We can obtain an expression for K by determining the reduced velocity of the steam W" from Eq. (5)

$$K = \left[\frac{C}{1+\left(\frac{\rho''}{\rho'}\right)^{1/4}}\right]^2 \left[D\sqrt{\frac{g(\rho'-\rho'')}{\sigma}}\right]^{1/2}. \tag{10}$$

Figure 3 shows the results of the analysis of the experimental data in terms of the criterion K as a function of the parameter $D\sqrt{g(\rho' - \rho'')/\sigma}$. The computed curves, obtained using Eq. (10) taking into account (8), are indicated here as well.

In the pressure range from 0.5 to 8 MPa, relation (10) agrees quite well with the experimental data and is recommended for use with calculations of critical heat loads in vertical tubes with the lower end sealed.

NOTATION

W', W", reduced velocities of water and steam; ρ', ρ", density of water and steam; B, diameter of the tube; g, acceleration of gravity; Q, total heat flux; r, heat of evaporation; σ, coefficient of surface tension.

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HYDRAULIC RESISTANCE FOR LIQUID FLOW WITH BOILING-UP IN A PARTIALLY HEATED CHANNEL

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UDC 532.5.013.12:536.423

An approximate analytic solution is obtained to the problem of calculating the hydraulic resistance. The hydraulic characteristics are analyzed and the region for static instability is determined.

At the present time, heat removal systems with long channels are constructed in cryogenic technology. These systems are characterized by a relatively low heat flux density, which allows for heat removal in the single-phase (economizing) region with subsequent boiling-up of the liquid along the adiabatic (unheated) section of the channel. Thus, for example, the low-temperature state of the heat-absorbing shields of thermal vacuum chambers is maintained in this manner with the help of a natural circulation loop with an external boiling zone [1], as is the cooling of cryogenic condensation pumps and cryogenic electrical devices with forced circulation of liquid nitrogen and helium in parallel channels.

Boiling-up of a liquid in this case is understood to be the process of evaporation caused by a decrease in pressure. References [3] and [3] examine primarily the boiling-up process as resulting from an increase in the flow rate due to a decrease in the channel cross section or a sudden change in pressure. In this case, the kinetic energy of the flow is low in comparison with the enthalpy, and for this reason, the change in the flow rate along the length of the channel can be neglected. However, the change in pressure along the channel,

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 39, No. 5, pp. 842-847, November, 1980. Original article submitted September 11, 1979.

0022-0841/80/3905-1209$07.50

caused by losses due to friction and decrease in the leveling component, is comparable to the absolute pressure of the liquid and is accompanied by additional evaporation. The relatively low rate at which the pressure changes along the length of the channel and the low mass vapor content create favorable conditions for thermodynamic equilibrium. Under these conditions, the problem can be formulated as follows.

A liquid, the enthalpy of which is known, enters into the vertical adiabatic channel (Fig. 1). A heat flux, which can be assumed to be concentrated in a single section, is input to the liquid in the initial section. The absolute pressure of the liquid decreases in proportion to its motion due to losses to friction and changes in the leveling component. For some cross section, at a distance l from the outlet, the liquid begins to boil and above this section there is a two-phase flow, the vapor content of which increases due a decrease in pressure. It is necessary to determine the hydraulic resistance as a function of the flow rate, assuming that the pressure at the channel outlet is known.

Let us make the following assumptions: the flow of the liquid is uniform; the two-phase flow can be assumed to be homogeneous with thermodynamic equilibrium between the liquid and vapor phases; the liquid is incompressible; losses due to acceleration can be neglected; the physical properties of the liquid and gas are constant; the vapor content at the outlet is relatively low; the coefficient of friction does not depend on the flow rate.

We will first examine the section of the channel in which boiling-up occurs ($L - l < z < L$). Taking into account the details of the problem and the assumptions made above, we will represent the equation of motion and energy for the adiabatic channel in the form

$$\frac{dP}{dz} = \xi G^2 v_m + \frac{g}{v_m}; \tag{1}$$

$$\frac{di_m}{dz} = 0, \tag{2}$$

where

$$v_m = v'(1 + xn); \tag{3}$$

$$i_m = i' + xr;$$

$$\xi = \frac{\lambda}{2d};\ n = \frac{v''}{v'} - 1. \tag{4}$$

In this case, it is useful to use the Clausius—Clapeyron equation for the equation of state:

$$\frac{dP_s}{dT_s} = \frac{r}{T_s(v'' - v')}. \tag{5}$$

If Eq. (5) is written in a finite-difference form and we change over to the change in enthalpy, then we obtain

$$i = a + bP_s, \tag{6}$$

where

$$a = i_2 - c_p T_{s2} \frac{v'' - v'}{r} P_{s2};\ b = c_p T_{s2} \frac{v'' - v'}{r}.$$

The values of the coefficients a and b in (6) are determined only by the physical properties of the liquid at the outlet. In particular, for liquid nitrogen in the pressure range 0.1-0.3 MPa, the values of these coefficients can be taken as follows: $a = 1.92 \cdot 10^4$ J/kg, $b = 0.1$ J•m²/kg•N, and for helium at pressures ranging from 0.1-0.2 MPa, $a = 3.8 \cdot 10^3$ J/kg and $b = 5.9 \cdot 10^{-2}$ J•m²/kg•N.

Since we are only considering the boiling region, taking into account thermodynamic equilibrium, we have

$$P = P_s. \tag{7}$$

The system of equations (1)-(4), (6), and (7) is closed with respect to the variables P, P_s, i_m, i, and x. Using the boundary condition at the outlet, we obtain a solution in the form

$$\Delta P = P - P_2 = \frac{u - [(u^2 + k)\exp(-2\xi G^2 v' \beta z) - k]^{1/2}}{\beta}, \tag{8}$$

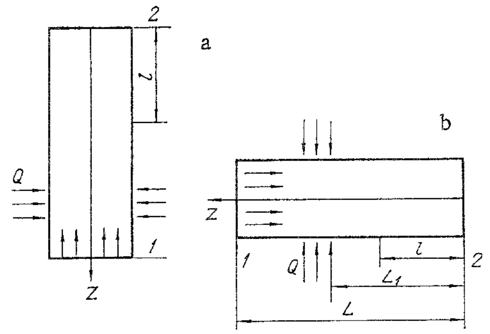

Fig. 1. Liquid flow with boiling in vertical (a) and horizontal (b) channels.

where

$$u = 1 + \frac{(i_1 - i_2')\,n}{r};$$

$$\beta = \frac{bn}{r};\; k = \frac{g}{\xi G^2 (v')^2}.$$

The quantity u represents the ratio of the specific volume of the mixture at the outlet to the specific volume of the liquid.

For a horizontal channel k = 0 and

$$\Delta P = \frac{u\,[1 - \exp(-\xi G^2 v' \beta z)]}{\beta}. \tag{9}$$

As indicated above, the liquid begins to boil-up in the section $z = l$, and here, $i^l = i'$. Using function (6) and taking into account the fact that the parameters of the mixture at the outlet are known, we obtain the pressure differential in the boiling section

$$\Delta P_f = \frac{i_l - i_2'}{b}. \tag{10}$$

Thus, the total pressure differential in the boiling region is determined by the difference in the enthalpies of the liquid at the inlet into the adiabatic region and on the saturation line for the conditions at the outlet and by the thermophysical properties of the liquid, and does not depend either on the orientation of the channel, or its geometric dimensions, or on the parameters that determine the regime. The influence of the latter factors is manifested only in the length of the boiling region, which is computed with the use of formulas (7) and (9), and for a vertical channel equals

$$l = \frac{1}{2\,\xi v' G^2 \beta} \ln \frac{u^2 + k}{1 + k}, \tag{11}$$

while for a horizontal channel, it equals

$$l = \frac{1}{\xi G^2 v' \beta} \ln u. \tag{12}$$

It is evident from the last two equations that as the flow rate increases and the diameter decreases, the length of the boiling region decreases, but the pressure differential across it, as indicated above, remains constant. In vertical channels, the length of the boiling region is less than in horizontal channels.

The results obtained permit us to proceed to the analysis of thermohydraulic regimes for partially heated channels.

Let us consider a horizontal channel (Fig. 1a), for which the total hydraulic resistance can be represented as the sum of the resistances for the single-phase region and for the boiling region. In this case,

$$i_l = i_1 + \frac{Q}{Gf} \tag{13}$$

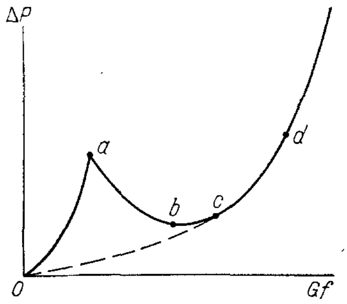

Fig. 2. Hydraulic characteristic of a partially heated channel with flow of a boiling liquid ($o\alpha bd$ and ocd are the unstable and stable characteristics, respectively). ΔP, N/m²; Gf, kg/sec.

and taking into account (9) and (12),

$$\Delta P = \psi\xi G^2 v' (L - l) + \frac{sQ}{Gfb}, \tag{14}$$

where

$$s = 1 + \frac{(i_1 - i_2') Gf}{Q}; \; \psi = 1 + \varphi_0 \frac{2d}{\lambda (L - l)},$$

while the value of l is determined from formula (12). For circulating cryogenic systems, we can assume that $i_0 = i'_2$ and $s = 1$.

As follows from formula (13), the hydraulic characteristic of such a channel with a constant value heat load has a peculiar behavior (Fig. 2). For large values of the mass velocity, the pressure differential is primarily determined by the hydraulic resistance in the single-phase region. The hydraulic resistance in this region decreases in proportion to the decrease in the velocity, as well as due to the shorter length, and the resistance increases in the two-phase region due to the increase in the enthalpy of the liquid beyond the heated section. At the same time, the value of the total hydraulic resistance can have a local minimum.

When the beginning of the boiling section reaches the inlet for the heat load ($l = L_1$), the dependence of the hydraulic resistance on the flow rate changes. In this case, the pressure differential is determined by formula (9) and

$$\lim_{G \to 0} \Delta P = 0.$$

The multi-valuedness of the hydraulic characteristic of the channel is physically similar to the same phenomenon that occurs with the motion of a vapor–liquid mixture in a heated channel with an economizer [4].

The lack of uniqueness in the characteristic could be a reason for the static instability [5], and it can be avoided as follows: operation in the single-valued region of the thermohydraulic characteristic; choosing the geometric dimensions of the channel so as to ensure uniqueness in the characteristic; increasing the local hydraulic resistance in the single-phase region, thereby changing the characteristics of the channel.

Operation in the single-phase region is achieved by choosing the flow rate so that the condition $\Delta P_d > \Delta P_\alpha$ is satisfied (Fig. 2). The pressure differential and the flow rate at the point α on the characteristic are determined from a simultaneous solution of Eqs. (9) and (12) under the condition that the length of the boiling segment equals the distance from the heated section to the channel outlet. The value of G_d is then determined by iteration.

In order to ensure uniqueness of the characteristic using the second method, let us determine the value of the flow rate G_b at the extremum point. First, we will show that the function (14) can be approximately represented in the form

$$\Delta P = \xi G^2 v' L + \frac{sQ}{Gfb}, \tag{15}$$

i.e., the decrease in the hydraulic resistance of the single-phase region due to its decreased length is not taken into account. This is explained by the fact that for large flow rates $l/L \ll 1$, while for low flow rates the second term in the equation is the determining term. Using this assumption and the fact that at the extremum point $(d\Delta P/dG) = 0$, we determine the value of the mass velocity at the point b

$$G_b = \frac{sQ}{2\,\xi v' L b f}. \tag{16}$$

As evident from Fig. 2, the characteristic will be single-valued if the point of discontinuity on the curve will lie to the right of the extremum point (curve oc). This is possible under the condition that the distance from the heated section to the channel outlet L_1 is less than the length of the boiling region for maximum velocity G_b. Using this condition and formula (15), it can be shown that the characteristic is single-valued if

$$\frac{L_1}{L} < \frac{2\,\psi \ln(1+R)}{R}, \tag{17}$$

where

$$R = \frac{b^{1/3}\,n\,(v')^{1/3}}{r}\;\frac{\lambda^{1/3}L^{1/3}\psi^{1/3}}{f^{2/3}\,d^{1/3}}\;(sQ)^{2/3}.$$

This relation established a connection between the heat load, the geometric dimensions of the channel, the coefficient of friction, and the local hydraulic resistances, on the one hand, and the thermophysical properties of the liquid, on the other. In particular, when liquid helium flows in a circular channel the condition for single-valuedness has the form

$$\frac{\lambda s^2 Q^2 L_1}{d^5}\;\frac{\psi}{(2\psi-1)^3} < 2.73\cdot 10^{14},\ \frac{\mathrm{W}}{\mathrm{m}^4}. \tag{18}$$

The strong influence of the diameter, heat load, and local hydraulic resistances on the thermohydraulic characteristic of the channel should be noted. As the hydraulic resistance of the adiabatic section increases, the multivaluedness of the hydraulic characteristic appears at lower values of the heat load. All of the results obtained can be applied to the case of a distributed heat load in the single-phase region, if the boiling zone is located on the adiabatic part of the channel.

Thus, the use of the homogeneous flow model together with the Clausius–Clapeyron equation has led to simple relations for calculating the hydraulic resistance with the flow of a boiling liquid in channels and has elucidated the properties of the thermohydraulic regimes of such channels.

In using the results of the present work, the appropriateness of the assumptions made for conditions of the problem should be kept in mind.

NOTATION

c_p, isobaric heat capacity; d, diameter; f, area of the transverse cross section; G, mass velocity of the liquid; g, acceleration of gravity; i, enthalpy (i' is the enthalpy of the liquid on the boiling curve); L, length of the channel; L_1, distance from the section at which the heat load is applied to the end of the channel; l, length of the section with the boiling liquid; P, pressure; ΔP, pressure differential; ΔP_f, pressure differential in the boiling region; r, heat of vaporization; T, temperature; v, specific volume (v' is the specific volume of the liquid, v" is the specific volume of the saturated vapor); x, mass vapor content; z, coordinate; λ, coefficient of friction; φ, coefficient of local hydraulic resistance in the single-phase region. The indices indicate the following: a, b, c, and d, parameters that correspond to points on the curve in Fig. 2; 1 and 2 are parameters that correspond to the inlet and outlet section of the channel; m, parameters of the two-phase mixture; s labels equilibrium parameters; l, parameters corresponding to the section with onset of boiling-up.

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CHOICE OF HEAT-TRANSFER AGENT FOR HEAT PIPES OPERATING IN THE TEMPERATURE RANGE 300-500°C

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UDC 621.565.931.94

The authors analyze possible heat-transfer agents for heat pipes operating within the structural members of a gas-turbine motor. The authors have examined silicone liquids, for which they present thermophysical properties and the results of heat-pipe tests.

A number of plant and equipment items operate under complex thermal conditions. For example, most elements of a gas-turbine motor (GTM) are under high thermal load, where the thermal stresses arising in the members decrease the operating potential and degrade the economics of the motor. One can reduce the thermal stresses by using heat pipes or by including structural members operating on the heat-pipe principle.

The main problem in making such members is to choose a nontoxic, explosion-proof, noncorrosive heat-transfer agent capable of operating at temperatures of 300-500°C. To bring a GTM rapidly to operation from startup and provide flexibility of a system based on the heat-pipe principle, the agent should have a low melting temperature and remain in liquid form at a temperature of −50°C. From the viewpoint of structural strength at low mass, the vapor pressure should not exceed 5-7 kg/cm^2 at a temperature of 500°C.

The choice of agent is made difficult because this temperature range is not suited to heat-pipe operation, due to a limited choice of working liquids. Practically a single substance is used, mercury, which has a number of negative factors (toxicity, corrosive to steel structures, high specific gravity, low wettability of metal surfaces).

Analysis of existing high-temperature heat-transfer agents has shown that the above requirements can be satisfied by silicone oils. These are colorless liquids, insoluble in water, but soluble in aromatic hydrocarbons and alcohols [1, 2]. They are nontoxic, free of explosion hazard, and chemically inert. Their most valuable technical properties are the low freezing temperatures (−70 to −140°C), the low dependence of viscosity on temperature, the low saturated vapor pressure, the good stability towards irradiation, and the high dielectric properties. The thermophysical properties of silicones are shown in Table 1.

It was noted in [1, 2] that silicones can be used at temperatures up to 250-300°C in air, and it was recommended that they be used only in the form of nonboiling liquids, since silicone vapors are unstable and decompose rapidly. However, taking into account the specific operating conditions in heat pipes (absence of oxygen, increased pressure, and the short period where the agent is vaporized), one can propose the possible use of silicone oils as heat-transfer agents in heat pipes at temperatures up to 500°C. To check this experimentally, we tested nine silicone oils whose thermophysical properties are given in Table 1. The capabilities of the silicone oil agents were determined from the characteristics of heat pipes filled with the test liquids.

A. N. Tupolev Kazansk Aviation Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 39, No. 5, pp. 848-851, November, 1980. Original article submitted September 13, 1979.

 0022-0841/80/3905-1214$07.50