INTENSIFICATION OF HEAT TRANSFER IN THE BOILING OF WATER IN A MICROTHERMOSIPHON WITH A SUPPLY OF HEAT TO THE END

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Results are presented from an experimental study of heat transfer in the vaporizing zone of a microthermosiphon with capillary-pore structures present on the heat-emitting surface.

The use of thermosiphons to cool devices which emit heat locally – particularly semiconductor devices – requires high-rate heat transfer to a limited surface area of the vaporizing zone. It is well known that one method of intensifying heat transfer during boiling is covering the heat-emitting surface with lyophilic or lyophobic capillary-pore structures. However, there is no unambiguous and systematic experimental data on these topics in the literature.

Presented below are results of an experimental study of heat transfer in the boiling of distilled water in a vertical microthermsiphon with a supply of heat to the end, which was covered with different combinations of brass grids and perforated teflon sheets. The "micro" referring to the thermosiphon indicates the small size of the heat-emitting surface, comparable to the mean separation diameter of vapor bubbles of the working liquid.

The experimental method and conditions were described in [1]. The heat-emitting surface of the copper end of the microthermosiphon had a roughness of 6.3 μm and a diameter of $4 \cdot 10^{-3}$ m. Two sizes of brass grids were used: a grid of $0.05 \cdot 10^{-3}$-m-diameter wire with an inside mesh of $0.08 \cdot 10^{-3}$ m (grid 0.08); a grid of $0.14 \cdot 10^{-3}$-m-diameter wire with an inside mesh of $0.28 \cdot 10^{-3}$ m (grid 0.28). The grids were held in acetone and washed in distilled water before use. The teflon sheets had the following parameters: $\delta = 0.1 \cdot 10^{-3}$ m, n = 7, d = $0.8 \cdot 10^{-3}$ m (sheet No. 1); $\delta = 0.6 \cdot 10^{-3}$ m, n = 7, d = $1 \cdot 10^{-3}$ m (No. 2); $\delta = 0.6 \cdot 10^{-3}$ m, n = 3, d = $1 \cdot 10^{-3}$ m (No. 3); $\delta = 0.1 \cdot 10^{-3}$ m, n = 4, d = $0.8 \cdot 10^{-3}$ m (No. 4). The grids and sheets were pressed tightly to the heat-emitting surface by a special star-shaped insert with three copper arms $0.4 \cdot 10^{-3}$ m thick. The insert was pressed to the heat-emitting surface by a spring. The height of the liquid layer during boiling was kept within the range $(5\text{-}10) \cdot 10^{-3}$ m.

Figure 1 shows the results of the tests, involving boiling water under atmospheric pressure on the end surface covered with a brass grid. It is apparent from the figure that the presence of the grid significantly intensifies heat transfer compared to boiling on a free end-surface, especially in the region of relatively small ΔT. The mesh sizes and the location of the grids relative to each other in a multilayered packet have no effect on heat transfer. This is in accord with the representations in [2] on the role of the geometry of the capillary-pore structure, as well as with the data in [3].

Figure 2 shows the results of tests of boiling on an end covered with the perforated teflon sheets.

The pronounced lyophobic properties of teflon facilitate formation of the vapor phase in the boundary layer of the liquid and thus help increase heat transfer [4]. The data in Fig. 2 shows that this takes place when "steaming" of the heat-transfer surface is avoided. Thus, relatively thin ($0.1 \cdot 10^{-3}$ m) sheet No. 1 was elastic enough to form a cavity with the heat-emitting end to allow circulation of the liquid, thereby increasing the heat-transfer rate in the region of small ΔT. Heat transfer of the same intensity was recorded in a special series of tests using a star-shaped teflon insert with five arms $0.5 \cdot 10^{-3}$ m thick and $5 \cdot 10^{-3}$ m high placed on the end without grids. Sheets $0.6 \cdot 10^{-3}$ m thick, being more rigid, were laid on the heat-emitting surface without an intervening gap. Heat transfer was thus reduced in this case due to "steaming" of the surface as a result of there being no inflow of liquid. Steaming also led to a low heat-transfer rate for thin sheet No. 4, since the

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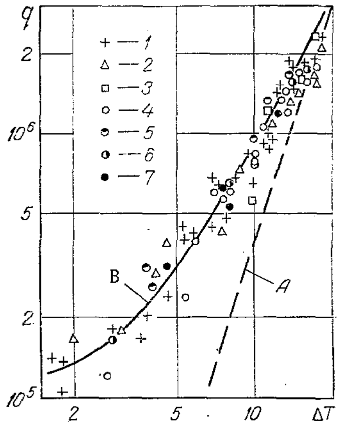

Fig. 1

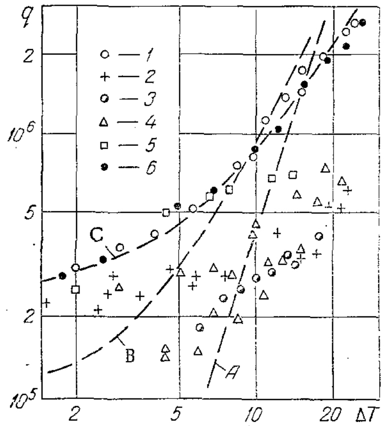

Fig. 2

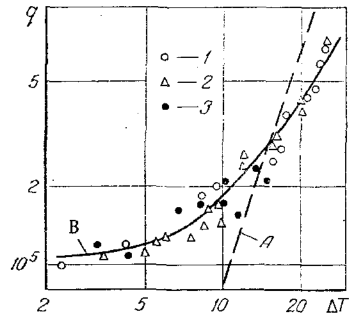

Fig. 3

Fig. 1. Dependence of q (W/m^2) on ΔT (°K) with brass grids on the surface and P=1 bar; 1) grid 0.08; 2) 0.08+0.28 (henceforth, the first grid is in contact with the end and the second grid lies on top of the first); 3) 0.08+0.28+0.28; 4) 0.28; 5) 0.28+0.08; 6) 0.28+0.08+0.08; 7) 0.28+0.28; A) no grids on end, calculation from [1]; B) calculation from Eq. (2).

Fig. 2. Dependence of q (W/m^2) on ΔT (°K) with perforated teflon sheets on the surface and P=1 bar: 1) sheet No. 1; 2) No. 2; 3) No. 3; 4) No. 4; 5) 0.08+No. 2; 6) star-shaped teflon insert; A) free end, calculation from [1]; B) from Eq. (2); C) from Eq. (1).

Fig. 3. Dependence of q (W/m^2) on ΔT (°K) with capillary structures on the surface and P=0.3 bar: 1) 0.08; 2) 0.08+No. 2; 3) No. 3; A) free end, calculation from [1]; B) calculation from Eq. (2).

fraction of its free cross section, equal to 0.16, was insufficient to permit free outflow of vapor from the end surface.

The test data for sheet No. 1 and the teflon insert at ΔT =1.5-25°K is approximated by the equation

$$q = 16\cdot 10^3 (\Delta T^{1,6} + 14.4). \tag{1}$$

Heat transfer is intensified considerably in the region of small ΔT with the placement of grids and sheets on the heat-emitting surface and for boiling at a pressure below atmospheric (Fig. 3). It can be seen from the figure that, in this case, the difference in heat transfer for different capillary-pore structures degenerates. This has to do with the features of boiling and a reduction in the effect of surface conditions on vaporization at low pressures [4].

The following heat-transfer equation was obtained as a result of analysis of the test data for the interval P=0.25-1 bar with brass grids on the surface

$$q = 8\cdot 10^3 (P^{1,9}\,\Delta T^2 + 12.5), \tag{2}$$

which corresponds to an interval ΔT =1.5-25°K at P=1 bar and ΔT =3-30°K at P=0.25 bar.

NOTATION

δ, thickness; n, number of holes; d, diameter of holes; $\Delta T = T_{st} - T_s$, temperature head; q, heat flux; P, pressure; T, temperature. Indices: st, heat-emitting surface; s, saturation.

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HEAT EXCHANGE ALONG THE INITIAL SEGMENT IN A HEAT EXCHANGER WITH A HELICAL FLOW

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The characteristics of local heat transfer along the initial segment and along the segment with stabilized air flow in longitudinal flow past a bundle of coiled oval tubes are established

The characteristics of heat transfer in heat exchangers with longitudinal flow past a bundle of coiled oval tubes were examined in a number of papers [1-4], wherein it was shown that the observed increase in heat transfer is explained by the properties of helical flows in channels with a complicated shape. An equation was proposed in [2, 3] that describes the heat-transfer process in a stabilized turbulent flow in the range of numbers $Fr_m = 232-2440$:

$$\overline{Nu} = 0.023 Re^{0.8} Pr^{0.4} (1 + 3.6 Fr_m^{-0.357}) (T_w/T_f)^{-0.55}, \tag{1}$$

where

$$Re = u_{mm} d_e \rho/\mu, \tag{2}$$

$$Fr_m = S^2/dd_e, \tag{3}$$

which differs from the equation for circular pipes [5]

$$Nu = 0.023 Re^{0.8} Pr^{0.4} (T_w/T_f)^{-0.55} \tag{4}$$

by a factor that depends on the number Fr_m. For Fr_m numbers less than 100, the heat-transfer coefficient increases to a larger extent than follows from Eq. (1). Thus, for values of the number $Fr_m = 64$, heat transfer in a bundle of coiled tubes can be described by the function [1]

$$\overline{Nu} = 0.0521 Re^{0.8} Pr^{0.4} (T_w/T_f)^{-0.55}. \tag{5}$$

However, Eqs. (1) and (5) describe only the locally averaged heat transfer, since along the segment with stabilized flow, the experimental data are observed to separate within a range of approximately ±15%, as noted in [1-3], which must be taken into account in the heat-transfer law. In addition, it is necessary to establish the law governing the change in the heat-transfer coefficient along the initial segment of the flow. This paper is concerned with solving these problems.

The heat transfer was investigated using a generally accepted technique on experimental setup described in [1] with air as the heat transfer agent. The heat exchangers consisted of 37 coiled tubes of length 500 and 750 mm and the porosity for the heat-transfer agent was $m = 0.527-0.544$. The tubes were made of Kh18N10T steel. The tubes were heated by passing an alternating electric current through them from a OSU-100 transformer, controlled by AOMK-180 autotransformer. Chromel–Alumel thermocouples, welded to the interior of the central tube in the bundle at five sections along its length, were used to measure the temperature of the walls of the tubes. The densely packed lattice of the bundle had an ordered structure with the coiled tubes touching one another at the output part of the bundle along the long axis of the oval tube profile. The flow of the heat-transfer agent entered the bundle axisymmetrically. The measuring system permitted determining the Nusselt number with a limiting relative error of ± 7% with the following range of parameters: $S/d = 6.2-34$; $Fr_m = 64-2440$; $Re = 2 \cdot 10^3 - 4 \cdot 10^4$; $T_w/T_f = 1.0-1.73$; and $x/d_e = 3.75-103$.

The nature of the change in the heat-transfer coefficient along the length of the bundle of coiled tubes can be seen in Fig. 1, where the results of the experimental investigation of heat transfer in the bundle with a number $Fr_m = 924$ is shown as the functional relation:

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