The character of the temperature distribution and Nusselt number curves is similar to that of the curves shown in Figs. 2 and 3, i.e., the effect of suction on temperature distribution and heat-exchange coefficient is the same for air and helium flow.

NOTATION

G	is the helium flow suctioned through tube wall, g/sec;
T	is the temperature, °K;
Nu	is the Nusselt number;
Re	is the Reynolds number;
l	is the length of porous tube, m;
T_x	is the temperature of inner surface of porous tube, °K;
T_0	is the temperature of inner surface of porous tube at input section, °K;
q	is the thermal flux density, W/m^2;
U	is the voltage, V;
I	is the current, A.

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CALCULATION OF CIRCULATION CHARACTERISTICS OF A TWO-PHASE THERMOSYPHON

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UDC 536.24

A new method is described for calculating the circulation characteristics of a thermosyphon with separate vapor and condensate channels. Calculation and experiment are compared.

At the present time, two-phase thermosyphons with separate vapor and condensate channels are widely used for heat transfer purposes [1]. The efficiency of thermosyphons of this sort is largely dependent on the circulation characteristics of the closed hydraulic circuit. The circulation of a boiling liquid in closed hydraulic circuits can be estimated in various ways [1-3].

In the present paper we propose a new method for solving this problem as applied to thermosyphons with separate vapor and condensate channels.

A schematic diagram of the circulation circuit is shown in Fig. 1. Here 1 and 2 are respectively the down- and up-pipes; 3 is the condenser. The vaporizer is located in the up-pipe. In the down-pipe there is only liquid; in the up-pipe there is a mixture of vapor and liquid. The equation of motion of the liquid and vapor in a closed circulation circuit (ignoring the compressibility of the components, energy losses on changing the interphase surface, and oscillations of vapor bubbles) can be brought to the form [4]:

$$g(\rho' - \rho'') L\varphi = \sum (\Delta P_{fr} + \Delta P_{acc}) \sin\beta. \qquad (1)$$

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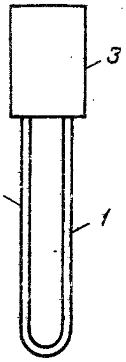

Fig. 1. Schematic diagram of thermosyphon. 1) Down-pipe; 2) vaporizer; 3) condenser.

Resistance in the down-branch and local resistances are neglected.

A special series of experiments were carried out in order to calculate the components of the pressure losses. The investigations were made on an experimental setup consisting of vertically mounted test thermosyphons and a device for heating part of the heat pipe by an ohmic heater. The test thermosyphons were constructed from brass pipes of wall thickness $\delta = 1$ mm.

The pressure was tapped at three measuring sections through 0.5-mm-diameter holes, the distance between which was varied depending on the length of the up-pipe. The pressure was measured by an inverted U-tube differential manometer connected by small-diameter hosepipes to the pressure takeoff units via compensating vessels.

The inertia of the takeoff system was thereby assured, and losses on friction could be measured with sufficient accuracy.

The liquid flow rate was determined in the down-branch by RS-type rotameters or by the method of calibrated resistances. In the experiments we also measured the power supplied to the vaporizer, the temperature of the liquid at the vaporizer inlet, the temperature of the mixture, and the pressure within the thermosyphon.

The technique of the thermal measurements, the sensors and the thermal control instruments were traditional.

The investigations were carried out in the following ranges of variation of the parameters: length of vaporizer L = 100, 150, 200 mm; diameter of vaporizer d = 4, 6, 8, 10 mm; length of condenser l_{cond} = 200 mm; diameter of condenser d_{cond} = 50 mm; pressure in circuit P = 1-4 bar. The heat-transfer medium was distilled water.

The results of the experiments were treated in the following sequence. The heater-induced relative vapor conversion in the mixture flow was determined via the formula:

$$X = \frac{i_{mix} - i'_{sat}}{r} = \frac{Q}{rG} . \tag{2}$$

The velocity of the mixture was then worked out:

$$W_{mix} = W_0 \left[X + \frac{\rho'}{\rho''} (1 - X) \right] , \tag{3}$$

where $W_0 = G/(\rho' f)$.

The results of the experiments were satisfactorily generalized by the empirical relationship proposed in [5]:

$$\xi = 0.04 / W_{mix}^{0.25} , \tag{4}$$

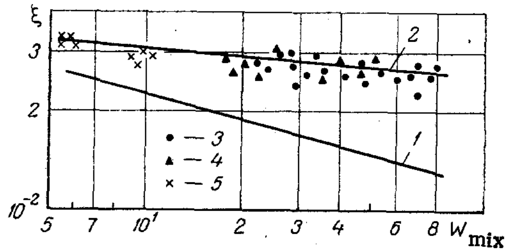

Fig. 2. Plots of ξ vs W_{mix}; 1) Summary graph [5]; 2) numerical calculation via homogeneous model, P = 2 bar; our experiments: 3) vaporizer diameter ϕ = 6 mm, q = (26.53-79.61) $\cdot 10^4$ W/m², L = 100 mm; 4) ϕ = 4 mm, q = (39.8-79.61) $\cdot 10^4$ W/m², L = 100 mm; 5) ϕ = 10 mm, q = 31.84 $\cdot$ 10^4; L = 100 mm, 200 mm.

where the frictional drag coefficient

$$\xi = \Delta P_{\text{fr}} \frac{d}{L} \frac{\rho W_{\text{mix}}^2}{2}. \tag{5}$$

Here $\rho = \rho'(W_0'/W_{\text{mix}})$.

Correlation of the experimental data in the approximation of the homogeneous model [6, 7] proved, however, to be more successful.

Indeed, in our case the hydrodynamic situation in the flow of the vapor–liquid mixture was characterized by a comparatively uniform distribution of the two phases over the cross section of the up-pipe. In these cases, instead of a truly two-phase flow, we are dealing with single-phase flow averaged parameters.

The frictional drag coefficient ξ is then determined from the equation for the turbulent isothermal motion of a single-phase liquid:

$$\xi = \frac{0.316}{Re^{0.25}}, \tag{6}$$

where

$$\text{Re} = Gd/f\mu, \qquad \mu = 1 \Bigg/ \left(\frac{X}{\mu''} + \frac{1-X}{\mu'} \right).$$

Figure 2 shows a comparison of our experimental results with the results of [5] and a numerical calculation based on the homogeneous model.

In this manner, the pressure drop due to friction can be represented in the form:

$$\Delta P_{\text{fr}} = \xi \frac{G^2}{f^2 2\rho'} \left[1 + X \left(\frac{\rho'}{\rho''} - 1 \right) \right] \frac{L}{d}. \tag{7}$$

The pressure drop due to acceleration of the flow can be represented as:

$$\Delta P_{\text{acc}} = \frac{G^2}{f^2} \left(\frac{1}{\rho} - \frac{1}{\rho'} \right). \tag{8}$$

Inserting (7) and (8) into (1), we obtain:

$$g(\rho' - \rho'') L\varphi = \left\{ \xi \frac{G^2}{f^2 2\rho'} \left[1 + X \left(\frac{\rho'}{\rho''} - 1 \right) \right] \frac{L}{d} + \frac{G^2}{f^2} \left(\frac{1}{\rho} - \frac{1}{\rho'} \right) \right\} \sin\beta. \tag{9}$$

The density of the vapor–liquid mixture can be determined through the true vapor content per unit volume:

$$\rho = \rho'(1 - \varphi) + \rho''\varphi. \tag{10}$$

The equation connecting the specific volume of a two-phase mixture with the heater-induced vapor content X and the true vapor content φ has the form:

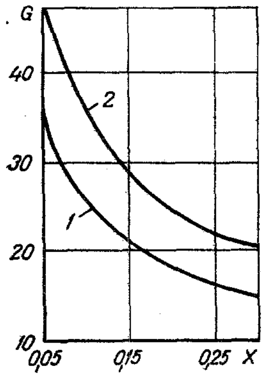

Fig. 3

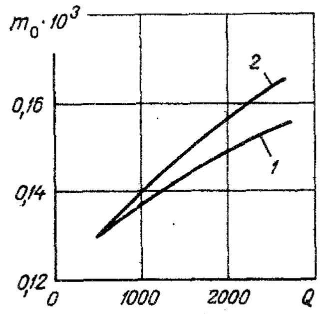

Fig. 4

Fig. 3. Dependence of G (kg/h) on X for 1) P = 1 bar; 2) 2 bar.

Fig. 4. Dependence of m_0 (kg) on Q (W) for 1) P = 1 bar; 2) 2 bar.

$$\frac{1}{\rho} = \frac{(1-X)^2}{\rho'(1-\varphi)} + \frac{X^2}{\rho'' X}. \tag{11}$$

Assuming that all the heat supplied goes on evaporation, we have:

$$Q = rGX. \tag{12}$$

The system of Eqs. (9)-(12) is nonlinear. It was solved numerically on an ES-1020 digital computer by the method of successive approximations. By way of example, we show in Fig. 3 the results of such a solution for a fixed thermosyphon geometry: L = 200 mm, d = 6 mm, β = 90°. The physical properties of the liquid and the vapor were taken to be constant and were determined through the saturation temperature at the given pressure.

The minimum quantity of liquid required to fill the thermosyphon m_0 can also be readily calculated utilizing the above results.

The quantity m_0 is determined by the sum of three components: by the mass of liquid m_1 in the vapor–liquid flow (the evaporation section – the up-pipe),

$$m_1 = \frac{G(1-X)L}{W_0}; \tag{13}$$

by the mass of liquid m_2 concentrated in the condensate film (m_2 was calculated in the approximation of the Nusselt model for a film condensate as done in [1]); and by the mass of liquid located in the down-pipe m_3:

$$m_3 = \rho' f (L + l_{\text{bend}}). \tag{14}$$

Simultaneous solution of the material balance equation and the system of Eqs. (9)-(12) gives the optimum value for a given thermosyphon geometry, pressure inside the cavity P, heat-transfer medium, and value of Q.

By way of example we show in Fig. 4 the dependence of m_0 on Q. Here the vaporizer length L = 100 mm, d = 4 mm, d_{cond} = 50 mm, l_{cond} = 200 mm, l_{bend} = 60 mm.

NOTATION

g	is the acceleration due to gravity;
ρ, ρ', ρ''	are the density of mixture, liquid, and vapor, respectively;
ΔP_{fr}, ΔP_{acc}	are the pressure losses on friction and acceleration;
φ	is the true vapor content per unit volume;
X	is the heater-induced relative vapor conversion per unit mass in mixture flow;

i_{mix}, i'_{sat} are the enthalpies of mixture and liquid on saturation line;
r is the heat of vapor formation;
G is the mass flow rate of mixture;
W_{mix}, W_0 are the velocities of mixture and of circulation;
ξ is the frictional drag coefficient;
μ, μ', μ'' are the dynamic coefficients of viscosity of mixture, liquid, and vapor, respectively;
L is the length of up-pipe;
β is the angle of slope of down-pipe;
d, d_{cond} are the diameters of thermosyphon and condenser;
l_{cond}, l_{bend} are the lengths of condenser and bend;
Q is the heat flux;
q is the heat flux density;
f is the cross-sectional area;
P is the pressure inside thermosyphon;
Re is the Reynolds number for mixture.

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