A NONSTEADY-STATE TWO-DIMENSIONAL MODEL AND ANALYSIS OF NONISOTHERMAL STATE OF HEAT PIPE SURFACES UNDER NONUNIFORM HEAT EXCHANGE ALONG THE PERIMETER AND LENGTH

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Results of theoretical and experimental studies of heat pipes with nonuniform heat exchange along the perimeter and length are presented.

In designing and developing temperature stabilization systems based on heat pipes the problems of calculating and analyzing temperature fields on heat removal surfaces, estimating operating temperatures and efficiency of heat removal, and optimization of the chosen heat pipe construction are characteristic. Solving these problems is important in organizing radiant heat flow, where the heat sources are located on highly conductive heat channels or directly on the heat pipe body [1, 2]. The sources may liberate significant power while having small dimensions, creating discrete heat liberation conditions. The experiments of [3] have shown that discrete heat removal affects the operating and limiting characteristics of a heat pipe for a small number of sources distributed nonuniformly over the body. Significant temperature elevations (up to 30-50°C) in the region of heat sources were found. Choice of construction, method of attachment, and position of the heat pipe exhaust surfaces (for example, radiation radiators [2]) requires an analysis of the temperature state of the pipe over length and perimeter.

The characteristics of heat pipes were studied theoretically in [4, 5] for nonuniform heat removal about the perimeter. A study state two-dimensional heat transport model for the evaporation zone [4] in polar coordinates for nonuniform heat removal was obtained with a number of assumptions which did not allow evaluating the effect of heat exchange intensity, axial thermal conductivity, or heat source location. The model of [5] differs from that one in its nonsteady state formulation of the problem.

To analyze the isothermal state of the evaporation and condensation zones of a heat pipe with consideration of thermal conductivity of the construction elements of their cross section (body wall, capillary structure, radiator, etc.) over length and perimeter, nonuniformity of heat exchange intensity over length, perimeter, and time, we have developed a nonsteady state two-dimensional model for calculation of heat pipe temperature fields and a function for approximate determination of the need for deeper analysis and determination of the degree of temperature variation of the pipe surfaces.

The mathematical model of heat transport is considered with the following major assumptions: the capillary structure is completely saturated with liquid; the vapor is in a saturated state and the vapor temperature is constant along the heat pipe; change in temperature of the wall, saturated capillary structure, and other construction elements occurs along the axis and perimeter of the heat pipe; the thermophysical properties of the heat exchange agent and pipe material are constant; the heat pipe operates below its limits; the body wall, the saturated capillary structure, and other components of the cross section are construction elements of the heat pipe zones through which heat transport by thermal conductivity along the perimeter and in the axial direction are considered.

The initial geometric dimensions and a diagram of heat transport are shown in Fig. 1. We formulate the problem for a general case. The system of differential equations for the evaporation ($i = 1$), transport ($i = 2$), and condensation ($i = 3$) zones obtained from the condition of thermal balance for a cylindrical element $rd\varphi dx$ of the heat pipe zones has the form

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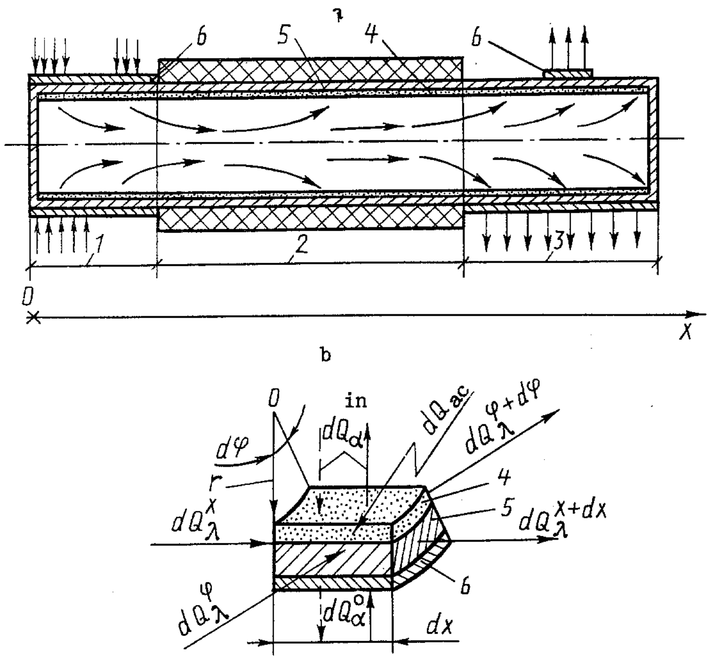

Fig. 1. Diagram of two-dimensional heat transport model in cylindrical heat pipe (a) and thermal balance for a cylindrical element $rd\varphi dx$ (b): 1) evaporation zone; 2) transport zone; 3) condensation zone; 4) saturated capillary structure; 5) heat pipe body; 6) radiator; dQ_α, dQ_λ, thermal flux introduced (removed) by convection (radiation) and thermal conductivity respectively; dQ_{ac}) change in internal energy of material.

$$C_i \frac{\partial T(x, \tau)}{\partial \tau} = A_i \frac{\partial^2 T(x, \tau)}{\partial x^2} + B_i \frac{\partial^2 T(x, \tau)}{\partial \varphi^2} \pm q^{o}_{l,i}(x, \varphi, \tau, T, T_s) - q^{in}_{l,i}(x, \varphi, \tau, T, T_v). \tag{1}$$

with coordinates arranged as follows: x = 0, beginning of evaporation zone, $\varphi = 0$, chosen arbitrarily on periphery of pipe.

System (1) was considered for the following initial conditions on the heat pipe surface:

$$T(x, \varphi, 0) = f(x, \varphi) \tag{2}$$

and boundary conditions on the end faces

$$\left.\frac{\partial T}{\partial x}\right|_{x=0} = 0, \quad \left.\frac{\partial T}{\partial x}\right|_{x=L} = 0. \tag{3}$$

To complete the system we have the integral equation for vapor temperature

$$\sum_{i=1}^{3} \int_0^{x_i} \int_0^{2\pi} \alpha_{l,i} (T - T_v)\, d\varphi dx = 0. \tag{4}$$

Integrodifferential system (1)-(4) was approximated with a conservative finite difference scheme of the Krank–Nicholson type. Nonlinearity in the coefficients of Eqs. (1) and (4) required performance of additional numerical experiments to determine stability conditions for the solution for characteristic parameters and operating regimes of low temperature heat pipes. Thus, for example, for variants of the algorithm constructed with an explicit scheme for finding the solution of system (1)-(4), the value of the dimensionless Fourier parameter defining the stability limit of the solution $Fo_{x,\varphi} \lesssim 0.22$ at thermal flux densities $q \lesssim 2$ W/cm^2 for a copper heat pipe and $Fo_{x,\varphi} \lesssim 0.15$ at $q \lesssim 3$ W/cm^2 for a stainless steel pipe.

We will demonstrate the capabilities of the proposed model by analyzing the temperature state of the evaporation zone of a heat pipe with complex boundary conditions on its surfaces. For the analysis we choose a low temperature heat pipe with characteristic body dimensions,

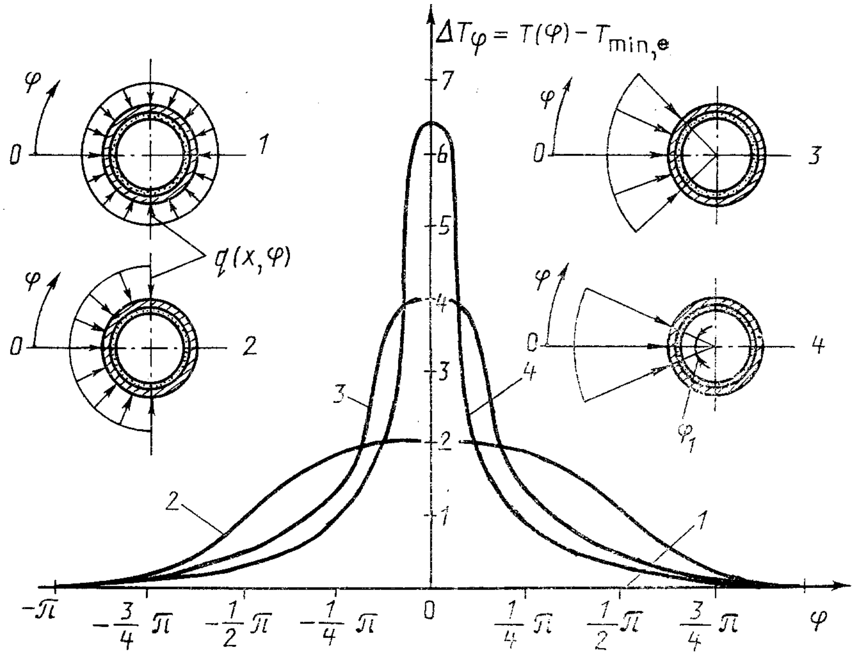

Fig. 2. Change of heating ΔT_φ of body of zone of evaporetion of heat pipe along perimeter for nonuniform heat supply: 1) $\varphi = 2\pi$; 2) π; 3) $\pi/2$; 4) $\pi/4$. ΔT_φ, K; φ, rad.

capillary structure parameters, and heat exchange conditions: evaporation zone length L_e = 0.03-0.1 m, transport zone length L_t = 0.1-0.2 m, condensation zone length L_c = 0.1-0.2, outer diameter d_o = 0.017 and 0.019 m, inner diameter d_{in} = 0.015 m, vapor channel diameter d_v = 0.012-0.014 m, porous structure porosity P = 0.65-0.85; body and porous structure materials: copper and stainless steel; heat exchange regimes on surface of evaporation zone within heat pipe: evaporation and boiling with change in regime along perimeter and length, and change in heat exchange intensity in vapor channel along zones α_e = 1000-18,000 W/(m²·K), $\alpha_t = \alpha_c$ = 1000-3000 W/(m²·K); heat exchange outside the heat pipe in evaporation zone, heat removal with type II boundary conditions, supplied thermal flux density q_e constant or varying along length and perimeter (heat supply angles $\varphi_1 = 2\pi$, π, $\pi/2$, $\pi/4$; value $q_e = (0.5\text{-}100)\cdot 10^4$ W/m²); in condensation zone, heat removal with type III boundary conditions with radiation, intensity of heat exchange with surrounding medium α_0 constant along length and perimeter, α_0 = 5-1000 W/(m²·K). It was found that the following factors had a dominating effect on uniformity of the evaporation zone temperature along perimeter and length: thermal conductivity of body wall, density and angle of heat supply, regime and intensity of heat exchange on inner surface of zone. A typical pattern of change in temperature along the perimeter of the evaporation zone is shown in Fig. 2 for a stainless steel heat pipe. The body thermal conductivity coefficient λ_{st} = 16 W/(m·K). Calculations were performed for a constant evaporative heat exchange regime. Growth in temperature head between the maximum value $T_{max,\ e}$ and the lowest value $T_{min,\ e}$ is observed upon decrease in the quantities λ_{st} and α_e. Decrease in the heat supply angle with unchanged total thermal flux leads to increase in the value $\Delta T_{\varphi,max} = T_{max,\ e} - T_{min,\ e}$. For a constant value of the heat exchange coefficient α_e along the perimeter in the nonsteady state regime the change of $\Delta T_{\varphi,\ max}$ is complex in character, and the highest values are reached in the steady state regime. Analysis of calculation results for α_e = const showed essentially a linear dependence of the function $\Delta T_{\varphi,\ max} = f[q(\varphi)]$ for other geometric and regime parameters constant.

An important advantage of the model developed is the possibility of estimating the degree of influence of heat exchange regimes along the perimeter and temperature profiles in the heat pipe zones. For the heat supply zone it was established that the largest temperature differentials $\Delta T_{\varphi,\ max}$ were achieved in the evaporative regime, and can exceed those values for boiling in the steady state. In the calculations transition from the evaporative operating regime to the boiling regime was determined from the experimentally confirmed dependences of [6], which describe the effect of the body and porous structure material, structural characteristics of the metal filament porous structure and properties of the heat exchange agent.

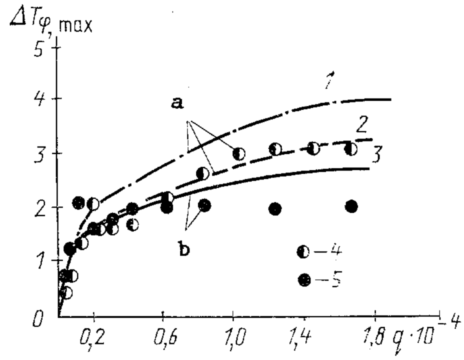

Fig. 3. Maximum heating of heat pipe evaporation zone $\Delta T_{\varphi,\max}$, K vs heat supply density g, W/m²: a) $\varphi = \pi/2$; b) $\varphi = \pi$; 1, calculation with Eq. (5); 2, 3, model of Eqs. (1)-(4); 4, 5, experiment.

The effect of the dominant factors on the highest values of the temperature differential $\Delta T_{\varphi,\max} = T_{\max}(\varphi, x_m) - T_{\min}(\varphi, x_m)$ over the body perimeter for various heat exchange regimes in the heat pipe zones can be evaluated approximately from the expression

$$\Delta T_{\varphi,\max} = \frac{\pm q \frac{\alpha_2}{\alpha_1} d_o d_m (2\pi - \varphi_1)}{\frac{4}{\pi} \lambda_{st} (d_o - d_{in}) \left(1 + \frac{\alpha_2}{\alpha_1} \frac{2\pi - \varphi_1}{\varphi_1}\right) + \alpha_2 (2\pi - \varphi_1) d_{in} d_m}, \quad (5)$$

$$0 < \varphi \leqslant 2\pi,$$

where α_1 is the heat exchange coefficient inside the heat pipe on the surface $0 \leqslant \varphi \leqslant \varphi_1$, $0 \leqslant x \leqslant L_e (L_e + L_t \leqslant x \leqslant L)$; α_2 is the heat exchange coefficient within the heat pipe on the surface $\varphi_1 < \varphi < 2\pi$, $0 \leqslant x \leqslant L_e (L_e + L_t \leqslant x \leqslant L)$. The values of $\Delta T_{\varphi,\max}$ calculated with Eq. (5), for example, for the evaporation zone of an experimental heat pipe agree satisfactorily with the results of calculations with the model of Eqs. (1)-(4) and experimental data (Fig. 3).

When necessary a more detailed analysis and determination of the degree of temperature uniformity of heat pipe zones with complex boundary conditions can be performed with the proposed model. Thus, Fig. 4 shows a typical nonsteady state variant of calculation of the two-dimensional temperature field of a heat pipe evaporation zone with two heat sources of differing power located nonuniformly along its length directly on the surface. The sources produce discrete heat supply conditions (contact area 10 × 10 mm, $q_{e,1} = 10.6 \cdot 10^3$ W/m², $q_{e,2} = 15.9 \cdot 10^3$ W/m²). For a stainless steel body and the chosen variant of source arrangement along the length the temperature fields do not interact with each other. Calculations were performed with consideration of the development and existence of a heat exchange coefficient $\alpha_e(x, \varphi)$ nonuniform over the perimeter.

Results of calculating the nonisothermal state of the condensation zone with system (1)-(4) for nonuniform heat removal (radiation from one side of a planar radiator) showed the need for choosing construction and location of the heat rejection surfaces and performing a more accurate calculation of the nominal temperature level of heat pipe operation or refining the thickness and form of the body, porous structure, and radiator material.

To confirm the theoretical principles developed and expand available information on nonsteady state regimes of heat pipe operation experimental studies were performed with a heat pipe of simple construction with nonuniform heat supply about the perimeter. Characteristics of the experimental heat pipe were: zone lengths $L_e = 0.14$ m, $L_t = 0.107$ m, $L_c = 0.14$ m, body outer and inner diameters $d_o = 24 \cdot 10^{-3}$ m and $d_{in} = 22 \cdot 10^{-3}$ m, body and porous structure material, stainless steel (1Kh18N9T), with metal filament porous structure (P = 0.81), porous structure thickness $\delta = 0.95 \cdot 10^{-3}$ m, heat exchange agent, methyl alcohol. Heat supply was uniform over length and variable about the perimeter: $\varphi_1 = \pi$, $\pi/2$, $\pi/4$; $q = (0.1\text{-}16.5 \cdot 10^3)$ W/m². The evaporation zone surface outside the heat supply zones was thermally insulated. Thermal flux extraction in the condensation zone was uniform over length and perimeter. The experiments were performed with the apparatus of [7], in which a type K-200 data measurement system

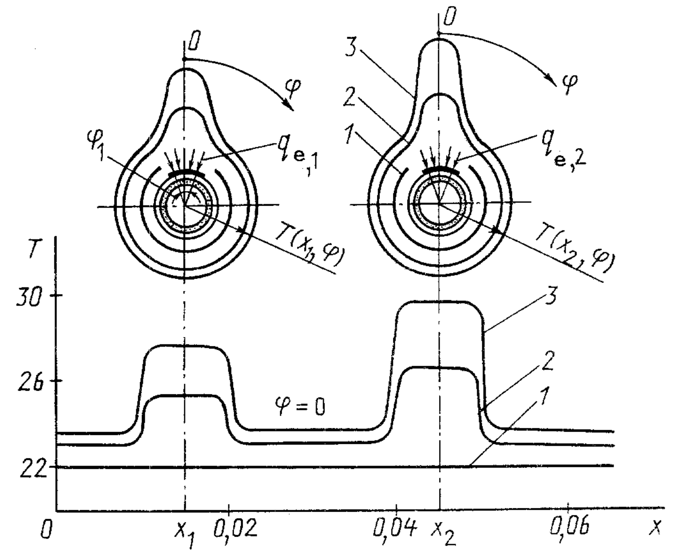

Fig. 4. Nonsteady state two-dimensional temperature field of heat pipe evaporation zone for nonuniform location of heat sources: $q_{e,1} = 10.6 \cdot 10^3$ W/m²; $q_{e,2} = 15.9 \cdot 10^3$ W/m²; 1) $\tau = 0$; 2) 60; 3) 500 sec. x, m; T, °C; φ, deg.

is used for recording the temperature in nonsteady-state regimes. Copper—constantin thermocouples calked into grooves in the heat pipe body were used as temperature sensors.

Figure 3 shows experimental results of determining maximum temperature differential over perimeter in the evaporation zone as a function of thermal flux density $\Delta T_{\varphi,\max} = f[q_e(\varphi)]$ for heat supply angles $\varphi_1 = \pi$ and $\pi/2$. On the curve $\Delta T_{\varphi,\max} = f[q_e(\varphi)]$ one can distinguish three characteristic regions in which various heat exchange regimes exist about the perimeter inside the heat pipe. At low thermal flux densities the most probable situation is existence of only an evaporative regime about the perimeter. When a certain density q_e is reached an intense vapor formation process commences in the porous structure [6]. From that time existence of both an evaporative and a boiling regime is possible. In the range $q_e \gtrsim 10^4$ W/m² (Fig. 3) only the boiling regime is characteristic of the given construction.

It has been established that for a cylindrical body construction with decrease in the heat supply angle (basically, for realization of discrete heat supply conditions) crisis-free operation of the heat pipe is possible at significantly higher thermal flux densities, which would prove to be limiting in the case of uniform heating about the entire perimeter. The experimental data confirm the proposition that the largest values of $\Delta T_{\varphi,\max}$ are possible under nonsteady conditions for densities q_e = const. Upon startup of the experimental heat pipe for the range $600 \lesssim q \lesssim 0.2 \cdot 10^4$ W/m² and $\varphi_1 = \pi/2$ the existence of an evaporative regime in the initial period produced a temperature head $\Delta T_{\varphi,\max}$ larger than in the steady state where a boiling process occurred stably.

NOTATION

C_i, A_i, B_i, reduced coefficients considering heat capacity and thermal conductivity along axis and perimeter of construction elements of heat pipe zone i (i = 1, evaporation zone; i = 2, transport; i = 3, condensation); $q^{o}_{\ell,i}(x, \varphi, \tau, T, T_s)$, linear thermal flux supplied to (removed from) outer surface in zone i; $q^{in}_{\ell,i}(x, \varphi, \tau, T, T_s) = \alpha_i[(x, \varphi, \tau) - T_v]$, linear thermal flux supplied to (removed from) inner surface in zone i; T, T_v, T_s, temperatures of heat pipe body, vapor, and surrounding medium; τ, time; r, radius; α_i, heat exchange coefficient in vapor channel of zone i; φ_1, heat supply (removal) angle; L, heat pipe length; q, thermal flux density; d, diameter; $\Delta T_{\varphi,\max}$, maximum temperature head over perimeter. Subscripts: e, t, c, evaporation, transport, and condensation zones; in, inner; o, outer; m, mean value; s, surrounding medium; v, vapor; ℓ, values referenced to unit length.

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DETERMINATION OF THE COEFFICIENT OF HEAT TRANSFER AT THE INNER SURFACE OF A TWO-PHASE HEAT EXCHANGER

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Heat transfer at the inner surface of a two-phase system of thermal regulation is investigated. A method of conjugate gradients for an inverse problem of nonsteady heat conduction is used. The time dependence of the heat transfer coefficient is found for the startup regime of the thermal regulation system, and the calculation accuracy is estimated.

The creation of new engineering models requires extensive experimental research with processing of test data, including that on thermal regimes. Heat exchangers based on a closed evaporation—condensation cycle are now being created, and one of the main problems, even in the preliminary design stage, is the determination of heat-transfer coefficients.

Let us consider the hydraulic loop of a thermal regulation system (Fig. 1) designed to stabilize the temperature of radio apparatus. The heat-releasing elements 6 are located on a heat plate 1 made of PK 01309 aluminum extrusion with a capillary structure in the form of rectangular grooves on the inner surface. The grooves 7 and the condensate pipe 5 are filled with acetone. The heat exchanger works on the heat-pipe principle. The heat released in the operation of instruments goes into heating and evaporating the coolant, the vapor goes through the pipe 3 into the condenser 4 where it condenses, and the liquid goes through the main 5 into the capillary structure of the heat plate 1. Heat and mass transfer occur due to capillary pressure generated at the phase interface of the capillary structure.

The high efficiency of the heat exchanger is achieved by heat transfer of the latent heat of vaporization of the coolant. The absence of moving mechanical parts, a power supply, or a system of automatic regulation makes evaporation—condensation devices more reliable than traditional devices and improves the weight characteristics.

The experimental determination of the heat-transfer coefficient in the evaporation zone is complicated by the fact that the height of a vapor channel is fairly small ($b = 0.004$ m). The methods of inverse problems of nonsteady heat conduction [1] therefore seem the most suitable for finding α. Here the boundary conditions are specified from experience.

The problem consisted in an investigation of the start-up regime of the thermal regulation system. The heat-releasing elements were simulated by pumping hot water through an auxiliary heat exchanger (not shown in Fig. 1). In the course of the experiment, we measured the applied heat flux q, the temperature T_W of the outer wall, and the vapor temperature T_V in the inner cavity (Fig. 2).

The amount of heat was determined by calorimetry and the temperatures were measured with Chromel—Copel thermocouples (wire diameter 0.003 m) and KSP-4 potentiometers.

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