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CHARACTERISTICS OF DIODE-TYPE HEAT PIPE IN FORWARD MODE AND IN REVERSE MODE OF OPERATION

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Results of an experimental and theoretical study are reported pertaining to a diode-type heat pipe with a liquid plug of the vapor channel in the reverse mode of operation.

Diode-type heat pipes are used in thermal stabilization systems [1-3]. Such heat pipes operating in the forward mode dump heat efficiently into the ambient medium and, for all practical purposes, insulate the object thermally whenever the ambient temperature exceeds its permissible level.

Several experimental studies have dealt with the laws governing the operation of basic thermodiode circuits [1-4]. Theoretical studies have been limited to consideration of a simplified model of transient heat propagation through a diode-type heat pipe with a liquid trap [5], a model which disregards conductive heat transfer in the axial direction. According to some studies [1, 3, 4], diode-type heat pipes designed without moving parts should be most widely used. These include diode-type heat pipes with a coolant trap [4-6] and with a liquid plug in the vapor channel in the reverse mode of operation [1-3].

Selection and application of any particular design version of diode-type heat pipe are largely determined by the transient characteristics. Transient conditions, however, have so far hardly been studied at all. Only the authors of study [6] have discovered that the transient characteristics of a diode-type heat pipe with a coolant trap depend on the magnitude of the initial heat load and on the instant this load has been applied.

These authors made a study of diode-type heat pipes with liquid plugging of the vapor channel under steady and transient conditions in the forward mode and in the reverse mode of operation.

The terminology of zones used in description of the forward mode of operation (thermal resistance R_{hp} minimum) is also used in description of the reverse mode of operation (thermal resistance R_{rev} approaching its maximum) (Fig. 4).

For the experimental part of the study, earlier equipment [7], adapted for tracking transients in a diode-type heat pipe, namely capable of producing a pulse change of ambient temperature and of heat transfer intensity within the heat transfer zone and at the reservoir surface was used.

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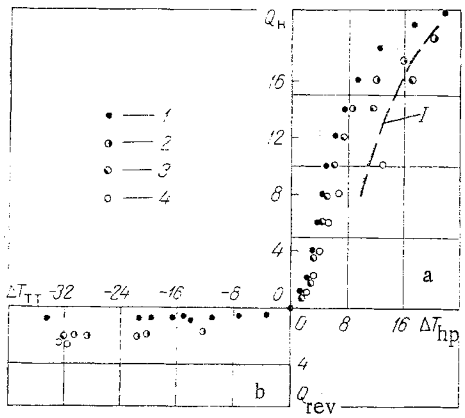

Fig. 1. Generalized steady-state characteristic $Q = f(\Delta T_{hp})$ of diode-type heat pipe in forward mode of operation (a) and in reverse mode of operation (b), at T_0 = 293°K: 1) $\varphi = 0$; 2) $\varphi = 2.5°$; 3) $\varphi = 5°$; 4) $\varphi = 10°$; $\Delta T_{hp} = \bar{T}_{w,i} - T_{w,e}$ (°K); Q_i, Q_{rev} (W).

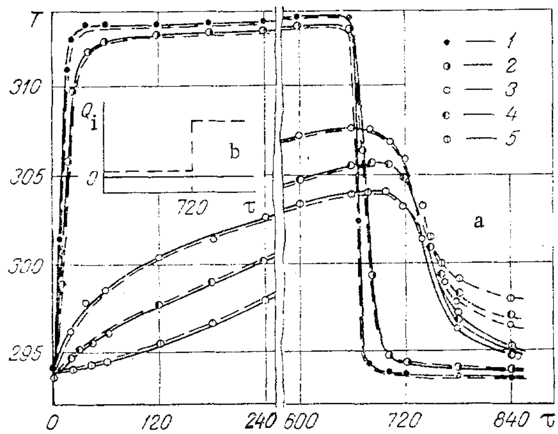

Fig. 2. Change of wall temperature in diode-type heat pipe during transitions from forward mode to reverse mode and from reverse mode to forward mode of operation (a), and change in thermal flux Q_i (W) (b): 1) $\bar{T}_0$, 2) $\bar{T}_{res}$, 3) T_{tr}, 4) $T_{i,tr}$, 5) $\bar{T}_i$ (°K); solid lines refer to pipe in horizontal position with $Q_i = 0$; dashed lines refer to pipe in horizontal position with Q_i = 5 W; time τ (sec).

The temperature field in a diode-type heat pipe was measured with copper—Constantan thermocouples (five in the heat influx zone, six in the transport zone, five in the heat efflux zone, one on the transition segment, and three on the reservoir).

The vapor temperature was recorded with three thermocouples inside thin sheaths made of stainless steel, at distances of 0.25, 0.1, and 0.03 m, respectively, away from the reservoir wall.

In the forward mode of operation a thermal flux Q_F was produced by a resistance heater and carried away by forced convection with a liquid simultaneously streamlining the heat efflux zone, the transition segment, and the reservoir. The operation of a diode-type heat pipe in the reverse mode was studied under transient and steady-state conditions. Transition from forward mode to reverse mode was effected through a pulse change of the ambient temperature around the heat efflux zone and the reservoir ($T_{01} \rightarrow T_{02}$) with power Q_i turned off ($Q_i = 0$). Heat efflux from the heat pipe was effected by minimizing the insulation and using a protective heater. Transition from reverse mode to forward mode was effected also through a pulse change of the coolant temperature, this time from T_{o2} to T_{o1}, with $Q_i \geqslant 0$. The temperatures were $T_{o1} = 293 \pm 0.5°K$ and $T_{o2} = 313 \pm 0.5$, 333 ± 1, and $363 \pm 1°K$.

Steady-state conditions in the reverse mode of thermodiode operation were attained as follows. At the beginning of the experiment a diode-type heat pipe was soaked for about 30 min at the ambient temperature $T_{o1} \pm 0.5°C$ with the aid of thermal insulation and a protective heater. At the instant cooling action (from T_{o2}) began, we also turned on the protective heater for the adiabatic zone. Heat losses in the steady state constituted less than 3% of the thermal flux Q_{rev} transferred in the reverse direction. The thermal flux Q_{rev} entering the heat influx zone was carried away by circulation heat exchangers installed in this zone for study of the steady-state reverse mode of operation.

The results pertaining to the forward mode of operation of diode-type heat pipes with liquid plugs indicate that in this mode the performance of such heat pipes is analogous to the performance of plain heat pipes [8, 9] (Fig. 1a).

Curve 1 in Fig. 1a depicts the limiting heat transfer capacity Q_{max}. The values of Q_{max} and R_{hp} obtained here agree with those according to calculations [8]. The kind of coolant selected, the operating temperature level, and the dimensions of the vapor channel are such that pressure losses in the vapor must be included in calculation of Q_{max} in the given diode-type heat pipe structure. The temperature distributions in the housing wall and in the vapor indicate the existence of a passive zone of condensation, a zone plugged by liquid and extending over a part of the heat efflux zone in the heat pipe, the transition segment, and the reservoir. This passive zone becomes longer with rising temperature level and with increasing pipe inclination angle $\varphi > 0°$, which can be explained by expansion of the coolant and its overflow due to irregularity of the capillary structure [8].

The steady-state characteristic $Q_{rev} = f(\Delta T_{hp})$ in the reverse mode of operation with T_{o2} changing from 303 to 353°K ($T_{o1} = 293°K$) is shown in Fig. 1b. The values of Q_{rev} have been obtained assuming a completely plugged heat influx zone, i.e., $L_i < L_{plug} \leqslant L_i + L_{tr}$ (Fig. 4).

The characteristics of heat transfer in a diode-type heat pipe during reverse operation depend appreciably on the amount of excess coolant liquid in the vapor channel. For this reason, additional data on this mode of operation were obtained with a model of a diode-type heat pipe including an attachment for visual observation of the liquid plug.

It has been established that in the reverse mode of operation (with $T_{o1} \to T_{o2}$) the reservoir temperature rises together with T_{o2}. During the initial stage the coolant, after evaporating from the reservoir, condenses in the adiabatic zone and then in the heat influx zone of the heat pipe. In this process the heat of condensation is expended on raising the heat content in the diode-type heat pipe (at the outer surface of the thermodiode, under adiabatic conditions) or is carried away into the ambient medium. Waves of condensate form and propagate through the adiabatic zone to the heat influx zone of a diode-type heat pipe. In the vapor channel a liquid plug begins to form, from the beginning of the heat influx zone on, its front having the shape of a meniscus which is concave toward the vapor space. There appears a segment in the vapor channel which is plugged by liquid but allows heat transfer by vapor. The change in length of the plugged zone depends on the mass rate of liquid evaporation from the condensation zone and the reservoir. During the initial period of heating of the reservoir for a diode-type heat pipe made of copper, when $dT_{res}/d\tau \gtrsim 1.2°K/sec$, ejection of coolant from pores of the capillary reservoir structure occurs and a part of the vapor channel in the heat pipe becomes plugged almost instantaneously.

The graph in Fig. 2 depicts the experimentally determined characteristics of transition from steady state ($Q_i = 0$, $T_{o1} = 293°K$) to reverse mode of operation ($T_{o2} = 313°K$) and of transition from reverse mode to forward mode of operation ($Q_i = 0$ and $Q_i = 5$ W, respectively). Under adiabatic conditions in the heat influx zone, transition to steady-state reverse mode of operation will continue until all temperatures in the diode-type heat pipe become equalized tending toward T_{o2}. In the experiments the length of time τ_{rev} of operation in the reverse mode was selected on the basis of the requirement that the mean temperature of the heat influx zone in the heat pipe does not exceed $(T_{o1} + T_{o2})/2$.

According to the graphs in Fig. 2, both $dT_{i,tr}/d\tau$ and $dT_{tr}/d\tau$ change sharply at instants of time $\tau \sim 20$ sec and $\tau \sim 40$ sec. This can be explained by the plugging of the vapor channel

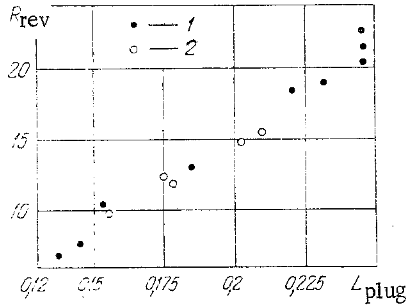

Fig. 3. Dependence of total steady-state thermal resistance R_{rev} (°K/W) in reverse mode of operation on length L_{plug} (m) of vapor channel plugged by liquid: 1) T_0 = 293°K, 2) T_0 = 313°K.

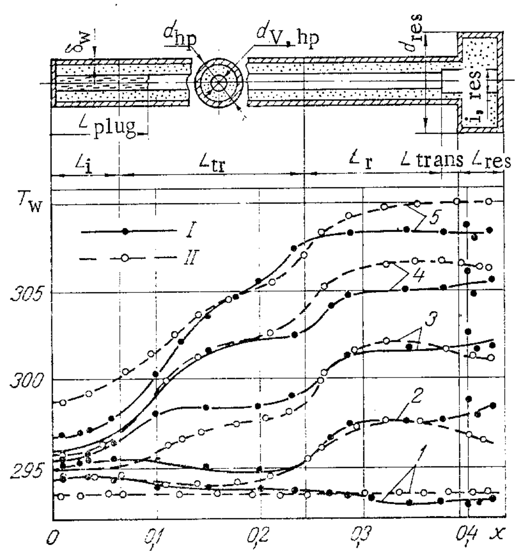

Fig. 4. Comparison of experimental and theoretical temperature fields in diode-type heat pipe during operation in transient reverse mode: 1) τ = 0; 2) τ = 20 sec; 3) τ = 40 sec; 4) τ = 80 sec; 5) τ = 120 sec; I) experiment; II) calculation according to relations (3)-(8); Q_i = 0, T_{01} = 293°K, T_{02} = 313°K; x coordinate (m), wall temperature T_W (°K).

by excess liquid evaporating from the condenser and the resulting change in the mode of heat transfer in these segments. Depending on the mass of excess liquid, the plugged zone can extend over the entire length of the heat pipe. In the experimental study of the transient reverse mode of operation this plugged zone was fixed on a segment $0 < L_{plug} \leqslant L_i + L_{tr}/2$. Beyond the upper bound of this range it was difficult to determine L_{plug}, because of heat leakage along the housing wall of the heat pipe (λ_W = 395 W/(m·°K)).

Transition from reverse mode to forward mode of operation always begins a decrease of T_{02} to T_{01}, with all other perturbation parameters remaining constant. A typical recovery of the forward mode after operation in the reverse mode (τ_{rev} = 660 sec) is shown in Fig. 2. Steady state in this transition mode was reached here after approximately 200 sec, a characteristic time for the given thermodiode configuration. In the experiment the recovery from operation in the reverse mode was stable within the φ = 0-10° range of pipe inclination angles and the T_0 = 293-353°K range of temperatures.

One basic performance characteristic of thermodiodes is the amount of heat W_{off} transferred by the diode-type heat pipe operating in the reverse mode during the transient period when it is completely turned off. In a survey of known methods available for determining W_{off} one must mention the theoretical model of a thermodiode with a liquid trap [5] and the design relation for a diode-type heat pipe with a liquid plug [3]. The theoretical model [5] does not take into account conductive heat transfer and distribution of coolant mass over thermodiode zones, while the design relation [3] is valid only for a special case. We will define the time τ_{off} during which a diode-type heat pipe is completely turned off as the period from the instant when the cooling temperature has begun changing to the instant when the entire vapor channel in the heat influx zone has become plugged with liquid. It should be considered, however, that reliable operation of a thermodiode requires that the plugged pipe segment be somewhat longer than L_i.

W_{off} was determined experimentally according to the relation

$$W_{off} = C_i\,[\overline{T}_i(\tau_{off}) - \overline{T}_i\,(0)] \pm Q_{amb}\,\tau_{off}\;. \tag{1}$$

The second term in expression (1) did not, according to measurements, exceed 2-10% of W_{off}. In the processing of data, τ_{off} was estimated graphically on the basis of the large change in the gradient of temperature T_{tr} (Fig. 2) in the transport zone, the latter located at a distance of 0.03 m from the end of the heat influx zone. In the experiment W_{off} varied from 1.5 J (at $\varphi = 0$ with $\tau_{off} \sim 8$ sec) to 165 J (at $\varphi = 5°$ with $\tau_{off} \sim 45$ sec). With the given thermodiode design it was possible to reduce τ_{off} to a minimum of approximately 8-10 sec, this time increasing somewhat at larger pipe inclination angles ($\varphi \leqslant 10°$) but remaining almost independent of changes in $dT_{res}/d\tau$.

An important characteristic of a diode-type heat pipe in reverse mode of operation is its thermal resistance R_{rev}, which was estimated here in the reverse mode under steady-state conditions. The trend of the dependence of this resistance on the length of the plugged zone is shown in Fig. 3. The experimental data have been processed according to the relation

$$R_{rev} = (\overline{T}_{w,e} - \overline{T}_{w,i})/Q_{rev}. \tag{2}$$

Transition from one length L_{plug} to another was effected through change of the pipe inclination angle, while a thermal flux Q_{rev} was produced by a Nichrome heater wound around the heat efflux zone of the heat pipe and around the reservoir. The graph in Fig. 3 indicates that the relation $R_{rev} = f(L_{plug})$ almost duplicates the trend of changes in thermal resistance due to conductive heat transfer.

The known methods of calculating the starting performance of heat pipes [10] and the transient modes in diode-type heat pipes [5] do not take into account the effect of axial heat conduction on the trend of the temperature field formation in these devices. The experimental results of other studies [1-3, 5, 7] as well as of this study indicate, meanwhile, that the heat transfer processes in diode-type heat pipes during transient periods depend appreciably on the axial thermal conductivity of the structural components.

For calculating the basic characteristics of diode-type heat pipes, a mathematical model of transient modes has been proposed which takes into account conductive heat transfer in the axial direction along components of the pipe structure. The assumptions made for simulation of transient heat-transfer processes are generally known and have been explained [10]. The model of transient heat transfer in a diode-type heat pipe is based on differential equations describing the balance of thermal fluxes in the form

$$C\,\frac{\partial T_w}{\partial \tau} = A\,\frac{\partial^2 T_w}{\partial x^2} + (T_V - T_w)/R_{int} - (T_w - T_0)/R_i + q(x,\ \tau). \tag{3}$$

Such equations are set up for the heat influx zone, the adiabatic zone, the heat efflux zone, and the reservoir (the origin of the x coordinate lies at the beginning of the heating zone).

Solution of system (3) by the composite numerical method [11] for the boundary conditions at the pipe ends

$$\frac{\partial T_w\ (0,\ \tau)}{\partial x} = 0, \quad \frac{\partial T_w\ (L_{hp},\ \tau)}{\partial x} = 0 \tag{4}$$

and the initial condition

$$T_w\ (x,\ 0) = f(x) \tag{5}$$

simultaneously with the integral equation for the vapor temperature

$$\sum_{j=1}^{m} \int_{0}^{L_j} [T_{W,j}(y_j, \tau) - T_V(\tau)]/R_{int,j} dy_j = 0; \; j = 1, \ldots, 4 \tag{6}$$

will yield the temperature distribution in the housing wall during operation in the forward mode.

A numerical performance analysis of diode-type pipes in the forward mode of operation has revealed an effect of axial heat conduction on the formation of the transient temperature field and on the length of the transient period. A comparison of the results of solution of problem (3)-(6) with experimental data pertaining to the given thermodiode configuration and with the results of tests performed on plain heat pipes indicates an agreement between the proposed model and the actual physical processes in these devices.

Because of difficulties encountered in the description of processes and conditions of transition to the reverse mode of operation, the following additional assumptions are made: 1) The coolant evaporated in the reservoir, while condensing, plugs the heat influx zone and the transport zone along the x axis; 2) there is no capillary coupling between the reservoir and the heat efflux zone of the heat pipe; 3) liquid fills the capillary reservoir structure uniformly over its length and radius. The system of Eqs. (3)-(6), moreover, is supplemented with relations for the length of the vapor channel plugged by excess liquid coolant and for the variable internal thermal resistance of the reservoir

$$L_{plug}(\tau) = \int_{0}^{\tau} \frac{[T_W(x, \tau) - T_0(x, \tau)]/\Omega_{i,res}}{\rho_L r_l \pi d_{V,hp}^2/4} d\tau, \tag{7}$$

$$0 \leqslant L_{plug} \leqslant L_i + L_{tr};$$

$$R_{int,res}(\tau) = \frac{\ln \dfrac{(d_{res} - 2\delta_W)}{\sqrt{\displaystyle\int_{0}^{\tau} \frac{4(T_W - T_0)/\Omega_{i,res}}{\pi r_l \rho_L L_{plug}(\tau)} d\tau + d_{i,res}^2}}}{2\lambda_{eff}/(d_{res} - 2\delta_W)}. \tag{8}$$

Here $R_{int,res} \geqslant 5 \cdot 10^{-5}$ $(°K \cdot m^2)/W$ [9]. Equations (7) and (8) are based on the balance of excess coolant mass in a diode-type heat pipe.

The graph in Fig. 4 depicts an example of how experimental and theoretical temperature fields in a diode-type heat pipe during operation in the reverse mode compare, this particular heat pipe having the design parameters $L_i = 0.07$, $L_{tr} = 0.175$, $L_r = 0.14$, $L_{trans} = 0.01$, $L_{res} = 0.025$, $d_{hp} \times \delta_W = (8 \times 1) \cdot 10^{-3}$, $d_{res} \times \delta_W = (23 \times 1) \cdot 10^{-3}$, $d_{V,hp} = 2 \cdot 10^{-3}$ m, and $d_{V,res} = 5 \cdot 10^{-3}$ m; the capillary structure made of sintered monodisperse fibers; the housing made of copper; and methyl alcohol used as coolant.

The initial temperature distribution $T_W(x, 0)$ was, for calculations, assumed to be uniform over the length. The thermal resistances inside and outside a diode-type heat pipe in Eqs. (3)-(8) were assumed to remain constant, except $R_{int,res}(\tau)$. The thermophysical properties were determined through calculations on the basis of temperature levels of thermodiode operation; R_{int} and λ_{eff} were determined according to recommendations [9].

The temperature of the heat efflux zone and that of the reservoir both rise during the initial part of the transient period. Owing to heat transfer by vapor, the transport zone and a part of the heat influx zone heat up. The process of predominant condensation in the heat influx zone during the $0 \leqslant \tau \leqslant 20$ sec period causes the temperature of this zone to rise higher than does the temperature of the transport zone. Because not all the condensed coolant returns from the heat influx zone through the capillary structure to the reservoir, a part of its vapor channel begins to become plugged by excess liquid. The temperature distribution in the housing of both influx heat zone and transport zone indicates that the mode of heat transfer within these zones of a diode-type heat pipe has changed. The temperature profiles in the heat influx zone and the adjacent region of the transport zone at instants of time $\tau = 40$, 80, and 120 sec are determined basically by the thermal conductivity of the housing, the saturated capillary structure, and the coolant liquid. The vapor temperature in a diode-type heat pipe remains below the reservoir temperature throughout the entire duration of the transient reverse mode. The difference between the trends in temperature changes at time τ = 120 sec is apparently caused by heat losses through the insulation.

On the whole, the agreement between theoretical and experimental data is satisfactory. The proposed mathematical model makes it possible to predict complex transient operating modes of heat pipes with variable thermal resistance.

NOTATION

C_i is the total thermal capacity of the heat influx zone; T_w, housing wall temperature of a diode-type heat pipe; $\bar{T}$, mean temperature; $T_{i,tr}$, temperature at the contact between heat influx zone and transport zone; Q_{amb}, heat loss to the ambient medium; q, linear thermal flux density; R_{int}, linear thermal resistance of the vapor—wall junction; R_i, linear thermal resistance of the wall—coolant junction; $C, A = \sum_{i=1}^{n} \lambda_i F_i$, coefficients in Eq. (3); λ_i, thermal conductivity of the i-th component along a section through the heat pipe (wall, saturated capillary structure, etc.); F_i, cross-sectional area of the i-th heat pipe component; L, length of a zone; ρ_L, density of the liquid coolant; r_l, specific latent heat of evaporation; d, diameter; Ω, total thermal resistance between reservoir and cooling medium. Subscripts: i, influx zone; tr, transport zone; r, heat removal zone; res, reservoir; V, vapor; o (amb), ambient medium; w, housing wall; "plug," segment plugged by liquid; L, liquid; "trans," transition segment; and yj, coordinate originating at the beginning of the j-th zone.

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