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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN THE EVAPORATOR OF A WATER HEAT PIPE

Yu. N. Filippov

UDC 536.423:536.248.2

The applicability of Gilmour's correlation equation in describing the heat transfer in the evaporator of a low-temperature heat pipe was experimentally confirmed.

An efficient heat-transfer device has lately been extensively studied and has already found practical application: It is the heat pipe, capable of axially transmitting a considerable amount of heat at a practically constant temperature.

To calculate the parameters of heat pipes and their thermal resistance, the heat-transfer coefficient upon vapor formation in the cells of the capillary structure of the heated section must be known. Only very limited data are available. The experimental results of [1] and [2] do not agree sufficiently with each other. The experiments in [1] were carried out on a special model-scale working element; those in [2], with a heat pipe. Therefore, a basic discrepancy between the data exists.

The present experimental work was carried out with an experimental heat pipe which is shown schematically in Fig. 1. The body was a copper pipe of $9.2 \cdot 10^{-2}$ m inner diameter, 1.17 m length, and $8 \cdot 10^{-3}$ m wall thickness. A capillary structure (wick) is tightly pressed against the inner surface. The wick was made of six layers of smooth woven brass netting (brass L80) with a mesh size of $0.14 \cdot 10^{-3}$ m. To ensure that the layers of the netting are tightly pressed against each other and against the wall, a steel ball of adequate size was drawn through, and then a layer of coarse-meshed smooth brass net 0.7 was inserted. All seven layers are held against the inner surface of the pipe by a special spring. The ends of the pipe were closed off by flanges through which thermocouples for measuring the steam temperature protruded. The working fluid, twice-distilled water, preliminarily degassed by boiling for more than 2 h, was poured into the pipe through special plugs in the flanges. Heat was supplied to the water in the heating section which was 0.2 m long and contained a resistance heater made of a strip of VZh-98 alloy. The strip was $0.3 \cdot 10^{-3}$ m thick, $4.5 \cdot 10^{-3}$ m wide, and 8 m long; its resistance was 8 Ω. The pipe was wrapped into a layer of electrically insulating silica fabric over which the heater strip was wound. Current to the heater was supplied from an RNO-250-10 regulating transformer. In view of the rising steam pressure, the power supplied to the pipe was limited to a maximum of 2.8 kW.

The condenser of 0.14 m length was cooled by circulating water supplied from a constant-level tank (cf. Fig. 2). The consumption of cooling water was measured by a volumetrically calibrated flow nozzle. The mean mixed inlet and outlet temperatures of the cooling water were measured by resistance thermometers. The electric power supplied was measured by a laboratory voltmeter and ammeter, instrument class 0.2.

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 33, No. 2, pp. 250-254, August, 1977. Original article submitted June 29, 1976.

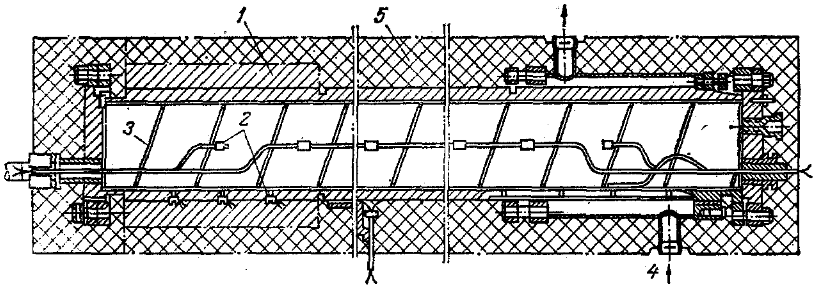

Fig. 1. Experimental heat pipe: 1) heater; 2) thermocouples; 3) spring; 4) cooling water; 5) thermal insulation.

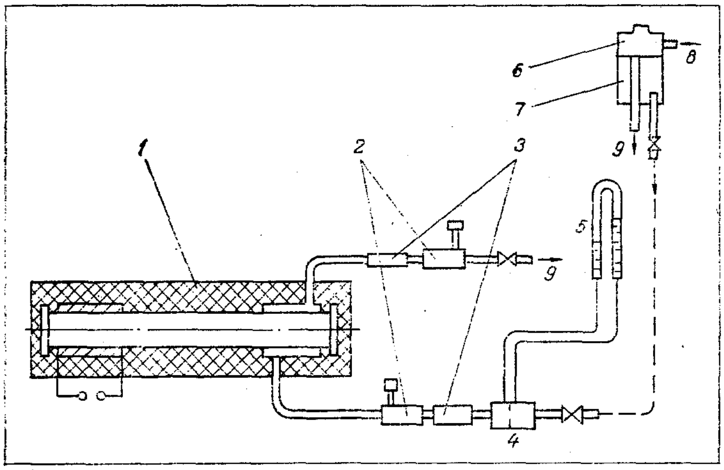

Fig. 2. Schematic diagram of the setup: 1) heat pipe; 2) resistance thermometers; 3) mixing chambers; 4) flow nozzle; 5) Π-shaped pressure gauge; 6) constant-level tank; 7) overflow pipe; 8) cooling water; 9) drains.

A total of 25 thermocouples were inserted into the wall of the heat pipe: 7 in the heating section, 13 in the heat-insulated section, and 5 in the condenser. The thermocouples were placed into milled grooves, and the hot junctions were caulked or secured by screws. Four thermocouple gauge heads were inserted through the flanges into the steam space for measuring the steam temperature in the heating section and in the heat-insulated zone. Chromel–Copel thermocouples were used, the diameter of the thermoelectrodes being $0.2 \cdot 10^{-3}$ m.

The production of the components and the assembly of the experimental heat pipe entailed stringent requirements as regards the cleanliness of all internal metal surfaces. The brass net was particularly thoroughly washed to ensure one of the basic conditions of the operation of the heat pipe — wettability of the wick material by the working fluid. The washing procedure was as follows: 4 days in a bath with benzene "Kalosha" (rubber solvent) the benzene being changed once every 24 h), drying, washing in boiling water for 1 h, drying in air, holding in a bath with alcohol for 2 h, and drying in air without blowing. Such washing ensured good wettability of the brass net by water. The inner surfaces of the pipe and other components were degreased and washed with ethyl alcohol.

The entire heat pipe, including the end faces and the cooling section, was heat-insulated. It was fixed on a stand in an almost horizontal position, with the heating section somewhat raised (making an angle of about 3° with the horizontal) so that the excess water was in the condenser, i.e., where the excess least affects the operation of the heat pipe. The air from its inner space was removed with the aid of steam: The heating section was lowered, the heating was switched on and the plug in the condenser flange opened, the steam expelled the air, and then the plug was screwed in again.

During the experiments the wall and steam temperatures were measured with a PP-63 class 0.05 potentiometer. The thermocouples operated with cold junctions. The measurements were carried out in the steady state.

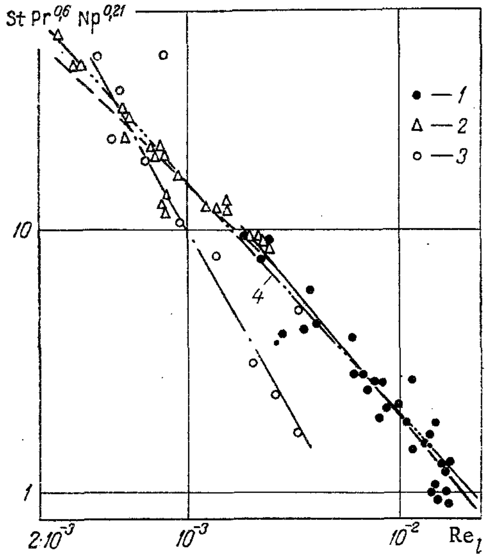

Fig. 3. Experimental data on heat transfer with boiling water in porous material: 1) experimental points $StPr^{0.6}Np^{0.21} = 0.02\ Re_l^{-0.99}$; 2) data of [1], $StPr^{0.6}Np^{0.21} = 0.072\ Re_l^{-0.77}$; 3) data of [2], $StPr^{0.6}Np^{0.2} = 0.00051\ Re_l^{-1.43}$; 4) $StPr^{0.6}Np^{0.21} = 0.035\ Re_l^{-0.88}$.

The heat pipe was easily started and operated without trouble within the range of power supplied, i.e., from 400 to 2800 W. The inlet heat flow in the evaporator varied between 0.71 and 4.8 W/cm²; the steam temperature, between 60.5 and 147°C.

The steam temperature was constant along the pipe under all test conditions. In the copper wall of the pipe the heat spread axially to a certain extent, amounting to a few percent of the energy supplied. The heat losses were preliminarily calibrated with the steam space evacuated.

Processing of the experimental data in dimensionless groups was carried out in conformity with the correlation equation proposed by Gilmour [3] and used by the authors of [1] and [2] in cases analogous to ours:

$$\left(\frac{\alpha}{c_pG}\right)^a\left(\frac{c_p\eta_l}{\lambda}\right)^b\left(\frac{\rho_l\sigma g}{p^2g_c}\right)^c=\varphi\left(\frac{d_hG}{\eta_l}\right)^d. \tag{1}$$

The left-hand side of the equation is Colburn's j-function.

Insofar as the liquid flow near the wall covered by a capillary structure is laminar, the ratio of the densities of the liquid and vapor, introduced by Gilmour for taking the turbulent transfer into account, has to be eliminated in the expression for the mass flow rate of the liquid, and the porosity of the capillary mass also has to be taken into account. Then

$$G = Q/F\varepsilon r. \tag{2}$$

The decisive dimension is the hydraulic diameter of the porous material d_h. The exponents in the criterial equation (1) for the Stanton and Prandtl numbers were taken as $a = 1$ and $b = 0.6$. The other parameters and the factor φ were calculated during the processing of the experimental data. The heat-transfer coefficient was calculated from the temperature difference between the steam and the inner surface of the evaporator of the heat pipe.

It should be pointed out that the heat-transfer coefficient thus calculated is in fact the heat-transfer coefficient that takes into account the thermal resistance of the wick, saturated with working fluid. It is impossible to determine the thermal conductivity of a wick layer with sufficient accuracy unless special experiments are set up. Rough calculations showed that the effect of the thermal resistance of the wick on the heat-transfer coefficient lies within the accuracy with which this parameter is determined. The defining temperature of the liquid was determined with the greatest accuracy by the authors of [1], where the thermal conductivity of the wick was known because it was practically the same as the thermal conductivity of water (high porosity of the packing, 0.963, and low thermal conductivity of the fibers, of the same order of magnitude as the thermal conductivity of water). In [2] the disposition of the thermocouples measuring the defining wall temperature was not indicated. In view of the fact that the wall and the wick were made of steel, the inaccuracy of the measurements could have been considerable and could have caused the discrepancy between the results and the data of [1].

It must also be borne in mind that the hydraulic diameters differ considerably in our work and in [1], being 200 and 82.4 μ, respectively, and the material of the wicks was also different. The hydraulic diameter of the wick in [2] was not stated, but it was probably close to 200 μ because a stainless-steel net of 100 mesh was used (the same standard mesh size as our brass net had).

In our experiments the temperatures of the wall and the steam were measured to within ±0.3°C, the temperature of the cooling water at the inlet and outlet of the condenser was measured with resistance thermometers to within ±0.2°C, the flow rate of the cooling water to within ±0.3%, the electric power supplied to within 1.5%, the relative error in the temperature difference between wall and steam was 7%, that of the heat flux 6%, and the error in the determination of the heat-transfer coefficient was 13%. The heat balance was established with an accuracy of ±5%.

In [1] the following exponents were obtained for the dimensionless groups of Eq. (1): pressure number Np, c = 0.21, Reynolds number for liquid d = −0.77, and proportionality factor φ = 0.072.

The experiments in [2] are described by the expression

$$\mathrm{St}\,\mathrm{Pr}^{0.6}\,\mathrm{Np}^{0.2} = 0.00051\,\mathrm{Re}_l^{-1.43}, \tag{3}$$

and the results obtained in the present work within the range of Reynolds numbers for liquid between $1.8 \cdot 10^{-3}$ and $1.7 \cdot 10^{-2}$ by the equation

$$\mathrm{St}\,\mathrm{Pr}^{0.6}\,\mathrm{Np}^{0.21} = 0.02\,\mathrm{Re}_l^{-0.99}. \tag{4}$$

The exponent of the pressure number was taken as 0.21, the same as in [1, 2] because when Re_l = const, the correlation $\mathrm{StPr}^{0.6} = f(1/\mathrm{Np})$ was found to be weak.

Bearing in mind the closeness of our experimental results to the data of Allingham and MacIntyre [1], an attempt was made to unify these data. The generalizing curve calculated by the least-squares method is described by the equation

$$\mathrm{St}\,\mathrm{Pr}^{0.6}\,\mathrm{Np}^{0.21} = 0.035\,\mathrm{Re}_l^{-0.88}. \tag{5}$$

The respective curves are plotted in Fig. 3.

An analysis of the results indicates that our data, obtained mainly with larger Reynolds numbers than the data of [1, 2], coincide with the data of [1] up to $\mathrm{Re}_l \sim 10^{-2}$.

The agreement between our results and the data of [1] indicates that the physical process occurring in the evaporator on the inner surface of the heat pipe and during boiling in the fibrous material on its outer surface are similar. Differences in the porosity and physical properties of the capillary materials did not cause any substantial differences in the results.

NOTATION

ε, porosity; d_h, hydraulic diamter, m; α, heat-transfer coefficient, $W/m^2 \cdot deg$; Q, heat flux, W; G, mass flow rate, $kg/m^2 \cdot sec$; r, evaporation heat, J/kg; c_p, specific heat, $J/kg \cdot deg$; η_l, liquid viscosity, $N \cdot sec/m^2$; λ, thermal conductivity, $W/m \cdot deg$; ρ, density, kg/m^3; σ, surface tension, N/m; F, heat-transfer surface area, m^2; p, pressure, N/m^2; g, acceleration due to gravity, m/sec^2; g_c, transition factor, $kg \cdot m/N \cdot sec^2$; $\mathrm{Re}_l = d_h G/\eta_l$, Reynolds number for liquid; $\mathrm{St} = \alpha/c_p G$, Stanton number; $\mathrm{Np} = \rho_l \sigma g/p^2 g_c$, pressure number; $\mathrm{Pr} = c_p \eta_l/\lambda$, Prandtl number.

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