HEAT TRANSFER IN EVAPORATION AND CONDENSATION ZONES IN HEAT PIPES WITH INTENSIVE HEATING OF THEIR ENDS

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An experimental study is made of heat transfer in the condensation zone and during boiling in the heating zone in heat pipes with intensively heated ends. Generalized relations are presented.

Heat pipes have come into wide use to ensure the required thermal operating conditions for elements in many power plants. Special heat pipes in which heat is transferred through the end of the evaporative part of the pipe must be used to cool these elements in certain cases. The heat transfer capacity of such a pipe is determined in large part by the rate of heat transfer in its evaporation and condensation zones. Heat transfer occurs most intensively in the boiling regime.

Relatively few works [1-11] have been devoted to the study of heat transfer during boiling and condensation of a heat carrier on surfaces with a gridlike structure, and even fewer of these researches have been conducted directly for heat pipe conditions. In the latter case, heat pipes with radial heating of the evaporative part have been investigated. However, these results cannot always be used for specific heat pipes, particularly those with end delivery of the heat. In connection with this, we experimentally studied heat transfer in the heating zone (during the boiling regime) and condensation zone for heat pipes with an end heat supply.

Figure 1 shows the heat pipe and experimental stand.

The experiment stand includes a powerful heat supply consisting of DKsTV-15000 xenon discharge lamps; equipment for measuring heat flow density and distribution [12]; several other devices for monitoring and control, shown in the figure. The lamps are connected by means of a high-voltage ignition system 6, 7. The ballast resistor makes it possible to vary the voltage on the lamp electrodes and thus change the power they consume. A flat or shaped cooled metal reflector 4 is used to obtain a uniform or area-varying heat flux. Heat fluxes up to 10^6 W/m^2 are reached on the above stand, and heat fluxes up to $2.5 \cdot 10^6$ W/m^2 are reached in the overload regime.

We studied five heat pipes made of copper: Four of them had a grid-type wick, while one had a metal-fiber wick. The grid wick was made of four simply woven layers of 12Kh18N9T stainless steel. The first two layers, adjacent to the heat-emitting surface, had cells of the dimensions 0.08 × 0.08 mm. The two subsequent layers had the dimensions 0.14 × 0.14 mm on the interface side. To intensify heat transfer, all of the wicks of the grid-type pipes had 1-mm-diameter vapor-conducting channels in the evaporation zone. The distance between them (5 mm) was chosen on the basis of visual observation of the boiling of a thin liquid layer on a free surface [8] and corresponded for the investigated range of q to a moderate spacing of the active-vapor formation centers. Three of the grid pipes had wick inserts which made it possible to more uniformly supply the evaporation zone with heat carrier.

The metal-fiber copper wick had fibers 70 μm in diameter and 5-10 mm long. The porosity of this wick was 85%. There were no vapor channels here. Table 1 shows characteristics of the investigated heat pipes (all dimensions are given in mm).

The heat pipes were equipped with 14 Nichrome—Constantan thermocouples, eight of which were caulked onto the outer surface of the evaporator end, three of which were caulked onto the outer surface of the condenser, and three of which were located in the vapor channel of the heat pipe.

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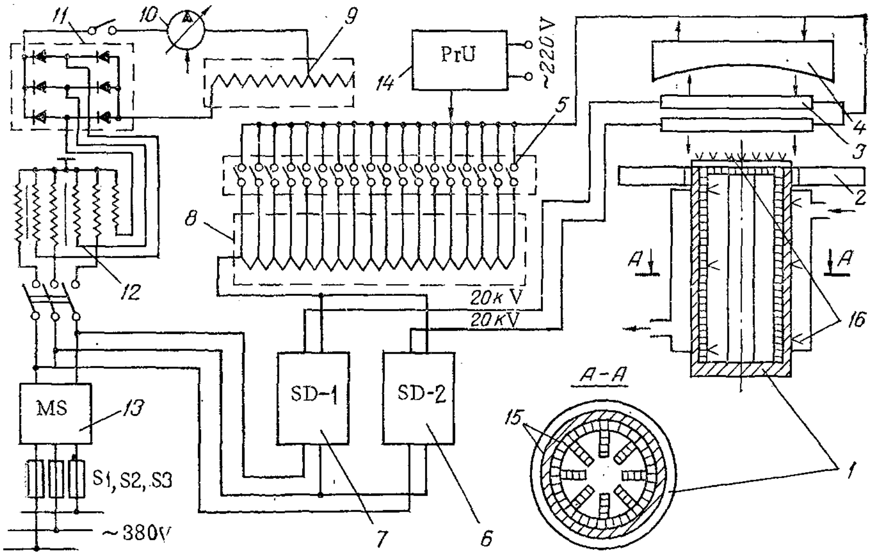

Fig. 1. Experimental stand and investigated heat pipes: 1) heat pipe; 2) cooled shield; 3) radiator; 4) cooled reflectors; 5) contactor group; 6, 7) starting devices; 8) ballast resistor; 9) rheostatic controller; 10) centrifugal pump for coolant water system; 11) rectifier; 12) power transformer; 13) magnetic starter; 14) program unit; 15) wick inserts; 16) thermocouples.

TABLE 1. Geometric Characteristics of Heat Pipes

Heat pipe	L	d_x	d_{in}	δ_k^e	δ_k^c	δ_k^{ar}	No. of wick inserts
TT 1-0	120	64	58	0,6	1,3	—	0
TT 2-4	120	64	58	0,6	1,6	0,8	4
TT 3-8	120	64	58	0,6	1,8	0,8	8
TT 4 k8	60	64	58	0,6	1,8	0,8	8
TT 5-0 MV	120	64	58	0,6	1,6	—	0

The heat carrier was distilled water (H_2O), water-alcohol mixtures (50% H_2O + 50% C_2H_5OH; 75% H_2O + 25% C_2H_5OH), and ethyl alcohol (C_2H_5OH). In the tests, we measured the heat flux on the pipe end, the temperatures of the outer surfaces of the evaporator and condenser, and the temperatures in the vapor channel. For each pipe we varied the thermal load ($q = 5\cdot10^4$–$100\cdot10^4$ W/m²), the type of heat carrier, the area of the condensation zone, and the rate of its cooling.

The resulting empirical data allows us to calculate the mean heat-transfer coefficients in the evaporation and condensation zones:

$$\bar{\alpha}_e = \frac{q}{\bar{t}_e - t_v}; \quad \bar{\alpha}_c = \frac{q\dfrac{F_e}{F_c}}{t_v - \bar{t}_c}. \tag{1}$$

Here $\bar{t}_e$ and $\bar{t}_c$ are the mean temperatures of the inside surfaces in the evaporation and condensation zones; F_e and F_c are the areas of the evaporator and condenser; t_v is the vapor temperature.

Figure 2 shows the relation $\alpha = f(q)$ for the investigated heat pipes. Data from [9] is also shown here. The experimental points in the figure correspond to different saturation temperatures in the vapor channel. The character of the effect of q on heat transfer in the evaporation zone of the pipe shows that the investigated pipes operate in the boiling regime. The authors of [11] explained the similar dependence of α on q without using boiling regime laws. Analyzing the heat-transfer data, we can note that up to $q \approx 40\cdot10^4$ W/m², heat transfer occurs on the metal-fiber wick in roughly the same manner as on the grid-type wick; in the range $q = (40\text{–}50)\cdot10^4$ W/m², heat transfer on the metal-fiber wick decreases compared to the grid wick and then remains nearly constant with a change in q. Heat transfer on the grid

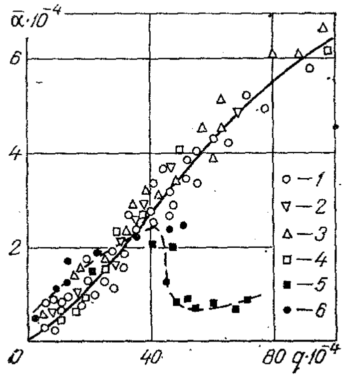

Fig. 2

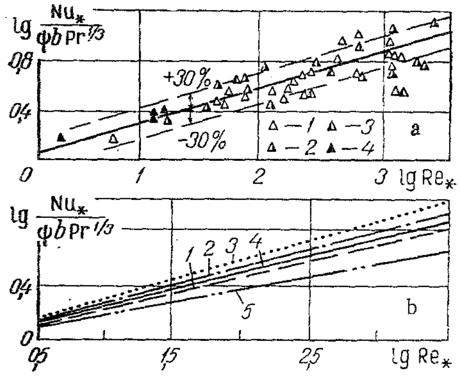

Fig. 3

Fig. 2. Dependence of the mean heat-transfer coefficients in the evaporation zone of heat pipes on the thermal load ($\bar{\alpha}\cdot 10^{-4}$ W/m² K; $q\cdot 10^{-4}$ W/m²): 1) HP 1-0; 2) HP 2-4; 3) HP 3-8; 4) HP 4-8; 5) HP 5-0 MV; 6) [9].

Fig. 3. Generalizing relations on heat transfer in the heating zone: a) heat carrier H_2O; 2) 25% C_2H_5OH; 3) 50% C_2H_5OH; 4) C_2H_5OH; 1) for HP 1-0 (k = 0.0262); 2) for HP 2-4 (k = 0.0243); 3) for HP 3-8 (k = 0.0233); 4) single relation for all HP (k = 0.026); 5) generalizing relation from [8].

wick increases continuously with an increase in q. In studying heat transfer on surfaces with a pore-capillary structure, the problem of selecting dimensionless complexes in analyzing and generalizing the test data is a very complicated one. Most studies have been conducted at relatively low heat fluxes. There are generalizations [10, 11] which describe heat transfer to pore-capillary structures up to large q. Analysis of the data obtained in accordance with the approaches developed in these works has shown that the empirical data is satisfactorily described by the proposed relations at low values of q. The difference between the test data and the generalizing relations increases with an increase in the thermal load, the test data being located above the generalizing lines. The substantial intensification of heat transfer is evidently due to the fact that here we used a composite grid structure with vapor channels, and the heat pipes were equipped with additional wick inserts.

The following relation has been successfully used in several works to generalize test data

$$\mathrm{Nu} = f(\mathrm{Re},\ \mathrm{Pr},\ \ldots). \tag{2}$$

To account for certain specific conditions of occurrence of the process (type of heat carrier, design parameters), special complexes have been added to the right side of this relation.

It was established in [8] that, when vapor channels are present, the exponent with the Reynolds number Re depends on the number and dimensions of these channels. It was found in generalizing our test data that heat transfer in boiling in the heating zone of an end heat pipe with a grid wick is satisfactorily described by the generalizing relation

$$\mathrm{Nu}_* = \psi b\, \mathrm{Pr}^{1/3}\, \mathrm{Re}_*^{0.66(1-km)}, \tag{3}$$

where the numbers Nu_* and Re_* ($\mathrm{Nu}_* = \alpha l_*/\lambda$, $\mathrm{Re}_* = q l_*/r\rho''\nu$, $\mathrm{Pr} = \nu/a$) were calculated from the characteristic dimension l_* – the radius of a vapor nucleus [13] ($l_* = c_p\rho'\sigma T/(r\rho'')^2$); ψ is a dimensionless complex characterizing the design parameters of the wick and the type of heat carrier ($\psi = 0.88\,(\rho'_{H_2O}/\rho')^{-0.32} \times (p_{at}/p'')^{0.062}\, m^{1.37} \exp(0.179f - 2.46 + \varkappa)$); m is a coefficient accounting for the presence of vapor channels ($m = F_{ga}/F_e$); $f = (d_{eqv} - d_p)d_p$; $b = 0.075 \times \left[1 + 10\left(\frac{\rho''}{\rho' + \rho''}\right)^{2/3}\right]$; F_{ga} is the area occupied by gaps; p_{at} is the atmospheric pressure.

In Eq. (3) $\varkappa = 1.0$ at m = 0 and $\varkappa = 0$ at $m \neq 0$, while the value of the parameter k depends on the presence and number of wick inserts.

Figure 3a shows test data on heat transfer in the boiling zone of an end heat pipe with eight inserts and Eq. (3) (k = 0.0233), which generalizes the data. Data for all of the in-

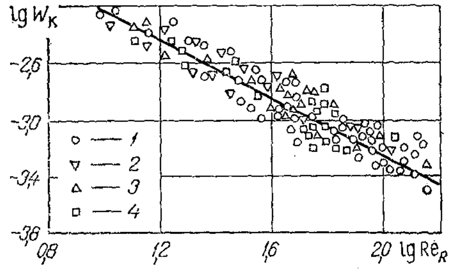

Fig. 4. Generalizing relation for heat transfer in the condensation zone $W_c = \left(\mathrm{St}\, N_p^n \left(\frac{F_c}{\varepsilon F_k}\right)^{1,17}\right) \Big/ \left(\mathrm{Pr}^{1,3}\left(\frac{p_v^2}{k}\right)^{0,47}\left(\frac{b}{R_v}\right)^{0.8}\mathrm{Ku}^{0.4}\right)$: HP 1-0; 2) HP 2-4; 3) HP 3-8; 4) HP 4-8.

vestigated heat carriers is shown. Nearly all of the empirical points (≈80%) are generalized by Eq. (3) with an accuracy no worse than ±30%.

The generalizing relations for heat pipes with different number of inserts are shown in Fig. 3b (to avoid darkening the figure, the empirical points are not shown). The figure also shows values of k for each heat pipe. With a somewhat greater scatter (±50%), all of the test results for α_e can be generalized by a single relation (3) with the parameter k = 0.026. Also shown here (line 5) is the generalizing relation from [8], obtained in the boiling of distilled water on a steel surface covered with a gridlike structure with vapor channels. There is a difference between the test data in [8] and our results (the latter are above the the data in [8]), which decreases with a decrease in the number Re_*. This difference can be explained by certain differences in the test conditions and by the effect of the wall material on which boiling occurs on heat transfer. Our results were obtained directly for an end heat pipe, in contrast to the results in [8], corresponding to a model — although one that approached the working conditions of heat pipes as closely as possible. The substantial effect of the wall material on heat transfer in the boiling of freons on porous surfaces was also seen in [8], where it was noted that heat transfer occurs more slowly during boiling on a steel surface than in the case of copper heat pipes.

In the condensation zone of the heat pipe, the test data on heat transfer is satisfactorily generalized by a relation which contains similitude numbers proposed in [1]:

$$\mathrm{St}\, N_p^n \mathrm{Re}_R \left(\frac{F_c}{\varepsilon F_k}\right)^{1,17} = c\,\mathrm{Pr}^{1,3}\left(\frac{R_v^2}{K}\right)^{0.47}\left(\frac{b}{R_v}\right)^{0.8}\mathrm{Ku}^{0.4}, \quad (4)$$

where $c = 6.2 \cdot 10^{-2}$; n = 0.7 at p > 0.1 bar; $c = 0.78 \cdot 10^{-2}$; n = 0.833 at p < 0.1 bar; $\mathrm{St} = \alpha/c_p'\rho'\vartheta'$ is the Stanton number; $\mathrm{Ku} = r/[c_p'(t' - t_c')]$, Kutateladze number; $\mathrm{Re}_R = R_v\vartheta_R/\nu''$, Reynolds number; $N_p = a\sigma/\rho'' R_v$, a pressure parameter (a is the meniscus form factor, with a = 2 for a spherical meniscus); $K = 4.305 \cdot 10^{-10}(\bar{b}/r)^{+0.5}$, coefficient of permeability of the wick; F_w, cross-sectional area of the wick in the condensation zone; ϑ_R, radial velocity of the vapor.

It is apparent from Fig. 4 that nearly all of the empirical points, with an accuracy no worse than ±15%, are described by Eq. (4).

NOTATION

α, heat-transfer coefficient; λ, a, ν, thermal conductivity, diffusivity and kinematic viscosity; r, latent heat of vaporization; ρ, density; σ, surface tension; q, heat flux; L, length; d, diameter; δ, thickness of wick; c_p, specific heat; p, pressure; d_p, diameter of pores; $d_{p\ eqv}$, equivalent diameter of pores; R_v, radius of vapor channel; F, area; ν, velocity; ε, porosity. Indices: ', coolant in liquid form; ", vapor; c, condensation zone; w, wall; max, maximum; x, external; in, internal; k, wick; e, evaporator; v, vapor.

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