HEAT TRANSFER IN A CENTRIFUGAL HEAT PIPE

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This paper presents experimental data and empirical relations describing the influence of heat flux density and vapor pressure on the heat transfer in a cylindrical heat pipe.

Cylindrical centrifugal heat pipes are used for cooling rotating bodies; e.g., the rotors of rotary electrical machinery [1, 2]. Therefore, it is important to investigate evaporation—condensation heat-transfer processes in closed systems operating in a centrifugal force field. It is known [3] that in open steady-state systems the heat-transfer intensity in the evaporation depends on the type of heat-transfer agent, the layer thickness, the heat flux density, and the vapor pressure in the system. In thin film evaporation the heat-transfer intensity is greater than in a large volume, but the influence of heat flux density and vapor pressure is less [4]. Nevertheless, in heat-transfer analysis in centrifugal heat pipes it is usual to start from the assumption either that the heat transfer in the heat pipe is determined only by the thickness of the condensate layer in the condensation zone [5, 6] or that the influence of the various factors on the heat transfer is the same [7, 8], which leads to known difficulties in explaining experimental results. In addition, the heat pipes investigated hitherto have been conical and coaxial and have been nearly full. In this paper we study heat transfer in a cylindrical centrifugal heat pipe with a minimum filling.

Experimental investigations of the influence of vapor pressure and heat flux density on heat transfer in a centrifugal heat pipe were carried out on the calorimetric equipment shown schematically in Fig. 1. The cylindrical centrifugal heat pipe is made of copper, has an inner diameter of $2.8 \cdot 10^{-2}$ m, an outer diameter of $4.0 \cdot 10^{-2}$ m, and evaporator-zone length of $18 \cdot 10^{-2}$ m, a transport-zone length of $17.6 \cdot 10^{-2}$ m, and a condensation zone length of $18 \cdot 10^{-2}$ m. Heat is supplied and removed convectively, the appropriate parts of the pipe being located in water heat exchangers. The water was circulated between the heating heat exchanger and the corresponding thermostat in a closed system. When required, type VPN-1.0 electrical boilers were used for supplementary heating of water in the thermostat. The temperature of the water entering and leaving the heat exchangers was measured by mercury thermometers with 0.1°C subdivisions, and the flow rate was determined by a volumetric method. The readings of thermocouples, located in the vapor space and attached to the pipe surface, were recorded using a specially calibrated type RAT-2 current recorder.

Before the experiments the pipe was rinsed with acetone, pumped down to a residual pressure of 10^{-3} torr, and charged with 15 cm^3 of degassed water (an average layer thickness of $3 \cdot 10^{-4}$ m).

The required vapor temperature in the tube and the heat flux density were provided basically by controlling the temperature of the water circulating in the heat exchangers. The tests were carried out at constant rotational speed of the pipe $\omega = 96\ sec^{-1}$. The values of the coefficients α and k were calculated from experimental data by conventional methods.

Figure 2 shows the test values of coefficients α and k for different values of heat flux density q and constant temperature $T_V = 62°C$ of the vapor in the pipe, corresponding to a saturation pressure of $P = 0.218$ bar. By keeping the rotational speed and the vapor temperature constant we could eliminate the influence of condensate layer thickness and saturation pressure on coefficients α and k.

From the data presented it can be seen that variation in heat flux density from $2 \cdot 10^4$ to $8 \cdot 10^4$ W/m² leads to a linear increase in the heat-transfer coefficient in the evaporator zones from $2 \cdot 10^3$ to $5.5 \cdot 10^3$ W/m²·deg. In the condensation zone the heat transfer increased

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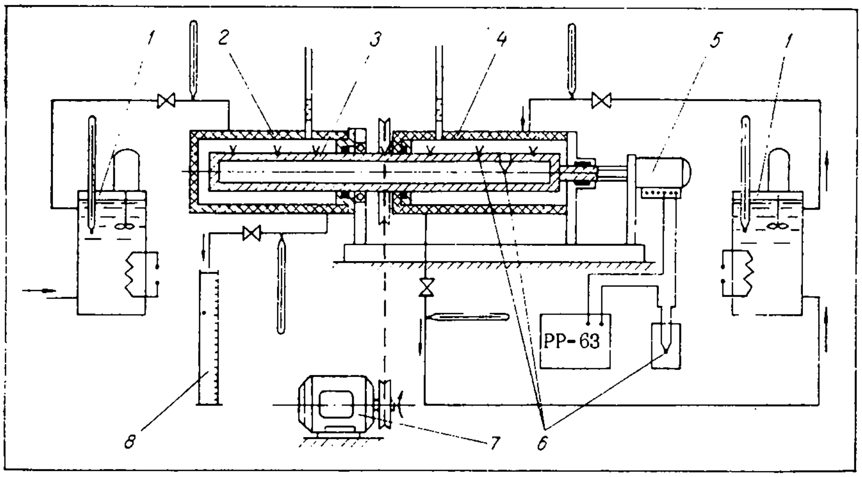

Fig. 1. Schematic of the calorimetric equipment: 1) thermostats; 2) cooler heat exchanger; 3) heat pipe; 4) heater heat exchanger; 5) thermocouple current recorder; 6) thermocouples; 7) drive motor; 8) measuring vessel.

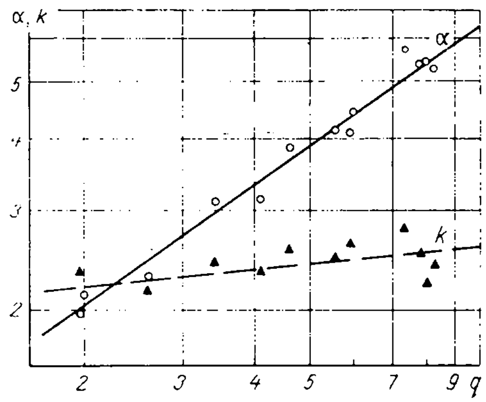

Fig. 2. The quantities α and k as a function of heat flux density at T_v = 62°C, P = 0.218 bar, and ω = 96 sec^{-1}. α, k, 10^{-3} W/m^2•deg; q, 10^{-4} W/m^2.

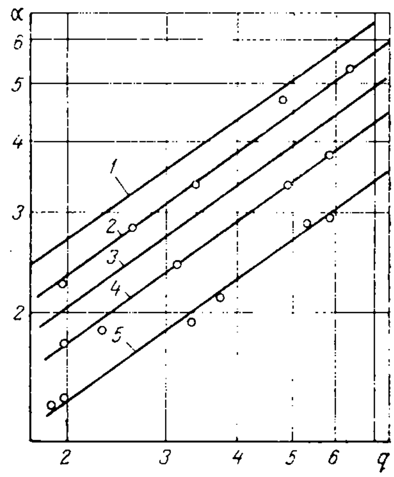

Fig. 3. The quantity α as a function of heat flux density q: 1) from Eq. (1); 2) at P = 0.312 bar; 3) 0.218; 4) 0.157; 5) 0.123 bar.

by approximately 8%. A linear dependence of the heat-transfer coefficient in the evaporator zone on heat flux density is clearly evident on the graph and can be approximated by the equation

$$\alpha = Aq^n, \quad (1)$$

in which A = 2.0, and the exponent n = 0.7. This result is noteworthy in the fact that the exponent 0.7 is typical for the analogous relation obtained in investigating the boiling of water in a large volume, and for P = 1.0 bar various authors have found values of A in the range 2.5 to 3.0 [3].

Alongside the straight line 1, which describes the intensity of boiling in a large volume with A = 2.6, Fig. 3 also shows values of α for other constant values of saturation pressure. In the entire pressure range investigated (from 0.312 to 0.123 bar) the straight lines approximating the relation $\alpha = f(q)$ remain practically parallel to line 1 and to each other. This means that in all cases the exponent of q in Eq. (1) remains constant and equal to 0.7.

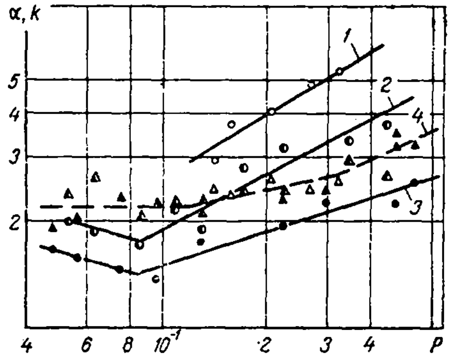

Fig. 4. The quantities α and k as a function of vapor pressure P at constant heat flux density: 1) α for $q = 6 \cdot 10^4$ W/m^2; 2) α for $q = 3.3 \cdot 10^4$ W/m^2; 3) α for $q = 1.9 \cdot 10^4$ W/m^2; 4) the corresponding heat-transfer coefficients in the condenser section k.

Figure 4 shows additional data illustrating the heat-transfer coefficient as a function of the vapor pressure in the tube. It can be seen from these data that an increase in vapor pressure in the tube at constant heat flux leads to an increase in the heat-transfer coefficient. The larger the heat flux density, the stronger the dependence is. Correspondingly, the value of A in Eq. (1) increases with increase of pressure.

Processing of the experimental data obtained shows that the dependence of heat-transfer coefficient in the evaporator zone on the vapor pressure and the heat flux density is described by the following empirical relation:

$$\alpha = 2.9 \sqrt[3]{1 + \lg P}\, q^{0.7}. \tag{2}$$

The formula is valid for the developed bubble-boiling regime at saturation pressures from 10^{-1} to 1.0 bar. One must bear in mind that the transition from boiling to evaporation from a surface is determined not only by the heat flux density, but also by the vapor pressure and the inertial acceleration. As the vapor pressure and the heating increase, the heat flux density at transition increases. This is confirmed by the data presented in Fig. 4. Curves 2 and 3, corresponding to heat flux densities of $3.3 \cdot 10^4$ and $1.9 \cdot 10^4$ W/m^2, reach a minimum and show a change in slope when the vapor pressure is reduced to $8.5 \cdot 10^{-2}$ bar. This is apparently due to transition from the bubble-boiling regime to the surface-evaporation regime. In steady-state systems at 1.0 bar this transition usually occurs for a heat flux density of $1 \cdot 10^4$–$2 \cdot 10^4$ W/m^2. It can be seen that the influence of a heat load $\eta = 13.7$ and a reduction of pressure to $8.5 \cdot 10^{-2}$ bar leads to an increase in the transitional heat flux density by a factor of 2-3.

In carrying out experiments like these one meets the known difficulties in determining the actual condensate layer thickness in the heat-exchanger sections, since it is practically impossible to ensure that the axis of rotation of the pipe is strictly horizontal. A direct indication of the layer thickness in the condensate zone can be provided by the heat-transfer coefficient at low heat flux and vapor pressure levels, when the effective thermal conductivity may be assumed to be the handbook thermal conductivity of water λ at the given temperature. It can be seen from Figs. 2 and 4 that the heat-transfer coefficient in the condenser zone is roughly 2200 W/m^2·deg at heat flux density of $2.0 \cdot 10^4$ W/m^2 and $P < 8.5 \cdot 10^{-2}$ bar. This corresponds to a thermal resistance of a water layer of thickness $\delta = \lambda/k = 0.66/2200 = 3 \cdot 10^{-4}$ m, which is roughly equal to the average layer thickness for the entire tube. However, because, according to the calculations of [9], the mean layer thickness with the pipe strictly horizontal and a near-optimal amount of liquid is 15-20% greater than the average for the entire tube and 30-35% less in the evaporator, one may conclude that in our case the tube has a slight slope toward the evaporator zone and that one can estimate the average layer thickness to be $3 \cdot 10^{-4}$–$3.5 \cdot 10^{-4}$ m for a minimum thickness in the dead-end part on the order of $2 \cdot 10^{-4}$ m. This means that in a heat pipe with a layer thickness $\delta > 2 \cdot 10^{-4}$–$3 \cdot 10^{-4}$ m the heat transfer during boiling follows laws identical to boiling in a large volume, as described by Eq. (2).

The data presented in Figs. 2 and 4 are evidence that the heat transfer through the liquid layer in the condenser zone is mainly due to conduction, but the effective thermal conductivity of the liquid increases with increase of the heat flux density. This may be due to an increase in Reynolds number for the liquid layer. There is a slight influence of vapor pressure on the value of k. Only three experimental points were obtained at $q = 1.9 \cdot 10^4$ W/m^2 and $P \approx 0.5$ bar (Fig. 4). The indicated growth of heat transfer with increase of vapor pressure may be due to an increase in the amount of liquid existing in the vapor state, and to a corresponding decrease in the condensate layer thickness.

Thus, the heat-transfer process in the conditions investigated is characterized mainly by the same laws which apply to open steady-state systems with an evaporation–condensation heat-exchange cycle. The well-known conclusion is confirmed that the magnitude of mass forces in developed boiling has practically no direct influence on the intensity of the evaporation and condensation processes [10, 11].

NOTATION

ω, frequency of rotation of the pipe; α, heat-transfer coefficient in the evaporator zone; k, heat-transfer coefficient in the condensation zone; q, heat flux density; T_v, vapor temperature in the pipe; P, saturation vapor pressure in the pipe; $\eta = g/g_0$, dimensionless acceleration (loading); g, inertial acceleration; g_0, acceleration due to gravity; λ, thermal conductivity of water; δ, thickness of condensate layer.

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