conditions and indicates that the mechanism of thermoacoustic oscillations for surface boiling ($P < P_{cr}$) and that for pseudoboiling ($P > P_{cr}$) are the same.

NOTATION

V, volume of the pseudobubbles; P, pressure; P_{cr}, critical pressure; t_{cr}, critical temperature; t_{in}, t_{out}, temperatures of the n-heptane at the input to the working section and at the output from it; t_s, temperature of the cooled surface of the heat-dissipating element; q, heat flow; W_γ, mass velocity.

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THEORY OF THE TEMPERATURE STABILIZATION PROCESS IN GAS-CONTROLLED HEAT PIPES

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This paper presents results of an analytical investigation of the temperature-stabilization process in gas-controlled heat pipes. Expressions are obtained for the temperature-sensitivity coefficients.

Heat-pipe technology includes the use of heterogeneous heat-transmitting systems with constant-volume change of state, i.e., gas-controlled heat pipes, which possess the temperature-stabilization property.

The investigators in [1] encountered phenomena associated with blockage of the condenser by a noncondensible gas. Cotter [3] noted that the length of the blocked zone depends on the heat flux. Katzoff [2] suggested using the condenser blockage effect for temperature stabilization. Bienert [4] was the first to describe analytically the temperature-stabilization process using a gas-controlled heat pipe, which was later accomplished by Marcus [5]. More complete information on gas-controlled heat pipes can be found in [6].

The present paper deals with an analytical investigation of the gas-controlled heat pipe and pays particular attention to the temperature-sensitivity coefficient.

In examining the temperature-stabilization process employing a gas-controlled heat pipe, we use the model shown in Fig. 1. In analyzing this one-dimensional model we make the following assumptions: 1) The heat pipe is a closed thermodynamic system; 2) the vapor—gas mixture in the blocked zone obeys all the ideal gas laws; 3) the vapor and the gas are incompressible; 4) the vapor motion is laminar; 5) the heat pipe is horizontal; 6) the heat-pipe regime is evaporative.

Equation of State for the Vapor—Gas in a Gas-Controlled Heat Pipe

In the equilibrium state of a gas-controlled heat pipe, in the absence of energy exchange with the surrounding medium, and with assumption 1, the heat-transfer agent and the

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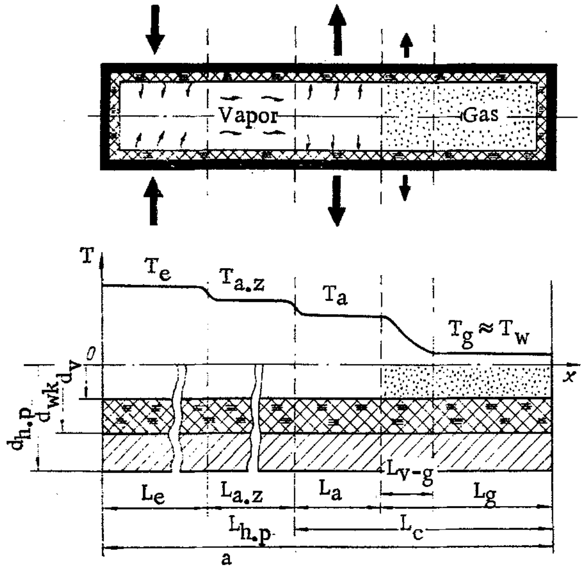

Fig. 1. Model of a gas-controlled heat pipe.

noncondensible gas are subject to the following conditions:

$$m_{\mathrm{hta}} = m_{\mathrm{v}} + m_{l} = \mathrm{const}, \quad (1)$$

$$V_{\mathrm{v.g.}l} = V_{\mathrm{v.g}} + V_{l} = \mathrm{const}, \quad (2)$$

$$P_{\mathrm{v.g}} = P_{\mathrm{v}} + P_{\mathrm{g}}. \quad (3)$$

We obtain the vapor pressure on the saturation line from the widely known Clausius–Clapeyron equation:

$$P_{\mathrm{v}} = \exp\left(C - \frac{M_{\mathrm{v}} r^*}{R_0 T}\right). \quad (4)$$

In Eq. (4) the constant of integration has a particular value for each heat-transfer agent, and the values of C calculated here are given in Table 1.

In the nonequilibrium state the thermal conditions will vary in a gas-controlled heat pipe. If the range of vapor temperature change is small, i.e., we are concerned with a gas-controlled heat pipe having a supplementary volume, we can use the Clausius–Clapeyron equation in the form

$$\ln \frac{P_2}{P_1} = \frac{M_{\mathrm{v}} r^*}{R_0} \left[\frac{T_2 - T_1}{T_1 T_2}\right]. \quad (5)$$

As is known [7], the temperature-stabilization process begins to operate from a specific value of the transmitted heat flux, called the minimum value. Therefore, after the binary vapor–gas mixture is separated, the equation for the total barometric pressure in a gas-controlled heat pipe will have the form

$$P = P_{\mathrm{a}} = P_{\mathrm{b}}. \quad (6)$$

In the blocked zone the molecules of gas and vapor have the same temperature, and therefore the Dalton law will hold. Taking account of the equation of state of an ideal gas and of the Clausius–Clapeyron equation for the vapor, both in the active and blocked zones, we obtain a general expression for the state of the vapor–gas mixture in a gas-controlled heat pipe, operating under steady-state conditions:

$$\frac{R_0 T_{\mathrm{g}}}{V_{\mathrm{b}}} \frac{m_{\mathrm{g}}}{M_{\mathrm{g}}} = \exp\left(C - \frac{M_{\mathrm{v}} r^*}{R_0 T_{\mathrm{v.a}}}\right) - \exp\left(C - \frac{M_{\mathrm{v}} r^*}{R_0 T_{\mathrm{v.b}}}\right). \quad (7)$$

Neglecting the partial pressure of the vapor in the blocked zone, we can represent its volume in the form

$$V_{\mathrm{b}} = \frac{m_{\mathrm{g}} R_0 T_{\mathrm{g.b}}}{M_{\mathrm{g}}} \left[\exp\left(C - \frac{M_{\mathrm{v}} r^*}{R_0 T_{\mathrm{v.a}}}\right)\right]^{-1}. \quad (8)$$

TABLE 1. Basic Parameters of Heat-Transfer Agents Used in Gas-Controlled Heat Pipes

Heat-transfer agent	Chemical formula	M	T_c, °K	$\frac{r^*M}{R_0}$, °K	$\frac{\sigma\rho r}{\mu}\cdot 10^{-6}$, W/cm²	C for P, torr
Helium	He	4,00	4,21	11,4	838	9,363
Hydrogen	H_2	2,01	20,38	116	518	12,347
Neon	Ne	20,18	27,09	213	3986	14,517
Nitrogen	N_2	28,01	77,36	685	8279	15,509
Carbon monoxide	CO	28,00	81,61	722	9442	15,502
Fluorine	F_2	38,00	84,95	774	17981	15,766
Argon	Ar	39,94	87,29	770	9666	15,476
Oxygen	O_2	32,00	90,18	841	17205	15,981
Methane	CH_4	16,00	111,7	1002	19199	16,625
Krypton	Kr	83,80	119,7	1112	962	15,945
F-14	CF_4	88,00	145,2	1553	2872	17,350
Xenon	Xe	131,3	165,1	1549	16576	16,037
Ethylene	C_2H_4	28,05	169,1	1516	23054	15,620
Ethane	C_2H_6	30,00	184,5	1803	31941	16,427
Acetylene	C_2H_2	26,03	189,6	2095	23425	17,704
F-23	CHF_3	70.00	191,2	2056	13569	17,406
F-13	CF_3Cl	104,5	191,8	1906	8916	16,592
F-116	C_2F_6	138,0	194,9	1983	5990	16,834
Propylene	C_3H_6	42,07	225,5	2280	11776	16,766
Propane	C_3H_8	44,09	231,1	2306	10113	16,633
F-22	$CHClF_2$	86,48	234,4	1849	12528	14,543
F-115	C_2ClF_5	154,0	234,5	2384	3150	16,821
Ammonia	NH_3	17,00	239,8	2862	20419	18,590
F-12	CCl_2F_2	120,9	243,4	2452	9675	16,729
F-40	CH_3Cl	50,49	249,4	2668	29415	17,352
Isobutane	C_4H_{10}	58.12	261.4	2600	14921	16,601
F-142	$C_2H_3ClF_2$	100,4	264,2	2716	9534	16.935
F-114	$C_2Cl_2F_4$	171,0	276,7	3089	5292	17,818
F-21	$CHFCl_2$	102,9	282,1	3061	15658	17,505
F-11	$CFCl_3$	137,4	301,9	2908	11333	16,287
Diethyl ether	$C_4H_{10}O$	74,10	307,6	3133	17544	16,840
Pentane	C_5H_{12}	72.15	309,2	3139	12744	16,805
F-113	$C_2F_3Cl_3$	187,4	320,7	3408	9990	17,281
Methanol	CH_4O	32,00	337,7	4424	49175	19,755
Carbon tetrachloride	CCl_4	153,8	349,9	3639	11990	17,055
Ethyl acetate	$C_4H_8O_2$	88,10	350,3	3887	19482	17,751
Ethanol	C_2H_6O	46,07	351,5	5266	26676	21,636
Benzene	C_6H_6	78,11	353,3	3801	42710	17,413
Acetone	C_3H_6O	58,08	359,3	3726	30237	17,025
Propanol	C_3H_8O	60,09	370,4	3190	16354	15,267
Heptane	C_7H_{16}	100.2	371,7	3915	12857	17,187
Water	H_2O	18,02	373,2	5894	48955	22,448
Toluene	$C_6H_5CH_3$	92.13	383,7	3487	17515	15,743
Pyridine	C_5H_5N	79,00	389,3	4347	61921	17,821
Butanol	$C_4H_{10}O$	74,12	390,9	5374	16333	20,402
Chlorobenzene	C_6H_5Cl	112,6	405,2	4497	21874	17,753
o-Xylene	C_8H_{10}	106,2	417,6	4501	20333	17,433
Cumene	C_9H_{12}	120,2	425,6	4682	13057	17,672
Naphthalene	$C_{10}H_8$	128,2	491,0	4963	—	16,763
Diphenyl	$C_{12}H_{10}$	154,2	528,2	5970	—	17,957
Dowtherm	—	165,8	530,1	5850	—	17,690
Diphenyl ether	$(C_6H_5)_2O$	170,1	531,0	5769	—	17,519
Glycerol	$C_3H_5(OH)_3$	92,04	563,0	8749	—	22,195

For a linear gas-controlled heat pipe we can write expressions for the lengths of the blocked and the active zones:

$$L_b = \frac{m_g R_0 T_{g.b}}{M_g F_g}\left[\exp\left(C - \frac{M_v r^*}{R_0 T_{v.a}}\right)\right]^{-1}, \tag{9}$$

$$L_a = L_c - \frac{m_g R_0 T_{g.b}}{M_g F_g}\left[\exp\left(C - \frac{M_v r^*}{R_0 T_{v.a}}\right)\right]^{-1}. \tag{10}$$

To obtain a high degree of temperature stabilization, an additional reservoir is used with a gas or an expanded volume. For gas-controlled heat pipes of this construction the expression (10) takes the form

$$L_a = L_c - \frac{m_g R_0 T_{g.b}}{M_g F_g}\left[\exp\left(C - \frac{M_v r^*}{R_0 T_v}\right)\right]^{-1} + \frac{V_r}{F_v}. \tag{11}$$

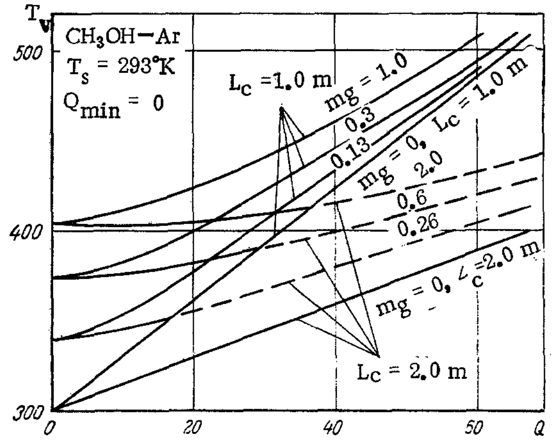

Fig. 2. Calculated vapor temperature as a function of transmitted heat flux. T_v, °K; Q, W.

Heat Transfer in a Gas-Controlled Heat Pipe

The thermal resistance of a simple (uncontrolled) heat pipe has the form

$$R_{h.p} = R_e + R_c = \frac{R'_e}{F_e} + \frac{R'_c}{F_c}, \tag{12}$$

where R is the thermal resistance referred to the heat removal or supply area and characterizes the heat pipe with the condition $F_e \neq F_c$. For a gas-controlled heat pipe Eq. (12) takes the form

$$R_{gchp} = R_e + \frac{R_a R_b}{R_a + R_b}. \tag{13}$$

Under the assumption of a constant cross section for the vapor flux along the condenser length, from Eqs. (10), (12), and (13) one can obtain the basic equation for heat transfer in a gas-controlled heat pipe:

$$Q = (T_v - T_s)\left\{\frac{F_c}{R'_a} - \frac{R'_b - R'_a}{R'_a R'_b}\cdot\frac{\Pi m_g R_0 T_g}{F_v M_g}\left[\exp\left(C - \frac{M_v r^*}{R_0 T_v}\right)\right]^{-1}\right\}. \tag{14}$$

In engineering calculations of gas-controlled heat pipes we neglect the heat transfer through the blocked zone in many cases ($R'_b \to \infty$):

$$Q = \frac{\Pi (T_v - T_s)}{R'_a}\left\{L_c - \frac{m_g R_0 T_b}{F_v M_g}\left[\exp\left(C - \frac{M_v r^*}{R_0 T_v}\right)\right]^{-1}\right\}. \tag{15}$$

The use of a supplementary reservoir or an elongated volume permits a high degree of temperature stabilization. For these structures and boundary conditions of the second and third kinds at the outer surface of the evaporator (a boundary condition of the third kind is assumed at the outer surface of the condenser), Eq. (15) takes the form

$$Q = \frac{\Pi (T_v - T_s)}{R'_a}\left\{L_c + \frac{V_r}{F_v} - \frac{m_g R_0 T_g}{M_v F_v}\left[\exp\left(C - \frac{M_v r^*}{R_0 T_v}\right)\right]^{-1}\right\}, \tag{16}$$

$$T_v = T_s + (T_{true} - T_s)\left\{2 + \frac{V_r}{V_c} - \frac{m_g R_0 T_g}{M_v V_c}\left[\exp\left(C - \frac{M_v r^*}{R_0 T_{v.a}}\right)\right]^{-1}\right\}. \tag{17}$$

Expression (17) was obtained from Eq. (15) under the assumption $R'_e = R'_a$ and $n_{h.p} = 1$.

By way of example we calculated a gas-controlled heat pipe with the following parameters: $L_c = 1$ m, $d_v = 10^{-2}$ m, $R'_a = 10^{-1}$ m·°K/W, and $T_s = 293$°K, the heat-transfer agent being methanol and the noncondensible gas being argon. The results are shown in Fig. 2, in the form of the relationship $T_v = f(Q)$. It was assumed in the calculations that $Q_{min} = 0$. One can see from the figure that the vapor temperature stabilization is greater, the higher the vapor temperature, or, in other words, the greater the mass of noncondensible gas, for constant geometrical parameters. The curves $T_v = f(Q)$ for a gas-controlled heat pipe without an additional gas reservoir draw near to the curve $T_v = f(Q)$ for a simple uncontrolled heat pipe

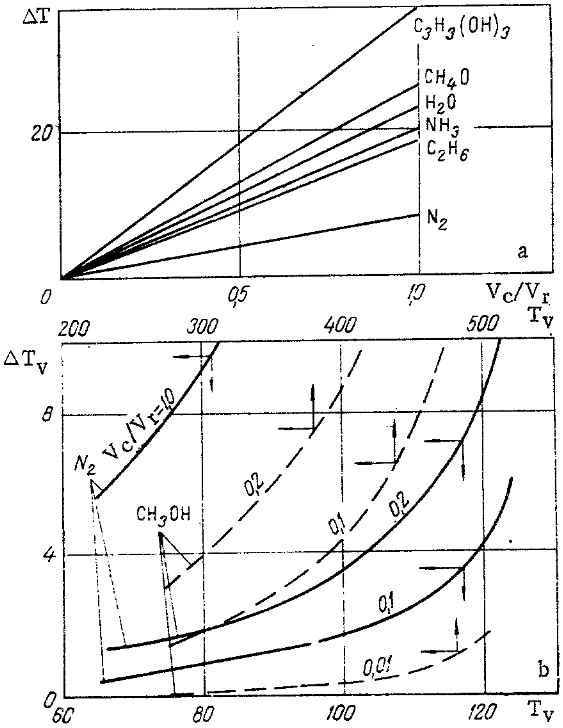

Fig. 3. The vapor temperature increment as a function of the ratio V_c/V_r (a) and of the absolute stabilization temperature (b).

as the transmitted heat flux increases. The slight slope of the curves near zero shows that a higher degree of temperature stabilization is achieved for small transmitted heat flux, corresponding to minimum vapor temperature in the stabilization range $Q > Q_{min}$. When an additional gas reservoir is used the slope of the curves $T_v = f(Q)$ decreases; i.e., there is more vapor temperature stabilization for such heat pipes. Analysis of the curves shown in Fig. 2 indicates that a gas-controlled heat pipe with an additional gas reservoir can be regarded as a simple gas-controlled heat pipe in which the condenser length is increased by the amount V_r/F_v. Here the shorter the length of the active zone (for constant condenser length), the greater the degree to which the vapor temperature is maintained constant.

In developing gas-controlled heat pipes one must determine the volume of the supplementary reservoir to achieve minimum vapor temperature variation for maximum variation of transmitted heat flux. Using the fact that the number of moles of gas m_g/M_g is constant and also the conditions $Q_{min} = 0$ and $Q_{max} = (F_c/F'_{sup})(T_{v.max} - T_s)$, one can use Eq. (16) to obtain an expression for the ratio of the supplementary reservoir volume to the condenser volume:

$$\frac{V_r}{V_c} = \left[\frac{P_{v\cdot max}}{P_{v\cdot min}} - 1\right]^{-1} \tag{18}$$

Using Eq. (5), we obtain an expression for the ratio (18) in the following form:

$$\frac{V_r}{V_c} = \left[\exp\frac{r^*M}{R_0}\left(\frac{T_{v\cdot max} - T_{v\cdot min}}{T_{v\cdot max}\cdot T_{v\cdot min}}\right) - 1\right]^{-1}. \tag{19}$$

With the equations obtained one can calculate the volume of a gas-controlled reservoir for a given condenser volume as a function of the vapor pressure increment [Eq. (18)] or of the allowable vapor temperature increment [Eq. (19)].

Temperature Sensitivity of a Gas-Controlled Heat Pipe

In the most general case the sensitivity coefficient is the ratio of an increment in the destabilizing parameter to an increment in the stabilized quantity. Analysis of the use of this parameter to evaluate the operation of heat pipes shows that in practice one can use both thermal or temperature coefficients and sensitivity coefficients relating to heat-flux density or gas pressure, and so on.

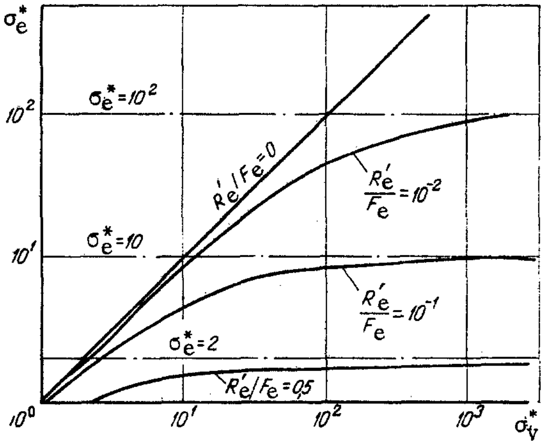

Fig. 4. Temperature-sensitivity coefficient for the external wall of the evaporator as a function of the vapor temperature-sensitivity coefficient σ_e^*, W/°K.

For the parameter describing control of vapor temperature in a gas-controlled heat pipe, we use the temperature-sensitivity coefficient $\sigma_v^* = dQ/dT_v$. Sometimes one uses the thermal-control coefficient as a characteristic of a gas-controlled heat pipe, this being the inverse of the temperature-sensitivity coefficient. From the engineering viewpoint, one is most interested in the temperature-sensitivity coefficient at the surface of the evaporator $\sigma_e^* = dQ/dT_e$ or of the condenser $\sigma_c^* = dQ/dT_c$. Without an appreciable error one can regard the characteristics as being linear for the majority of gas-controlled heat pipes with an additional gas reservoir, assuming that the heat transmission is constant along the heat pipe. Then from the practical viewpoint one is most interested in the temperature-sensitivity coefficient in the form

$$\sigma^* = \frac{Q_{\max} - Q_{\min}}{T_{\max} - T_{\min}}. \tag{20}$$

Analysis of this expression shows that σ^* may increase due to a decrease in $Q_{\min}$ or to a decrease in the denominator. The temperature difference $T_{\max} - T_{\min}$ can be expressed in terms of the volume ratio V_r/V_c [Eq. (19)]. To do this we write the exponent function in Eq. (19) in the form of a MacLaurin series:

$$\exp\left(\frac{r^*M}{R_0}\cdot\frac{T_{v,\max}-T_{v,\min}}{T_{v,\max}\cdot T_{v,\min}}\right) = 1 + \frac{r^*M}{R_0}\cdot\frac{(T_{v,\max}-T_{v,\min})}{T_{v.\max}\cdot T_{v,\min}} + \left(\frac{r^*M}{R_0}\cdot\frac{T_{v,\max}-T_{v,\min}}{T_{v,\max}\cdot T_{v,\min}}\right)^2 + \ldots. \tag{21}$$

In analyzing this expression we restrict attention to the first two terms, because the subsequent terms are small. Then, taking account of expressions (19) and (21), we can write the denominator in Eq. (20) in the following form:

$$\Delta T_v = \frac{R_0\bar{T}_v^2}{Mr^*}\cdot\frac{V_c}{V_r}, \tag{22}$$

where the average vapor temperature is

$$\bar{T}_v^2 = T_{v.\max}\cdot T_{v,\min} + \frac{\Delta T_v^2}{4}.$$

From Eq. (22) we can conclude that the vapor temperature increment increases with increase in the ratio V_c/V_r. Figure 3a shows a group of straight lines for some heat-transfer agents, calculated using Eq. (22). From the figure it can be seen that cryogenic heat-transfer agents possess appreciable advantages in comparison with low-temperature agents, since they have low vapor temperature increments ΔT_v, which result from the low value of the stabilization temperature T_v.

To investigate the behavior of the function $\Delta T_v = f(T)$ over the whole operating temperature range $\Delta T_0 = T_{cr} - T_{tr}$, ΔT_v was calculated for methanol and nitrogen (Fig. 3b) heat pipes.

From analysis of Fig. 3b one can conclude that a gas-controlled heat pipe with a constant gas reservoir will show better thermal stabilization at temperatures close to the triple point.

Taking account of Eq. (22), one can write an expression for the general temperature-sensitivity coefficient with respect to the vapor:

$$\sigma_v^* = \frac{\Pi L_{a \cdot max}}{R'_a} \cdot \frac{T_v - T_s}{T_v^2} \frac{Mr^*}{R_0} \cdot \frac{V_r}{V_c}. \tag{23}$$

The expression obtained for the temperature-sensitivity coefficient is valid for small changes in vapor temperature. It follows from Eq. (23) that to obtain large values of σ_v^* one must choose a heat-transfer agent with a greater curvature in the relationship between vapor pressure and temperature at the saturation line. A promising method for increasing σ_v^* is to increase the ratio V_r/V_c.

When one deals with applied problems using a gas-controlled heat pipe one frequently has to stabilize the temperature of the external condenser surface, for which the temperature-sensitivity coefficient has the form

$$\sigma_c^* = \left[\sigma_v^{*-1} - \frac{Q_{max} R'_a / F_{max} - Q_{min} R'_a / F_{min}}{Q_{max} - Q_{min}} \right]^{-1}. \tag{24}$$

Analysis shows that the second term in Eq. (24) begins to dominate for large values of σ_v^*.

The general temperature-sensitivity coefficient for the external evaporator surface can be written in the form

$$\sigma_v^* = \left[\sigma_v^{*-1} + \frac{R'_e}{F_e} \right]^{-1}. \tag{25}$$

In this expression σ_v^* is determined in conjunction with Eq. (23).

Analysis of Eq. (25) shows that an unlimited increase in the temperature-sensitivity coefficient of the vapor leads to a situation where the second term in the expression begins to dominate. Figure 4 shows calculated values of $\sigma_e^* = f(\sigma_v^*)$ for certain ratios R'_e/F_e. From the figure it can be seen that each of the curves represented has two asymptotes. With an increase in σ_v^* the curve draws near to the asymptote $\sigma_e^* = F_e/R_e$, and for a decrease in σ_v^* it draws near to the asymptote $\sigma_e^* = \sigma_v^*$. Therefore, in order to obtain a high evaporator temperature stabilization, one must increase the heat supply area and decrease the thermal resistance of the evaporator. In analogy with the limit in heat transmission in porous heat pipes, one can introduce a bound on thermal stabilization in gas-controlled heat pipes:

$$\sigma_e^* \to \frac{F_e}{R'_e} \quad \text{as} \quad \sigma_v^* \to \infty. \tag{26}$$

For an experimental check of the design relationships obtained, we investigated low-temperature gas-controlled heat pipes using distilled water and ethanol. The experimental results for the temperature-stabilizing parameters were presented in [8, 9]. Good agreement was obtained between the experimental data and the theoretical relationships (16), (22), and (23). The experimental values of the temperature-sensitivity coefficient differed from the theoretical values by 15%.

NOTATION

L, length; T, temperature; F, area; Q, heat flux; M, molecular weight; R_0, universal gas constant; m, mass; r*, latent heat of vaporization; V, volume; Π, perimeter; R, total thermal resistance; R', reduced thermal resistance; σ*, temperature-sensitivity coefficient; P, pressure; σ, surface-tension coefficient; ρ, density; μ, dynamic viscosity. Indices: h.p, heat pipe; g, gas; a, active zone of condenser; c, condenser; s, heat source; r, gas reservoir; e, evaporator; o, operating range; cr, critical point; tr, triple point; w, wall; wk, wick; *l*, liquid; a.z, adiabatic zone; v, vapor; hta, heat-transfer agent.

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EXPERIMENTAL INVESTIGATION OF THE THERMODYNAMIC CRISIS OF FILM BOILING

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UDC 536.423.1

It is shown on the basis of experimental data that the thermodynamic crisis of film boiling depends on the material at the surface of the cooled solid. A theory substantiating the effect of the thermophysical characteristics of the surface on the crisis temperature in steady-state film boiling is proposed.

Reduction of the cooling time for metal structures washed by low-boiling liquids constitutes a pressing problem involving many cryogenic systems and devices. In cooling objects to operating temperatures under conditions where the vapor temperature near the vapor film—dispersoid boundary is close to the saturation temperature, the total cooling time can be reduced by diminishing the relative share of the wall's heat resource taken off during film boiling, i.e., by switching to intermediate, more intensive types of boiling at a temperature higher than $T = T_{lp}$.

Many investigators have noted on the basis of experimental results that the film boiling crisis, the transition to nucleate boiling, depends on the material at the surface of the cooled solid [1-5]. In particular, it has been found that coating of the solid to be cooled with a thin film of a material possessing a low thermal-conductivity coefficient sometimes leads to a higher temperature of thermodynamic crisis of film boiling. With the exception of [1], the many attempts at explaining this phenomenon were unsuccessful. In our opinion, this is due to a lack of a sufficient amount of experimental data and the contradictory nature of the data on the relative effect on the film boiling crisis of coating the parts to be cooled with various materials. It has been proved [3] that a stationary model cannot be used to explain the effect of coatings (or the effect of the material at the surface). On the other hand, in many cases an increase in the coating thickness (beyond certain values) had a similar effect on the cooling rate of a solid, as in the case of the stationary model. This necessitated new theoretical and experimental investigations.

Figures 1 and 2 show the effect of various coatings and of their thickness on the cooling time of specimens in liquid nitrogen. The cooling is accelerated as a result of passage to efficient types of boiling (in comparison with stable film boiling) at higher mean temperatures of the specimen.

The experimental values of the mean-mass temperature of a solid cooled in liquid nitrogen corresponding to the crisis of stable film boiling are obtained by processing the cooling curves $T(\tau)$ for the experimental specimens, the dimensions of which are determined (on the basis of theoretical estimates, using the similarity theory and regular conditions) with an allowance for the virtual absence of a temperature gradient along the wall thickness; i.e., $\mathrm{Bi} \ll 1$ and $T_w \cong T_{w_0}$. The temperature of the experimental specimen at the time of sharp change in the slope of the cooling curve is used as the temperature boundary of the thermodynamic crisis of film boiling (Fig. 3).

The absence of a clearly defined temperature boundary of change under the heat-transfer conditions is a characteristic of the film boiling crisis. This cannot be explained only by the adopted method of experimental data processing; it is connected with the complex effect

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