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ON THE INHOMOGENEITY OF THE TEMPERATURE FIELDS IN THE CROSS SECTION OF THERMAL TUBES

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The problem on determination of the temperature field in the cross section of a thermal tube intersecting the zone of heat transfer to the outer surface of the tube has been solved with regard for the inhomogeneity of the boundary conditions on the outer surface of this tube and the materials from which the body and wick of the tube are made. It has been established that the temperature difference along the angular coordinate of nonmetal tubes with a very inhomogeneous heat transfer on their outer surface does not exceed 0.5 K, while this temperature difference in nonmetal tubes is more significant under adequate conditions.

Introduction. The work of thermal tubes (TT) used for stabilizing the heat conditions of power-consuming radioelectronic devices [1] can be analyzed using different models [2] defining, in more or less detail, the thermophysical processes occurring in these tubes. Of practical importance is the three-dimensional heat and mass transfer in a small neighborhood of such thermal tubes. The characteristic cross-section sizes of the thermal tubes (several millimeters) used in the majority of temperature-control systems are much smaller than the sizes of units of radioelectronic devices (tens of centimeters). However, the processes occurring in thermal tubes provide the transfer of the excess energy from the zones of relatively high temperatures to the zones of relatively low temperatures, and the heat transfer in a small neighborhood of the thermal tubes can be three-dimensional in many cases. Up to now, there has been no work done to solve the three-dimensional problem on the heat transfer in a thermal tube in a complete formulation, and a pressing problem is taking into account the heat transfer in the body and wick of this tube in the peripheral direction. On the one hand, the majority of materials from which the body and wick of thermal tubes are made [2, 3] have a high heat conductivity; therefore, the inhomogeneity of the temperature field along the angular coordinate of a thermal tube should be relatively small if it is made from, e.g., aluminum or its melts. On the other hand, however, the operation of thermal tubes used in temperature-control systems of radioelectronic devices [4] and spacecraft, e.g., communication satellites [5], depends on the specific heat transfer from the energy sources to the evaporation zone of a coolant. It is difficult to provide a heat flow that would be homogeneous throughout the evaporation surface.

At present, the advisability of using thermal tubes whose body or capillary system are made from composite materials, such as ceramics, is also a question [6]. These materials have, along with obvious disadvantages (e.g., a relatively low heat conductivity) significant advantages (a low density and a good wettability). Because of this, to make an impartial assessment of the efficiency of thermal tubes whose body and wick are made of composite materials, it is necessary to analyze the temperature fields in the cross sections of these tubes with account for the heat transfer along their peripheral coordinate under typical operating conditions. Moreover, the miniaturization of both power-consuming radioelectronic devices and, accordingly, the thermal tubes used for their cooling significantly complicates the schemes of heat transfer to the surface of these tubes. Therefore, it is important to analyze the features of heat transfer in thermal tubes operating under inhomogeneous conditions of delivery of heat to their outer surface.

It is practically impossible to experimentally determine the temperature field in the cross section of thermal tubes because of the small [7] temperature gradients along all of their coordinate directions, the small sizes of these tubes, and the errors introduced by thermocouples built into their body. Therefore, numerical modeling is most probably the main tool for solving the indicated problem.

The aim of the present work is to numerically simulate the temperature fields in the cross section of a low-temperature thermal tube intersecting the zone of energy transfer from a heat-release source and, consequently, the

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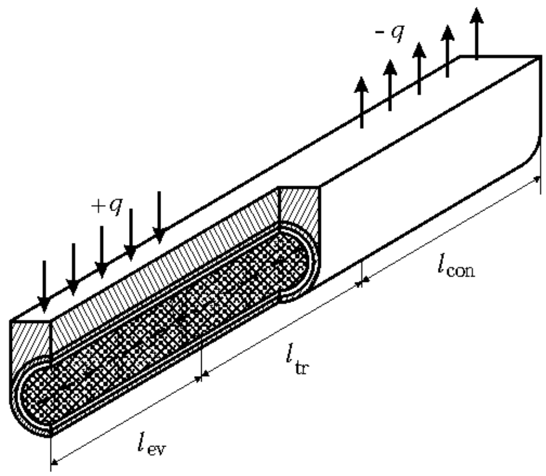

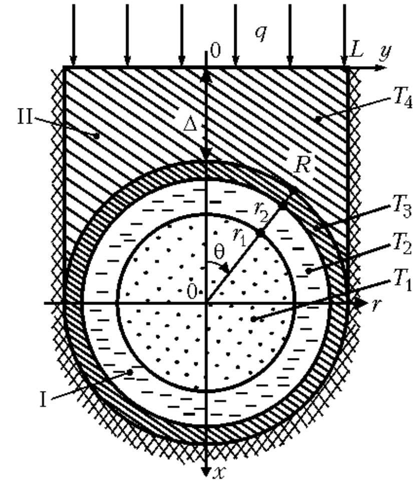

Fig. 1. Diagram of the connecting element–thermal tube system.

Fig. 2. Simplified scheme of the computational region of the problem: I) thermal tube; II) connecting element.

evaporation zone of a coolant, by the example of thermal tubes made from different materials having very different thermophysical parameters.

Formulation of the Problem. The most general problem on heat and mass transfer in a thermal tube, formulated as early as 1974 [8], has not been solved numerically up till now. The axially symmetric model [9], accounting for the main mechanisms of heat transfer [2] and many factors influencing the operating conditions of thermal tubes, takes no account of the energy transfer along their peripheral coordinate. The extension of the approach proposed in [9] to the three-dimensional formulation of the problem considered makes the calculation procedure very complex. Therefore, we solved the indicated problem with the use of a nonstationary quasi-three-dimensional model of energy transfer in a thermal tube, which accounts for the heat transfer not only along the peripheral and radial coordinates but also (approximately) along the longitudinal coordinate. We considered the heat transfer in a system, the schematic diagram of which is presented in Fig. 1.

A thermal tube (I) was simulated by a three-layer cylinder consisting of a body, a wick, and a vapor channel, which was coupled with a connecting element (II) through an ideal thermal contact (Fig. 2). The problem was solved with account for the heat transfer, arising due to the heat conduction in the body, wick, and vapor, and the thermal evaporation of a coolant. At the boundaries of a computational region, the conditions of heat-flow inhomogeneity were set. To take into account the energy transfer along the coordinate z, we introduced corresponding source terms into the energy equations for the body, wick, and vapor channel. The problem was formulated with the following main assumptions:

1) the thermophysical parameters in the region of the thermal tube are calculated as effective parameters with account for the volume fraction of each component;

2) the flow rate of a liquid coolant flowing to the evaporation zone is equal to the flow rate of the vaporous coolant in this zone (the mass-transfer process is stationary) and the coordinates of the evaporation boundary do not change;

3) the thermophysical parameters (heat conduction, heat capacity) are temperature-independent;

4) the whole outer surface of the computational region is heat-insulated, except for the region $x = 0$, $0 \le y \le L$ (region of heat delivery);

5) the contacts at the interphase boundaries as well as at the boundaries between the thermal-tube body and a condensate and between the tube and the connecting element are ideal;

6) the temperature difference ΔT along the length of the thermal tube is not larger than 3 K. Numerous experimental investigations [4] have shown that this value of ΔT is typical for well-operating tubes;

7) the processes of heat release in the condensation zone have no direct influence on the temperature field formed in the evaporation zone.

As was already noted, the temperature drop in actual thermal tubes operating in the steady-state regimes comprises 3–4 K; this temperature drop is due to the interrelated processes of heat and mass transfer occurring simultaneously in a thermal tube and in its neighborhood.

The energy transfer in a low-temperature thermal tube is defined within the framework of the model proposed by the following system of nonstationary differential equations with corresponding boundary conditions:

$$\frac{\partial T_1}{\partial \tau} = a_1 \left(\frac{\partial^2 T_1}{\partial r^2} + \frac{1}{r} \frac{\partial T_1}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_1}{\partial \theta^2} \right) - \varphi_1 \frac{U \Delta T}{l} , \tag{1}$$

$$\frac{\partial T_2}{\partial \tau} = a_2 \left(\frac{\partial^2 T_2}{\partial r^2} + \frac{1}{r} \frac{\partial T_2}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_2}{\partial \theta^2} \right) - \varphi_2 \frac{\lambda_2 \Delta T}{C_2 \rho_2 l^2} , \tag{2}$$

$$\frac{\partial T_3}{\partial \tau} = a_3 \left(\frac{\partial^2 T_3}{\partial r^2} + \frac{1}{r} \frac{\partial T_3}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T_3}{\partial \theta^2} \right) - \varphi_3 \frac{\lambda_3 \Delta T}{C_3 \rho_3 l^2} , \tag{3}$$

$$C_3 \rho_3 \frac{\partial T_4}{\partial \tau} = \lambda_3 \left(\frac{\partial^2 T_4}{\partial x^2} + \frac{\partial^2 T_4}{\partial y^2} \right) . \tag{4}$$

The initial conditions have the following form:

$$\tau = 0 , \quad T_1 = T_0 , \quad T_2 = T_0 , \quad T_3 = T_0 , \quad T_4 = 0 . \tag{5}$$

The boundary conditions are as follows:

$$r = 0 : \quad \frac{\partial T_1}{\partial r} = 0 , \tag{6}$$

$$r = r_1 , \quad 0 \le \theta \le \pi : \quad - \lambda_1 \frac{\partial T_1}{\partial r} = - \lambda_2 \frac{\partial T_2}{\partial r} - QW , \quad T_1 = T_2 , \tag{7}$$

$$r = r_2 , \quad 0 \le \theta \le \pi : \quad - \lambda_2 \frac{\partial T_2}{\partial r} = - \lambda_3 \frac{\partial T_3}{\partial r} , \quad T_2 = T_3 , \tag{8}$$

$$r = R , \quad 0 \le \theta \le \pi/2 : \quad - \lambda_3 \frac{\partial T_3}{\partial r} = - \lambda_4 \frac{\partial T_4}{\partial r} , \quad T_3 = T_4 , \tag{9}$$

$$r = R , \quad \pi/2 \le \theta \le \pi : \quad \frac{\partial T_3}{\partial r} = 0 , \tag{10}$$

$$x = 0 , \quad 0 \le y \le L : \quad - \lambda_4 \frac{\partial T_4}{\partial x} = q , \tag{11}$$

$$y = 0 , \quad 0 \le x \le \Delta : \quad \frac{\partial T_4}{\partial y} = 0 , \tag{12}$$

$$y = L, \quad \Delta \le x \le \Delta + R: \quad \frac{\partial T_4}{\partial y} = 0, \tag{13}$$

$$\theta = 0, \quad 0 \le r \le r_1: \quad \frac{\partial T_1}{\partial \theta} = 0, \tag{14}$$

$$\theta = 0, \quad r_1 \le r \le r_2: \quad \frac{\partial T_2}{\partial \theta} = 0, \tag{15}$$

$$\theta = 0, \quad r_2 \le r \le R: \quad \frac{\partial T_3}{\partial \theta} = 0, \tag{16}$$

$$\theta = \pi, \quad 0 \le r \le r_1: \quad \frac{\partial T_1}{\partial \theta} = 0, \tag{17}$$

$$\theta = \pi, \quad r_1 \le r \le r_2: \quad \frac{\partial T_2}{\partial \theta} = 0, \tag{18}$$

$$\theta = \pi, \quad r_2 \le r \le R: \quad \frac{\partial T_3}{\partial \theta} = 0. \tag{19}$$

The mass rate of evaporation of a coolant is calculated by the formula

$$W = \frac{A\,(P^{\mathrm{sat}} - P)}{\sqrt{2\pi R_0 T_{\mathrm{int}}/M}}. \tag{20}$$

Method of Solution. The problem considered was solved by the finite-difference method with the use of partial differential equations represented in the form of two-dimensional difference equations [10]. A change to a new time layer was performed in two "fractional steps" by the splitting scheme [10]. The heat transfer along the coordinates x and y in the connecting element was calculated in the first and second fractional steps, respectively, with the use of one-dimensional difference equations. The heat transfer in the thermal tube was calculated along the coordinates r and θ.

The system of one-dimensional difference equations was solved using the sweep method by the implicit four-point difference scheme [10]. The pressure of the saturated vapor was determined by the Riedel–Planck–Miller method [11].

The heat transfer in the connecting element was calculated using a rectangular difference grid. The heat transfer in the thermal tube (in the three-layer cylinder) was calculated using a spherical difference grid. The rectangular and spherical grids were conjugated at the interfaces between regions. A peculiarity of the problem is the existence of locally concentrated, highly active sites of heat absorption in a small-thickness region in the coolant-evaporation zone. The evaporation process was physically simulated only at one point, lying on the coordinate r of the difference scheme, at which the evaporation conditions are realized. Therefore, in deciding on grid parameters, prominence was given to the control of the conditions under which the iteration process converges. The steps were selected along the time and space coordinates such that the conditions of iteration-process convergence were fulfilled.

Numerical calculations were carried out for the thermophysical parameters of the body and connecting element of thermal tubes made from an aluminum alloy, a steel, and a composite and for the working medium NH_3 (in the liquid and vapor phases) [11, 12]. The quantity U was calculated by the mass rate of the vapor.

TABLE 1. Thermophysical Parameters of Thermal Tubes

TT elements	Material	λ	C	ρ
Body	Aluminum alloy	120	500	2700
Wick net	»	60.251	2484	1691.5
Heat-transfer agent	Ammonia	0.022	2160	0.8
Body	Steel 45	32	561	7794
Wick net	»	16.251	2514.5	4238.5
Heat-transfer agent	Ammonia	0.022	2160	0.8
Body	Composite	1.5	1700	1850
Wick net	»	1.001	3084	1266.5
Heat-transfer agent	Ammonia	0.022	2160	0.8

Note: For the wick, the effective heat conductivity, heat capacity, and density are given.

Results and Discussion. Numerical investigations were carried out for thermal tubes with bodies and wicks of an aluminum alloy, a steel, and a composite. In all cases, ammonia served as the coolant, and the specific heat transferred to the surface of a tube was equal to q = 10 W/m^2. The values of the main parameters used in the investigations are presented in the Table 1.

Since we did not have literature data on the experimentally determined temperature fields in the bodies and wicks of thermal tubes, it was difficult to directly substantiate the results of our numerical investigations of these fields. The local temperatures determined, e.g., in [7], under the conditions of large temperature differences along the axial coordinate of a thermal tube cannot be used for comparison with the corresponding calculation data because of the absence of a complete set of initial data necessary for identification. Therefore, the reliability of the results obtained was verified by internal testing. Grid parameters were selected after a steady-state regime of operation of a thermal tube was established and the temperature field became independent of the steps along the space and time coordinates. The criterion of the convergence of iterations was a 0.1% discrepancy between the temperature fields.

Figure 3 shows the temperature fields in the typical cross section of an aluminum tube, namely, in the evaporation zone, after the establishment of a steady-state regime of operation of this tube. It is well seen that the temperature differences throughout the perimeter of the cross section considered do not exceed 0.5 K despite the fact that the heat transfer conditions on the outer surface of this tube are very inhomogeneous. Analogous temperature fields are presented in Fig. 3. for a steel tube. In the latter case, the maximum temperature difference in the cross section with r = const is equal to 2.3 K. This difference cannot be also considered as large because the rates of coolant evaporation at minimum and maximum temperatures of the evaporation surface differ only by 9–10%. This difference can be considered as a small deviation and can be disregarded when the parameters and conditions of heat transfer from heat-release sources to the thermal tube are determined.

For a thermal tube with a body and wick made from a composite, we obtained maximum temperature differences along its peripheral coordinate as compared to those of the other tubes considered (Fig. 3). However, even in this case, the absolute temperatures at the outer boundary of the connecting element differed from those of a thermal tube with a body and a wick made from an aluminum alloy by a value not larger than 25.9 K at identical geometric characteristics and heat-transfer conditions. The results obtained allow the conclusion that an inhomogeneity of the boundary conditions at the outer surface of a thermal tube does not influence the temperature distribution along its peripheral coordinate in the case where the body and wick of this tube are made from a material having a high heat conductivity. If the body and wick of a thermal tube are made from a material having a low heat conductivity, the body will be heated inhomogeneously and a temperature difference will arise along the angular coordinate; the value of this difference will increase with decrease in the heat conductivity. However, in this case, the temperature at the boundary between a radioelectronic device and the connecting element, characterizing the operating conditions of this device, changes fairly moderately. In passing from an entirely aluminum thermal tube to an entirely composite thermal

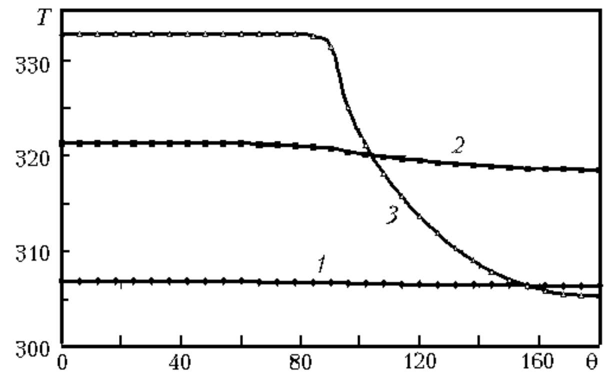

Fig. 3. Temperature distribution along the angular coordinate in the evaporation zone at $r = R$: 1) aluminum; 2) steel; 3) composite. T, K; θ, deg.

tube, the temperature at the boundary between the connecting element and a heat-release source increases from 306.9 to 332.8 K under the typical operating conditions considered.

Numerical analysis of the energy transfer along the body of a thermal tube to its wick caused by the heat conduction in the axial direction, has shown that the contribution of these elements of thermal tubes made from an aluminum alloy, a steel, or a composite to their heat-transfer ability is not larger than 3–4% in the case where the operating parameters of the tubes change in a fairly large range. Consequently, the heat transfer in the body and wick of a thermal tube can be ignored in the process of simulation of its operation. It should be noted that a connecting element of a metal or an alloy, having a high heat conductivity (e.g., aluminum), can play a more significant role in the process of heat transfer. However, in this case, the mass of a temperature-control system, the basic elements of which are thermal tubes, will be significantly increased, which prevents the miniaturization of radioelectronic devices. In systems that are not strictly limited in mass and specific consumption of materials (e.g., high-temperature thermal tubes in temperature-control systems of nuclear power plants) the above-considered design of thermal tubes can be reasonable and appears worthy of further investigation.

The data obtained allow one to determine special features of simulation of the heat transfer in thermal tubes. Comparison of the results of the present work and the results obtained in [9] shows that mathematical simulation of inhomogeneous temperature fields in the connecting-element–tube body–wick system gives fairly exact data on these field. It is very difficult to experimentally determine the indicated fields because the characteristic sizes of the seal of typical thermocouples (several tenths of a millimeter) are comparable with the cross-section sizes of the wick and the body of thermal tubes (several millimeters).

It should be noted that a small number of available experimental data on the local temperatures at certain points on the outer surface of thermal tubes were obtained with systematic and random errors. For example, the temperature of the surface of a thermal tube is usually measured under the conditions of a fairly intensive heat exchange with the environment, whose parameters (e.g., heat-transfer coefficient) are not controlled exactly or are measured with large errors. Measurement of the temperatures of thermal tubes in experimental measuring systems is also accompanied by a number of unavoidable processes leading, according to the data of [13], to large errors in determining the main parameters of these tubes. Therefore, numerical simulation of the temperature fields in thermal tubes with the use of known basic mathematical models [2, 8] and their modifications can be considered as an effective tool for estimating the operating parameters of thermal tubes in various temperature-control systems. Integral experimental characteristics (local temperatures of the outer surface of a thermal tube) are necessary for estimating the reliability of the corresponding theoretical data.

Conclusions. The results of our investigations allow the conclusion that, when the body and wick of a thermal tube are made from a material with a high heat conductivity, an inhomogeneity of the boundary conditions on the outer surface of this tube has practically no influence on the temperature field in the tube cross section intersecting the evaporation zone. In the case where the indicated elements of a thermal tube are made from a material having a low heat conductivity, the temperature in the cross section of the tube depends on θ.

NOTATION

A, accommodation coefficient; a, thermal diffusivity, m^2/sec; C, heat capacity, J/(kg·K); L, size of the heat-delivery zone, m; l, length of a tube, m; M, molecular weight, kg/mole; $\bar{n}$, normal to the outer surface of a thermal tube; P, pressure, Pa; Q, phase-transition heat, J/kg; $+q$ and $-q$, specific heat flows delivered and removed, W/m^2; R, outer radius of a thermal tube, m; r, radius, m; R_0, universal gas constant, J/(mole·K); T, temperature, K; U, longitudinal velocity component of a vapor flow propagating along the coordinate z, m/sec; W, mass rate of evaporation, kg/(m^2·sec); x, y, coordinates, m; Δ, distance from a heat source to the upper point on the symmetry axis of the body of a thermal tube, m; ΔT, temperature difference along the longitudinal coordinate, K; θ, angular coordinate, rad; λ, heat-conductivity coefficients, W/(m·K); ρ, density, kg/m^3; τ, time, sec; φ, fraction of the cross section of an element of a thermal tube. Subscripts: 0, initial; 1, vapor phase; 2, wick zone; 3, body of a thermal tube; 4, connecting element; int, interphase; ev, evaporation zone; con, condensation zone; sat, saturated; tr, transfer zone.

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