# METHOD OF COMPUTING THE THERMAL RESISTANCE OF LOW-TEMPERATURE HEAT PIPES WITH METAL-FIBER WICKS

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A method is elucidated for computing the thermal resistance of heat pipes with metal-fiber wicks, which has been developed on the basis of analytic and experimental investigations of heat-pipe characteristics.

The thermal resistance of a heat pipe is one of the main characteristics of effective heat elimination from a heat-liberating object into the environment or to another heat sink. The thermal resistance $R_t^{co}$ is ordinarily understood to be the ratio between the heat drop $\Delta t_t$ and the total heat flux transmitted by the heat pipe [1, 2], although methodologically it is more correct (as will be shown below) to compute $R_t^{co}$ by using the heat-flux density in the heat-supply zone $q_h$. The resistivities of the heat-supply and heat-elimination zones yield the main contribution to the heat-pipe thermal resistance.

The thermal resistance of the heat-pipe heating zone depends on the modal parameters of their operation, the thermophysical properties of the working fluid, and the geometric, structural, and thermophysical characteristics of the wicks. Two modes of operation exist for the heating zone of heat pipes with metal-fiber wicks: evaporative (for $q$ = const and small heat loads, e.g., to 1 W/cm$^2$ for water), and the boiling mode (as experiment shows, for $q_h \gtrsim 1$ W/cm$^2$). The heat transfer in the heating zone is realized in the evaporative mode by heat conduction through the fluid-saturated metal-fiber wick and by evaporation from the surface turned to the vapor channel. In this case the thermal resistance is practically independent of the modal parameters of the process and is determined for a completely saturated wick (without taking account of the deepening of the evaporation surface) by its equivalent thickness, its effective heat conduction, and the quality of its fastening to the casing:

$$(R_h^{co})_{evap} = \frac{\delta_w}{\lambda_{ef}} + R_{c.t} \tag{1}$$

The effective heat conduction of wicks baked from monodisperse discrete copper fibers is determined from the equation [3]

$$\lambda_{ef} = \lambda_1 \left[ (1-\Pi)^2 M_{ef} + \Pi^2\theta + \frac{4\theta\Pi(1-\Pi)}{1+\theta} \right],$$
$$\theta = \frac{\lambda_2}{\lambda_1}; \quad M_{ef} = \bar{y} + \frac{2B(1-\bar{y})\sqrt{1-\bar{y}^2}}{(1-\bar{y}) + B\sqrt{1-\bar{y}^2}}; \tag{2}$$
$$B = \sqrt{\pi}\,\frac{\theta}{1-\theta}\left(\frac{1}{1-\theta}\ln\frac{1}{\theta} - 1\right); \quad \bar{y} = 0.043\exp\left(\lg\frac{l_f}{d_f}\right).$$

An analysis of experimental results [4] shows that the boiling mode is the fundamental mode of operation of low-temperature heat pipes with metal-fiber wicks. The heat transfer is intensified during boiling because of turbulization of the fluid in the wick and the additional transport of heat by the vapor phase. In this case, the thermal resistance depends on the geometric and structural characteristics of the wick, the physical properties of the working fluid, the modal parameters, and the orientation in the gravity force field, and can be computed for $\Delta t < 4\sigma T_{sat}/r\rho_v d_f$ by means of the formula

$$(R_h^{co})_{boil} = 1.17\, q_h^{-2/3}\, \varepsilon_F^{1/3} \left(\frac{\lambda_2^2}{\sigma\nu T_{sat}}\right)^{-1/3} n^{-1}, \tag{3}$$

for $\varphi = 0°$ $n = 1$, for $0° < \varphi \leqslant 90°$ $n = 1.1 - 1.5$;

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$$\varepsilon_F = \frac{F_h + F_f}{F_h} = \frac{4F_w(1-\Pi)}{\Omega_m d_f} + 1. \quad (3)$$

The quantity $\varepsilon_F$ takes account of the influence of the wick structural characteristics on the intensity of the heat-transfer properties. The presence of this influence is explained by the fact that a metal-fiber wick is a structure of the volume mesh type whose fiber surface $F_f$ can considerably exceed the inner surface of the heat-pipe casing in the heating zone $F_h$. The effect of the heat-pipe orientation in the gravity force field is characterized by the factor n. The reduction in the thermal resistance of the heating zone with the increase in slope is due to the overflow of a quantity of fluid from the pores of the wick in the condensation zone and by the partial drainage of the wick in the heating zone.

The thermal resistance of the condensation zone consists of the thermal resistivity of the wick, the condensate film, and the wick contact with the casing:

$$R_c^{co} = R_w + R_{fi} + R_{c.t} \quad (4)$$

The wick resistivity $R_w$, whose magnitude is determined by the ratio between the equivalent wick thickness and its effective heat conductivity, yields the main fraction of $R_c^{co}$.

A condensate film, whose thermal resistivity equals the ratio between the film thickness and the heat conductivity of the working fluid, is formed on the wick surface during condensation of the heat carrier vapors. The experimentally determined fluid film thickness (water, ethyl and methyl alcohol) is ($\Pi$ = 65–96%, $d_f$ = 20–70 μm, $l_f$ = 3–10 mm), on the surface of metal-fiber wicks, ~10 μm.

The thermal resistance of the wick contact with the casing is due to the technology of heat-pipe fabrication. Consequently, it has been established that

$$\begin{aligned} &\text{for } \Pi \geqslant 80\ \% \quad R_{c.t} \approx 0, \\ &\text{for } \Pi < 80\ \% \quad R_{c.t} \approx 5\cdot 10^{-5}\ \text{m}^2\cdot{}^\circ\text{K/W} \end{aligned} \quad (5)$$

The thermal resistivity of the phase transition and the presence of gases which don't condense and are practically always present although in negligible quantities despite the careful preparation of the working fluid and the degassing of the inner cavity of the heat pipes (at 300°C for 5 h, with the vacuum $10^{-3}$ mm Hg) exert a significant influence on the intensity of the condensation in the heat-pipe operating range for low values of the vapor pressure $p_{sat} < 0.1$ bar. The presence of gases which do not condense was negligible in the heat pipes investigated since the heat-pipe condensation zones were characterized by high isothermy.

The influence of the phase transition and the presence of the gases which do not condense cannot possibly be taken into account analytically at this time. The thermal resistivity of the condensation zone during operation of low-temperature heat pipes in the reduced saturation pressure domain is determined sufficiently accurately by the empirical formula [5]

$$R_c^{co} = \frac{R_w + R_{fi} + R_{c.t}}{\sqrt{p_{sat}/p_{sat}^*}}, \quad (6)$$

TABLE 1. Structural Characteristics of Cylindrical Heat Pipes

| Heat pipe number | Geometric dimensions | | | | | | Wick characteristics | | | | Thermal resistance | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $d_t$ | $d_{in}$ | $d_{v.c}$ | $L_t$ | $L_h$ | $L_c$ | $\delta_w$, mm | $\Pi$, % | $d_f$, μm | $\lambda_{ef}^p$, W/m·°K | $R_c^{co}\cdot 10^5$, m²·°K/W | $R_h^{co}\cdot 10^5$, m²·°K/W | $R_t^{co}\cdot 10^5$, m²·°K/W |
| 1 | 28 | 24 | 20,4 | 500 | 25—150 | 200 | 1,7 | 87 | 40 | 2,8 | 57,7 | 7,3 | 65 |
| 2 | 28 | 24 | 22,8 | 500 | 25; 150 | 200 | 0,5 | 83 | 40 | 4,3 | 13,1 | 5,5 | 18,6 |
| 3 | 28 | 24 | 20,4 | 500 | 25; 150 | 200 | 1,7 | 68 | 40 | 13,4 | 18,4 | 9,8 | 28,2 |
| 4 | 6 | 5 | 3,4 | 250 | 50 | 100 | 0,8 | 93 | 20 | 1,5 | 45 | 5,7 | 50,7 |
| 5 | 10 | 8 | 5,4 | 250 | 100 | 100 | 1,3 | 89 | 20 | 2,7 | 41 | 7,8 | 48,8 |
| 6 | 10 | 8 | 5,8 | 250 | 50 | 100 | 1,1 | 88 | 20 | 3,1 | 31,6 | 7,6 | 39,2 |
| 7 | 10 | 8 | 5,9 | 250 | 50 | 100 | 1,05 | 86 | 20 | 3,9 | 25 | 7,9 | 32,9 |
| 8 | 10 | 8 | 4,7 | 250 | 50 | 100 | 1,65 | 83 | 20 | 5,5 | 24,4 | 9,4 | 33,8 |
| 9 | 10 | 8 | 4,4 | 450 | 50 | 100 | 1,8 | 77 | 20 | 9,3 | 20,9 | 10,6 | 31,5 |
| 10 | 12 | 10 | 8,8 | 250 | 50; 100 | 100 | 0,55 | 77 | 20 | 9,3 | 12,1 | 7,8 | 19,9 |
| 11 | 10 | 8 | 4,2 | 250 | 50 | 100 | 1,9 | 76 | 20 | 10,3 | 20,2 | 11,0 | 31,2 |
| 12 | 18 | 16 | 13,9 | 225 | 30; 60 | 100 | 1,05 | 75 | 20 | 10,7 | 15,7 | 9,8 | 25,5 |
| 13 | 14 | 9 | 4,6 | 350 | 50 | 80 | 2,2 | 63 | 50 | 16,0 | 16,3 | 9,8 | 26,1 |
| 14 | 28 | 24 | 20,4 | 500 | 50 | 200 | 1,7 | 87 | 40 | 2,2 | 77 | 13,8 | 90,8 |
| 15 | 18 | 16 | 14 | 225 | 50 | 100 | 1,0 | 75 | 20 | 9,9 | 19,2 | 18 | 37,2 |

TABLE 2. Structural Characteristics of Plane Heat Pipes

| Heat pipe number | Geometric dimensions, | | | | | | Wick characteristics | | | | Thermal resistance | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | $a$ | $h$ | $\delta_{wa}$ | $L_t$ | $L_h$ | $L_c$ | $\delta_w$, mm | $\Pi$, % | $d_f$, μm | $\lambda_{ef}^{co}$, W/m²·°K | $R_c^{co}\cdot 10^5$, m²·°K/W | $R_h^{co}\cdot 10^5$, m²·°K/W | $R_t^{co}\cdot 10^5$, m²·°K/W |
| 16 | 16 | 8 | 1 | 300 | 100 | 100 | 0,5 | 85 | 50 | 3,3 | 22,9 | 5,0 | 27,9 |
| 17 | 8 | 3 | 1 | 300 | 100 | 100 | 0,5 | 75 | 50 | 7,7 | 13,1 | 5,7 | 18,8 |
| 18 | 23 | 2 | 1 | 300 | 100 | 100 | 0,5 | 75 | 50 | 7,7 | 13,1 | 5,8 | 18,9 |

where $0.03 \le p_{sat} \le 0.1$ bar; $p_{sat}^* = 0.1$ bar.

The thermal resistance of the transport zone, caused by the presence of a temperature gradient in the vapor, is negligible compared to the resistivities $R_h^{co}$ and $R_c^{co}$. The thermal resistance of the heating and condensation zone casings is insignificant in the resistance chain and can often be neglected. Hence, the thermal resistance of heat pipes can be computed from the equations

$$R_t^{co} = \frac{\Delta t_t}{Q} = \frac{R_h^{co}}{F_h} + \frac{R_c^{co}}{F_c}, \tag{7}$$

$$R_t^{co} = \frac{\Delta t_t}{q_h} = R_h^{co} + R_c^{co}\frac{F_h}{F_c}. \tag{8}$$

An analysis of (7) for $F_h = F_c$ shows that the thermal resistance of heat pipes depends not only on the modal parameters of its operation, the wick characteristics, and the heat carrier, but also on the size of the condensation and heating zones, i.e., the same heat pipe can have quite distinct values of $R_t^{co}$ depending on $F_c(F_h)$. Considerable difficulties occur even in operating with the thermal resistance concept when comparing heat pipes with different geometric constructions. The thermal resistance of a heat pipe computed by (8) is determined only by the intensity of the heat-transfer processes taking place within the apparatus, and compliance with the condition $F_h = F_c$ is ordinarily sufficient for an estimate of the thermal resistivity of heat pipes of different construction. Conservation of the adequacy of the modal operating parameters is necessary for a more accurate comparison.

The method of computing the thermal resistance of heat pipes with metal-fiber wicks includes the following steps:

1) determination of the wick effective heat conductivity by (2);

2) computation of the thermal resistivity of the wick, the condensate film, and the wick contact with the casing;

3) computation of the thermal resistivity of the condensation zone by (4) for $p_{sat} > 0.1$ bar, and by (6) for $p_{sat} < 0.1$ bar;

4) computation of the thermal resistance of the heating zone in the evaporative mode ($q_h < 1$ W/cm²) by (1) and in the boiling mode ($q_h \gtrsim 1$ W/cm²) by (3);

5) determination of the zone length filled by the overflowing fluid during heat-pipe operation against the gravitation forces ($\varphi > 0°$) [6]:

$$L_1 = \left\{\left[e^{mBL_t}\left(1+\frac{F_{v.c}}{F_w \Pi}\right)-1\right] - \sqrt{\left[e^{mBL_t}\left(1+\frac{F_{v.c}}{F_w \Pi}\right)-1\right]^2 - 2\left[1-e^{mBL_t}\left(1-mBL_t\right)\right]}\right\}(mB)^{-1},$$

$$B = \rho_{fl} g \sin\varphi/\Delta p_{ch}, \quad \Delta p_{ch} = 4\sigma/D_m, \tag{9}$$

$$m \approx S/3;\; S = 0.63 \ln[0.45(D_m - 45)],\; 50 < D_m < 200 \text{ mm};$$

6) determination of the heating and condensation zones surface for cylindrical heat pipes

$$F_h = \pi d_{in} L_h, \quad F_c = \pi d_{v.c}(L_c - L_1) \tag{10}$$

and plane heat pipes

$$F_h = 2(a+h)L_h, \quad F_c = 2(a + h - 2\delta_w)(L_c - L_1); \tag{11}$$

7) computation of the thermal resistance of heat pipes by (8).

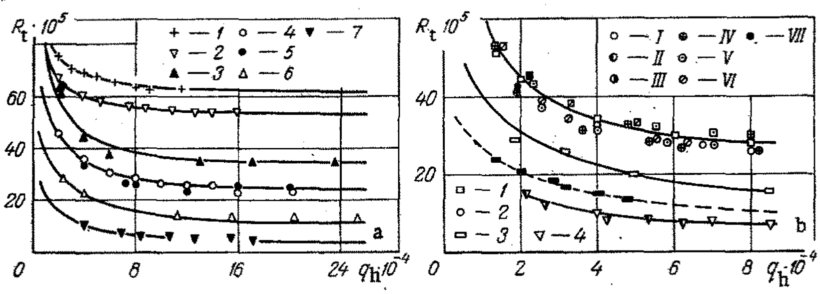

Fig. 1. Thermal resistance of low temperature heat pipes with metal-fiber wicks. a) For a horizontal orientation: 1) Heating pipe HP No. 5; 2) No. 1 ($L_h$ = 150 mm); 3) No. 4; 4) No. 6; 5) No. 8; 6) No. 1 ($L_h$ = 25 mm); 7) No. 2 ($L_h$ = 25 mm). b) During operation against the gravitational force: 1) HP No. 6; 2) No. 8; 3) No. 12; 4) No. 2 ($L_h$ = 25 mm); I) $\varphi = 0°$; II) 5; III) 10; IV) 15; V) 30; VI) 45; VII) 90°; the shading denotes the slope for all kinds of point, e.g., open circle 0°, . . . , completely blackened 90°. The curves are for a computation by (8); the dashes are for HP No. 12 operating against gravity for $\varphi$ = 90°, and the solid lines are for the horizontal position $\varphi$ = 0°. $R_t$, $m^2 \cdot °K/W$; $q_h$, $W/m^2$.

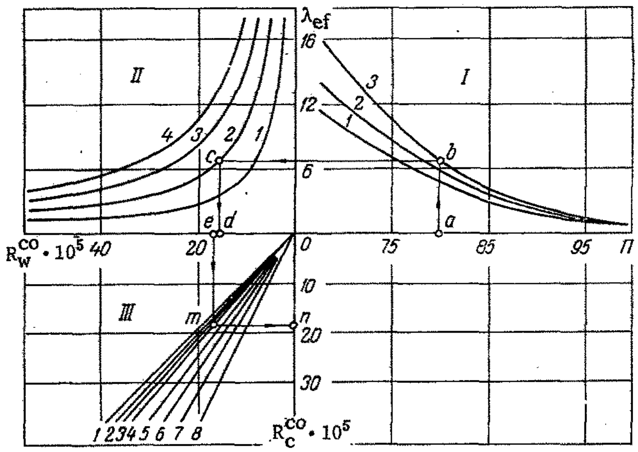

Fig. 2. Nomogram to compute the thermal resistance of the condensation zone. I) Diameter of the discrete fibers $d_f$, $\mu m$: 1) 70; 2) 40; 3) 20. II) Wick thickness $\delta_w$, $\mu m$: 1) 0.5; 2) 1; 3) 1.5; 4) 2. III) Saturation pressure $p_{sat}$, bar: (1) 0.1; 2) 0.09; 3) 0.08, etc., $\lambda_{ef}$, $W/m \cdot deg \cdot K$; $R_w^{co}$, $m^2 \cdot deg\ K/W$; $\Pi$, %.

The results of a computation by the method elucidated were compared with experimental data obtained on 18 heat pipes whose structural characteristics are presented in Tables 1 and 2. The casing and wick material is copper and the heat carrier is water (ethyl alcohol in heat pipes Nos. 14 and 15). Also represented in Tables 1 and 2 are the results of computing the thermal resistance of the condensation and heating zones (for $q_h$ = 5 $W/cm^2$ and $t_{sat}$ = 50°C) and of the heat pipe as a whole (for $F_h/F_c$ = 1 and $\varphi$ = 0°).

In the evaporative mode, which was often accompanied by operation at reduced saturation pressures because of intense cooling [4], the investigated heat pipes are characterized by a high specific thermal resistance whose value is reduced sharply with the passage over to boiling of the heat carrier in the wick, as well as with the increase in saturation pressure (Figs. 1a and b). The influence of the modal parameters is reduced in the domain of developed bubble boiling and $p_{sat} > 0.1$ bar, and the $R_t^{co,ex}$ of heat pipes at $\varphi = 0°$ is determined mainly by the thermal resistance of the condensation zone ($R_c^{co} \gg R_h^{co}$). Substantial lowering of the thermal resistance $R_t^{co}$ can be achieved in this operating mode because of the increase in the condensation zone surface.

In some cases, when the thermal resistance of the heating and condensation zones is commensurate, a reduction in $R_t^{co,ex}$ occurs with the increase in the slope of the heat pipe relative to the horizontal plane (the heat-supply zone is above the heat-elimination zone). The slant of these heat pipes ($\varphi > 0°$) causes partial drainage of the wick and intensifies the heat-transfer process in the heating zone, which results in a noticeable reduction in the thermal resistance of the heat pipe as a whole (HP No. 12 in Fig. 1b).

A histogram of the discrepancies between the computed and test values of the thermal resistivity of the 18 heat pipes was constructed from the results of the comparison for different orientations in the gravity field.

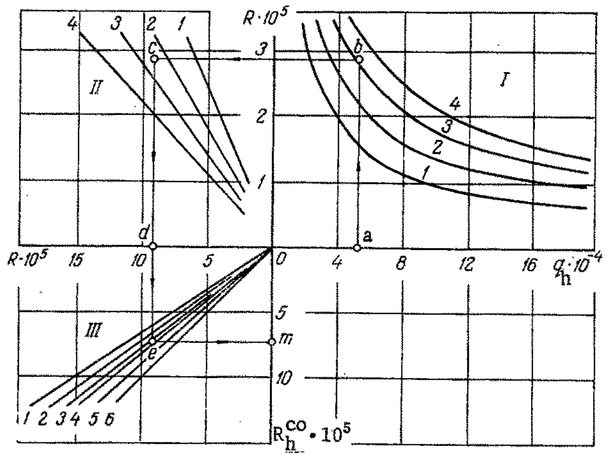

Fig. 3. Nomogram to compute the thermal resistance of the heating zone. I) Saturation temperature $t_{sat}$, °C: 1) 200; 2) 80; 3) 40; 4) 20. II) The parameter $\varepsilon_F$: 1) 10; 2) 20; 3) 50; 4) 100). III) The slope $\varphi$, deg (1) 90; 2) 45; 3) 30; 4) 15; 5) 2.5; 6) 0. $q_h$, W/m².

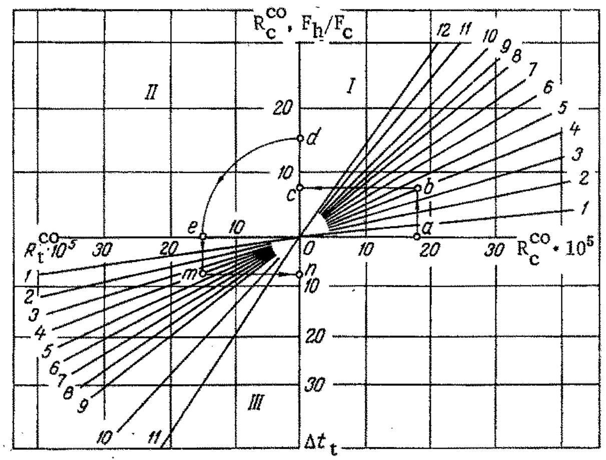

Fig. 4. Nomogram to compute the thermal resistance of heat pipes. I) The ratio $F_h/F_c$: 1) 0.1; 2) 0.2, ... ; 10) 1.0; 11) 1.2; 12) 1.4). III) The heat flux density $q_h$, W/cm²: 1) 2; 2) 3; ... ; 9) 10; 10) 14; 11) 20).

The nature of the histogram is similar to a normal distribution curve and no systematic deviations were observed. The rms discrepancy for 330 points is ±12%.

A method to determine the thermal resistivity of heat pipes by using nomograms (Figs. 2–4) was developed to simplify the computations. We will show the use of the nomograms in computing the thermal resistivity of heat pipes.

Let it be required to determine the thermal resistance of a plane heat pipe of length $L_t$ = 300 mm ($L_h$ = 50 mm, $L_c$ = 150 mm) with inner casing dimensions 15 × 4 mm (wall thickness is $\delta_{wa}$ = 1 mm) which has a copper metal-fiber wick ($d_f$ = 20 μm, $l_f$ = 3 mm) of the thickness 1 mm, porosity Π = 80%, and operating with water as heat carrier for Q = 100 W and $t_{sat}$ = 40°C. The orientation in the gravity field was $\varphi$ = 30°.

We find the thermal resistance of the wick (Fig. 2) along the line *a*bcd for given geometric and structural characteristics of the wick, and the type of heat carrier. We lay off the segment de corresponding to the thermal resistance of the condensate film $R_{fi} = 1.67 \cdot 10^{-5}$ m² · deg K/W as well as the wick contact with the heat pipe casing (in this case $R_{c.t}$ = 0). We draw em to intersect the line $p_{sat}$ = 0.074 bar ($t_{sat}$ = 40°C). The thermal resistance of the condensation zone equals $R_c^{co} = 18 \cdot 10^{-5}$ m² · deg K/W (the segment 0n).

We find the thermal resistance of the heating zone ($\varepsilon_F$ = 37) for $\varphi$ = 0° (the segment 0d) from the known heat flux density ($Q_h = Q/F_h$ = 5.3 W/cm²) in Fig. 3 by the line *a*bcd. We draw de to intersect the line $\varphi$ = 30°. The desired thermal resistance $R_h^{co}$ is $7 \cdot 10^{-5}$ m² · deg K/W (the segment 0m).

We lay off the segment 0*a* equal to $R_c^{co}$ along the abscissa axis in Fig. 4, and we draw *a*b parallel to the ordinate axis to intersect the line $F_h/F_c = 0.42$. We draw bc parallel to the abscissa axis. We lay off the segment cd equal to $R_h^{co}$ on the ordinate axis. We find the heat-pipe thermal resistance, equal to 0d or 0e ($R_h^{co} = 15 \cdot 10^{-5}$ m$^2 \cdot$deg K/W). The temperature drop along the length of the heat pipe $\Delta t_t = 8°C$ can be determined by means of the line emn.

## NOTATION

D, wick pore diameter; d, diameter; F, surface area; $F_w$, wick cross-sectional area; $a$, h, inner width and height of plane heat-pipe casings; L, $l$, length; $\Pi$, porosity; p, pressure; $\Delta p_{ch}$, available capillary head; Q, heat flux transmitted by the heat pipe; q, heat flux density; R, thermal resistivity; r, heat of vapor formation; t, temperature; $\Delta t$, temperature drop; $\delta$, thickness; $\lambda_{ef}$, coefficient of effective wick heat conduction; $\lambda_1$, $\lambda_2$, coefficients of heat conduction of the metal and liquid; $\nu$, coefficient of kinematic viscosity; $\rho$, density; $\sigma$, coefficient of surface tension; $\varphi$, slope of the heat pipe relative to the horizontal plane; $\Omega$, perimeter. Indices: f, fiber; in, inner; fl, fluid; c, condensation zone; h, heating zone; sat, saturation; v, vapor; v.c, vapor channel; fi, film; co, computed; m, mean; t, heat pipe; w, wick; ex, experimental.

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