned and degreased in acetone, and then in alkaline solution, etched in a 10% solution of hydrochloric acid, and flushed with distilled water after each operation. The corrugated strips prepared in this way were mounted on the heater surface at a separation distance of one half pitch and fastened with a lattice clamp. The results were reduced in the form of the relation $q = f(\Delta T)$, where $\Delta T = T_W - T_S$. The partial laws are shown in Fig. 1.

The heat transfer with vapor generation in continuous corrugated metal structures depends on the characteristic dimension of the corrugations, the modulus, and as the modulus increases the heat-transfer intensity α_i is reduced, and for structures with f = 0.5 it approaches the level for boiling in a free volume. The heat transfer increases with increase of the strip width b. No influence of the corrugation material on α_i was observed.

From the partial results obtained one can formulate the following physical representation of the heat-transfer mechanism during vapor generation in a corrugated heat-pipe wick:

at low heat flux $q < 10^5$ W/m^2 at the bottom of the cavity of each channel formed by the corrugation profile a meniscus of the evaporating liquid is formed. The heat is transferred from the wall via elements of the corrugation contacting the wall and via the liquid layer to the interphase boundary;

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TABLE 1. Geometrical Parameters of the Corrugated Capillary Structures

Structure f	Metal thickness $\delta^* \cdot 10^3$, m	Corrugation height $h^* \cdot 10^3$, m	Pitch $t \cdot 10^3$, m	Number of corrugation in the heat pipe	Strip width $b \cdot 10^3$, m	Equivalent dimension $d_e\, 10^3$, m
0,2	0,1	0,21	0,64	49	10,5; 19,0	0,40
0,3	0,1	0,56	0,85	30	10,5; 20,7	0,49
0,5	0,1	0,95	1,68	18	5,5; 10,5; 20,7; 40,0	0,95

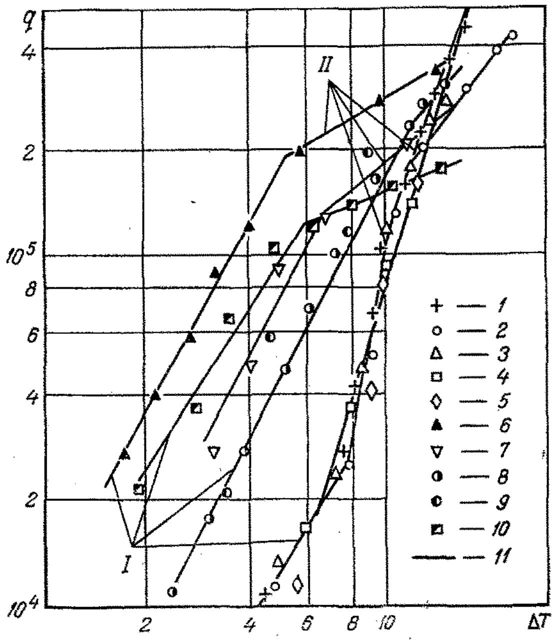

Fig. 1. Heat transfer under vapor formation in corrugated capillary structures: modulus f = 0.5; stainless steel: 1) b = 5.5 mm; 2) 10.2 mm; 3) 20.7 mm; 4) 40 mm; 5) modulus, b = 40 mm; f = 0.3; 6) stainless steel, b = 20.7 mm; 7) copper, b = 20.7 mm; 8) stainless steel, b = 10.5 mm; 9) copper, b = 10.5 mm; 10) f = 0.2, copper, b = 19 mm; 11) boiling in a free volume. q is in W/m^2, ΔT, K.

with increase of heat flux $q \geq 10^5$ W/m^2 vapor generating centers become active on the heater surface, and their density can be estimated as for conditions of boiling in a free volume

$$n_F = \left(\frac{r\rho''\Delta T}{\sigma T_s}\right)^k; \tag{1}$$

the vapor charge increases due to vaporization of the microlayer at each corrugation base and to the action of the vapor-generating centers.

For small q we observe fluctuations of the vapor–liquid mixture within the corrugations — this mechanism corresponds to section I of the curves in Fig. 1. In some of the series of tests, we made visual observations through a transparent model structure. For large q ($\approx 10^5$ W/m^2) the internal channels of the structure are filled with vapor, while the liquid wets the framework of the structure and the heater surface in the form of a microfilm — section II in Fig. 1.

The hydraulic resistance to the outflow of the vapor phase is determined by the cross section which is formed by the apertures at the places where the corrugated strip makes contact with the heater surface. The narrower are the strips, the larger are the butt joints and the larger is the effective outflow cross section S_{out} for the vapor, and the lower is the heat-transfer intensity. This also serves to explain the dependence of the heat transfer and the limiting power transfer on the corrugation width.

The heat-transfer coefficient can be estimated as the reciprocal of the thermal resistance of the microlayer of liquid:

$$\bar{\alpha}_i \sim \frac{\lambda'}{\bar{\delta}} . \tag{2}$$

We assume that the mean thickness of the microlayer $\bar{\delta}$ is formed in the growth process of the system of vapor bubbles whose birth and development is accomplished simultaneously at all the vapor generation centers of the heater surface.

A natural postulate is that the boiling mechanism in the corrugation channels is the same as in capillary or slot channels. Then one can use the known assumptions of the model for boiling in slots, as developed at OTIKhP [5].

To estimate the nature of the dependence of the liquid microfilm thickness on the main factors, we use the following relation [6]:

$$\bar{\delta} \sim \sqrt{\frac{\nu' \dot{R}}{\frac{1}{\rho'} \frac{dP}{dR}}} \sim \sqrt{\frac{\nu' (dl_c/d\tau)}{\frac{1}{\rho'} (dP''/dl_c)}} . \tag{3}$$

The rate of growth of the vapor column we estimate from the energy growth model

$$\frac{\pi d_c^2}{4} dl_c = \frac{q}{r\rho''} l_c t d\tau, \tag{4}$$

whence we have

$$\frac{dl_c}{d\tau} = \frac{4qtl_c}{\pi d_c^2 r\rho''} . \tag{5}$$

The pressure drop in the system vapor column — liquid in the volume beyond the channel where the vapor column is formed is determined by the hydraulic resistance to the outflow of liquid:

$$P'' - P_\infty = \text{const}\, \rho' w^2. \tag{6}$$

We find the liquid outflow rate from the mass balance equation — the growing vapor column is ejected from the vapor channel and the liquid phase:

$$n_F \frac{\pi d_c^2}{4} dl_c = w \frac{\pi d_e^2}{4} \varphi d\tau + w \frac{\pi d_e^2}{4} (1 - \varphi)\, d\tau. \tag{7}$$

Then

$$w = \frac{dl_c}{d\tau} \left(\frac{d_c}{d_e}\right)^2 n_F . \tag{8}$$

Differentiating Eq. (6) and allowing for Eqs. (5) and (8), we obtain

$$\frac{dP''}{dl_c} = \text{const}\, \rho' \left(\frac{q}{r\rho''} \frac{t}{\pi d_e^2} n_F\right)^2 l_c. \tag{9}$$

Substitution of Eqs. (5) and (9) into Eq. (3) gives the expression

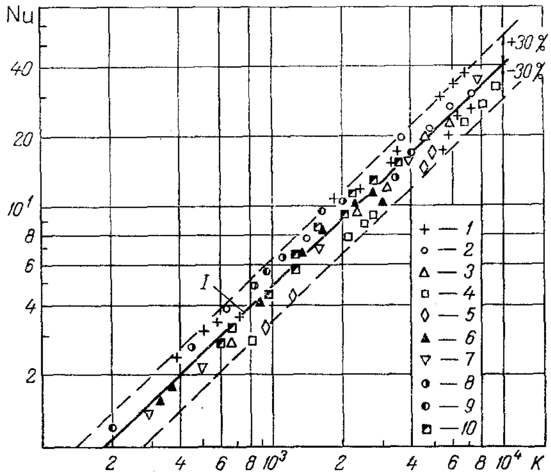

Fig. 2. Results of reducing the test data on heat transfer on the basis of Eq. (12). The remaining notation is the same as in Fig. 1. 1) Nu = $3.9 \cdot 10^{-3}$ K + 0.9.

$$\bar{\delta} = \text{const}\sqrt{\frac{\nu' r\rho''}{qt}}\left(\frac{d_e}{n_F}\right), \tag{10}$$

and the average heat-transfer coefficient $\bar{\alpha}_i$ is determined by the relation

$$\bar{\alpha}_i = \text{const}\frac{\lambda' n_F}{d_e}\sqrt{\frac{qt}{\nu' r\rho''}}. \tag{11}$$

Transforming to dimensionless (or correlation) form, we find

$$\mathrm{Nu} = C_* l_*^{2/3}\,\mathrm{Re}_*^{5/6}\,(t\cdot b)^{1/3}, \tag{12}$$

where

$$l_* = \frac{(r\rho'')^2\nu'}{\sigma T_s\lambda'};\quad \mathrm{Re}_* = \frac{qd_e}{r\rho''\nu'}$$

with the exponent k = 2 in Eq. (1).

We designate $K = \{l_*^{2/3}\mathrm{Re}_*^{5/6}(t\cdot b)^{1/3}\}$ and represent the results of reducing the test data in Nu–K coordinates (Fig. 2). The satisfactory correlation of the test data on the basis of the generalized relation (12) with $C_* = 3.9\cdot 10^{-3}$ is evidence of the validity of the physical model assumed.

In parallel with the study of heat transfer we determined the limiting heat flux from the intense growth of the heater surface temperature. The test results on limiting heat flux, obtained for various modules and structure strip widths at zero level of heat transfer relative to the heater surface are shown in Table 2. It follows from analysis of these data that an increase of structure strip width reduces the limiting heat-flux density. A reduction of the corrugation module leads also to the same result.

We can assume that the limiting heat depends on the hydraulic resistance to motion of the vapor–liquid medium in the channels of the wall part of the structure and the resistance to outflow of vapor at the places where the strip of corrugations makes contact.

The hydrodynamic heat-pipe model was used to develop the idea of the governing influence of the processes in the evaporation zone. The original hydrodynamic heat-pipe equation can be written in the form

$$\Delta p_{\text{cap}} \geqslant \Delta p' + \Delta p'' + \Delta p_{\text{M}}, \tag{13}$$

TABLE 2. Limiting Heat Flux $q_m \cdot 10^{-4}$, W/m²

Structure modulus f	Width of structure strip $b \cdot 10^3$ m			
	5,5	10,5	20,7	40,0
0,2	—	32	25	—
0,3	—	39,5; 36,0	29,6; 35,0; 28,5;	—
0,5	46	39,0; 44,0	33,0; 32,0	24

in which the terms are estimated in accordance with the assumptions made.

The pressure losses in the condensation and transport zones can probably be neglected, since the feeding of heat transfer agent to the internal channels is accomplished by excess of liquid in the external channels via the places where the strip of corrugations makes contact.

We determine the value of the available capillary potential

$$\Delta p_{\text{cap}} \simeq \frac{4\sigma}{d_e}. \tag{14}$$

Gravity forces are allowed for in the well-known way:

$$\Delta p_{\text{м}} = \pm \rho' g l_{\text{tr}} \sin\beta, \tag{15}$$

where the plus or minus sign corresponds to operation of the heat pipe in the direction of or against the gravity force.

We find the pressure losses in the vapor phase Δp" from the expression

$$\Delta p'' = \sum_i \zeta_i \rho'' w''^2, \tag{16}$$

where $w = \dfrac{Q_m}{r\rho'' s_{\text{out}}}$ is the rate of vapor generation.

Simultaneous consideration of Eqs. (14)-(16) gives the inequality

$$\frac{4\sigma}{d_e} \pm \rho' g l_{\text{tr}} \sin\beta \geqslant \text{const}_1 \left(\frac{b}{l_i}\right)^2 \frac{1}{\rho''} \left(\frac{Q_m}{r s_{\text{out}}}\right)^2 + \text{const}_2 \frac{\nu' l_i}{d_e^2} \left(\frac{Q_m}{r s_{\text{out}}}\right). \tag{17}$$

We solve Eq. (17) for the group (Q_m/rs_{out}):

$$\left(\frac{Q_m}{r s_{\text{out}}}\right) \frac{d_e^2 b^2}{\nu' \rho'' l_i^2} = \text{const}_1 \left(\sqrt{1 + \text{const}_2 \frac{d_e^4 b^2}{\nu'^2 \rho'' l_i^4} \left[\frac{4\sigma}{d_e} \pm \rho' g l_{\text{tr}} \sin\beta\right]} - 1\right). \tag{18}$$

We introduce the notation:

$$Y_0 = Q_m/rs_{\text{out}};\quad Y = Y_0 \frac{d_e^4 b^2}{\nu' \rho'' l_i^2};\quad X = \frac{d_e^4 b^2}{\nu'^2 \rho'' l_i^4} \left[\frac{4\sigma}{d_t} \pm \rho' g l_{\text{tr}} \sin\beta\right]. \tag{19}$$

Then Eq. (18) takes the form

$$Y = C_1(\sqrt{1 + C_2 X} - 1). \tag{20}$$

Equation (18) reflects the dependence of the limiting power transferred by the heat pipe Q_m on the geometric and regime parameters. For example, Q_m increases with increase of S_{out} and the length of the evaporation zone l_i with reduction of d_e and the width of the wick section b. Analysis of the term $\left(\dfrac{4\sigma}{d_e} + \rho' g l_e \sin\beta\right)$ shows that Q_m increases with increase of the slope angle from 0 to +90° and sharply decreases for changes of β from 0 to −37°. Figure 3 shows graphs of dependence of Q_m on β for the heat-pipe operating regimes. For angles $\beta \simeq -40°$ the heat-pipe hydrodynamic limit sets in, when the quantity Δp_{cal} becomes less that $\Delta p_{\text{м}}$, Eq. (15). This means that the capillary forces governed by the quantity $(4\sigma/d_e)$ cannot achieve transport of liquid to the evaporation zone against the gravity forces.

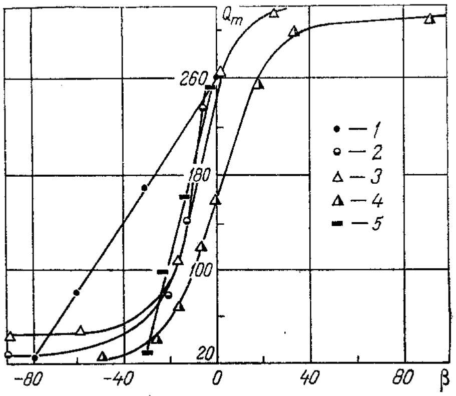

Fig. 3. The maximum power transferred Q_m (W) as a function of the orientation angle β (deg) for operating regimes of a heat pipe with a corrugated structure with modulus f and charge volume V_{ch}: 1) m = 0.3; V_{ch} = 2.13 cm³; 2) 0.3 and 3.7; 3) 0.2 and 5.8; 4) 0.2 and 2.17; 5) 0.5 and 2.6.

$(4\sigma/d_e)$ cannot achieve transport of liquid to the evaporation zone against the gravity forces.

Figure 4 shows the results of reducing data from a series of tests of heat-pipe specimens with corrugated structures set up by the authors of [1, 7] at slope angles from −37 to +90°, and also data of modeling limiting heat fluxes in the evaporation zone obtained by the authors. The results are correlated satisfactorily by Eq. (20) with C_1 = 8.0 and C_2 = 0.0183. The tests of [7] have shown that heat pipes with corrugated wicks operate in a stable manner at angles from −37 to −90°, which cannot be explained in the framework of the relations assumed above.

A detailed analysis of possible causes of these peculiarities in the operation of heat pipes with continuous corrugated structures has shown that in operation "against" the forces of gravity the following mechanism is most likely. If $(4\sigma/d_e)$ is less than $\rho' gh$, where h is the height of the capillary volume of liquid in the internal channels of the corrugated structure, then the liquid is removed in the internal channels, with a corresponding decrease of level. Here the liquid remains at the places where the corrugated structure touches the surface. This means that under these conditions on application of the heat flux the liquid motion will occur in the channels formed by the profile of corrugations and the heat-pipe surface at the places of contact (Fig. 5a). For this case we write the heat-pipe equation in differential form

$$-\frac{\sigma}{R^2}\,dR = 32\,\frac{\rho'}{d_e}\left(\frac{Q_m}{Nr\rho' S_\ell}\right)^2 dz. \tag{21}$$

We rewrite Eq. (21) in the form

$$-\frac{dR}{R^2} = C_3\,\frac{\Pi_\ell}{S_\ell}\,\frac{1}{\sigma\rho'}\left(\frac{Q_m}{Nr}\right)^2 dz. \tag{22}$$

Expressing the wetted area S_ℓ and the perimeter Π_ℓ as certain functions of the coordinates, we obtain

$$-\left[\frac{\varphi_1(x)}{x}\right]^3 \frac{1}{\varphi_2(x)}\,dx = C_3\,\frac{1}{\sigma\rho'}\,\frac{1}{R}\left(\frac{Q_m}{Nr}\right)^2 dz, \tag{23}$$

where

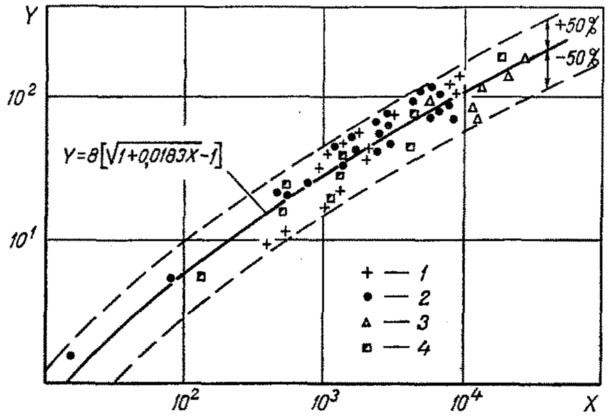

Fig. 4. Generalization of the experimental data on the limiting characteristics of heat pipes using Eq. (21): 1) modulus f = 0.2; 2) 0.3; 3) 0.5; 4) results of modeling (from Table 2).

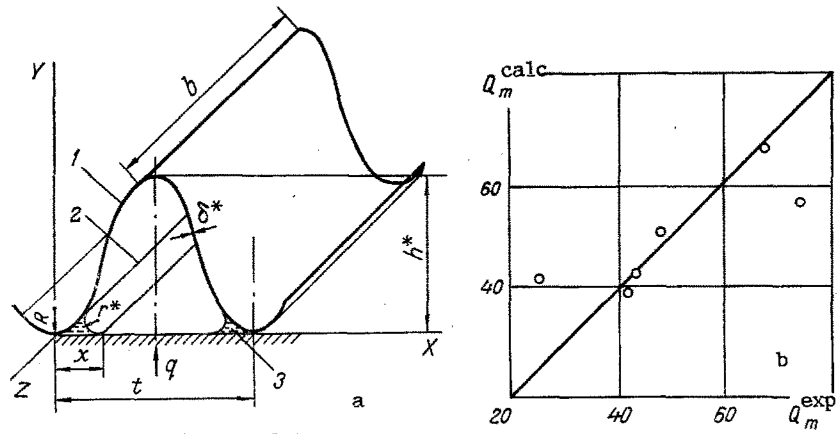

Fig. 5. Maximum heat-transfer capability of a heat pipe with a corrugated structure at slope angles β from −37 to −90°: a) to explain the mechanism of operation of a heat pipe against gravity forces (1, corrugation profile; 2, internal channel; 3, liquid); b) results of comparing the experimental and theoretical investigations.

$$\varphi_1(x) = S_\ell = x^2\left[\frac{A}{2} + \frac{\pi}{2}A^2 + 2A^3\right];$$

$$\varphi_2(x) = \Pi_\ell = 2x;\quad A = \frac{x}{4R}.$$

Integrating Eq. (23) and assuming that $(x/R) \ll 1$, we obtain the solution for Q_m:

$$Q_m = C_3 t^2 Nr\sqrt{\frac{\rho'\sigma}{b\,|\sin\beta|}}. \tag{24}$$

With Eq. (24) and C_3 = 0.0469 we can evaluate the maximum heat-transfer capability of a heat pipe with a corrugated structure at slope angles from −37 to −90°. The results of comparing the experimental and theoretical investigations are shown in Fig. 5b.

NOTATION

b, width of corrugated strip; d, diameter; k, exponent; l, length; L, Labuntsov dimension; f) modulus; N) number of corrugations; Nu, Nusselt number; n, density of vapor generation centers; P, p, pressure; Q, heat flux; q, heat flux density; R, radius; Re*, modified Reynolds number; r, specific heat of vaporization; S, cross section; T, temperature; t, pitch; ω, velocity; α, heat-transfer coefficient; β, slope angle; δ, thickness; ζ, resistance coefficient; λ, thermal conductivity; ν, kinematic viscosity; ρ, density; σ, surface tension coefficient; τ, time; φ, vapor content. Subscripts: F, free volume; m, maximum; s, parameters on the saturation line; out, outflow; liq, wetted by liquid; ev, evaporation; cap, capillary; M, forces of gravity; c, column; w, wall; tr, transport; equiv, equivalent; ('), liquid; ("), vapor.

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MACROSCOPIC BOUNDARY ANGLES OF WETTING OF SINTERED CAPILLARY STRUCTURES OF HEAT PIPES

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We measured the macroscopic boundary angles of wetting by water, acetone, and pentane of the capillary structures of antigravity heat pipes in the 293-363°K temperature range.

Most of the studies investigating the wetting of capillary structures of heat pipes have been devoted to metal-wire materials [1-3] and have practically no relation to sintered-powder capillary structures, which have also come to be rather widely used in heat pipes.

In order to estimate the wetting of capillary structures, investigators have used both conventional methods of studying compact (nonporous) materials [4, 5] and improved methods or methods specially designed for porous bodies [1, 2], in which account is taken of the variation of the structure of the material surface and of the influence of porosity on the wetting process. The boundary angle may be regarded as a fixed physicochemical characteristic only for systems which are in a state of thermodynamic equilibrium. For capillary structures of heat pipes it is difficult to regard the angle in this way because their

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