HEAT EXCHANGE IN THE HEAT-SUPPLY ZONE OF TWO-PHASE THERMOSIPHONS FOR SMALL DEGREES OF FILLING

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The intensity of the heat exchange in evaporation and boiling in the heat-transmitting element is investigated as a function of the geometrical, physical, and operating parameters.

Two-phase thermosiphons, which are closed cavities partially filled with a heat carrier, the heat transmission in which occurs due to simultaneously occurring boiling and condensation processes, are used in various areas of technology [1-4].

Analysis [2-9] has shown that in addition to the well-known parameters (q, p, l, d_{in}), which affect the heat exchange in the heat-supply zone and the general heat-transmitting capabilities of the thermosiphon, the degree of volume filling of the heat-carrying element Ω, or more accurately the degree of volume filling of the heat-supply zone ε, is particularly important. It was established in [4, 6] that the overall heat-transmitting capacity is independent of the degree of filling. Nevertheless the authors, while noting the features of heat exchange for $\varepsilon = 1$-40%, did not analyze the process in the heat-supply zone and obtained no general relations. In [3] the effect of the degree of filling on the heat exchange was also noted. The author expressed the opinion that for small values of ε the heat-transfer coefficients can be calculated from the well-known relations for the heat exchange in films, which does not agree with visual observations [5]. The results of the generalization [2] contradict their visual observations and do not represent the physics of the heat-exchange process on the heating side for small values of the degree of filling. In [7] the possibility of a closed two-phase thermosiphon operating in two modes (a discharging film when $\varepsilon < 50\%$, and a two-phase mixture when $\varepsilon \gtrsim 50\%$) was noted, and the suggestion was also put forward that in the discharging-film mode a strictly vertical position of the longitudinal axis of the thermosiphon is required, since any departure from this position leads to disturbance of the uniformity of the film of liquid flowing over the internal surface.

In this paper we present the results of an experimental investigation of heat exchange in the heat-supply zone of two-phase copper thermosiphons with d_{in} = 6, 8, 10, 12, 18, and 24 mm, and length 250-700 mm in the range of heat-flux densities $q = (0.16$-$50) \cdot 10^3$ W/m^2, slopes of the heat-transmitting element $\Psi = 5$-$90°$ to the plane of the horizontal, for degrees of volume filling of the supply zone $\varepsilon = 1$-40%. In all the thermosiphons investigated either Freon-11, 113, or 142 was used as the working liquid, as well as ethanol and acetone. The internal surface of the thermosiphons was processed to the eighth-to-ninth classes in accordance with GOST 2789-59, decreased with caustic soda, and then washed with distilled water and dried. The thermosiphon was filled with the working liquid on a special charging bench, which ensured the purity and deaeration of the heat carrier, and also evacuated the cavity of the element to a residual pressure $p = 10^{-3}$ torr and filled the thermosiphon with a measured quantity of liquid with an accuracy of ±5%.

The investigations were made on experimental equipment provided with air and water cooling. The velocity of the cooling air was varied in the range $w \approx 1$-60 m/sec, and that of the water in the range $w = (2$-$9) \cdot 10^{-2}$ m/sec. The length of the cooling zone in all the experiments was 200 mm. The pressure in the cavity of the thermosiphon was varied in the range 0.3-12 bar.

The heat was supplied by a resistive heater. The length of the heat-supply zone l = 50-350 mm. The power supplied was measured with a type D57 wattmeter (accuracy class 0.1). The temperature distribution in the walls of the thermosiphon was found using copper – Constantan thermocouples (wire diameter 0.16 mm), stamped into longitudinal grooves on the external surface. Depending on the geometrical dimensions of the thermosiphons and the lengths of the heat-supply zones, the thermocouples were arranged two or three together in sections spaced 10-15 mm from one another. The saturation temperatures were measured with thermocouples introduced into the internal cavity of the element by means of sleeves soldered into the end plugs. The

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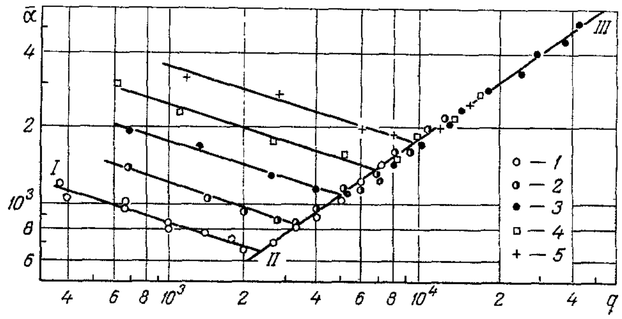

Fig. 1. Heat-transfer coefficient as a function of the heat-flux density, the internal diameter, and the length of the heat-supply zone (Freon-11, Ω = 3%, Ψ = 90°): I-II) evaporation region; II-III) boiling region; 1) d_{in} = 24 mm, l = 200 mm; 2) l = 100 mm; 3) l = 50 mm; 4) d_{in} = 12 mm, l = 100 mm; 5) d_{in} = 6 mm, l = 100 mm.

emf of the thermocouples was measured with an Shch68000 digital electronic voltmeter–ammeter (accuracy class 0.01). From the temperature distributions obtained along the length and perimeter of the heat-supply zone, and the saturation temperature for each mode of operation we determined the mean heat-transfer coefficients $\bar{\alpha}$.

Figure 1 shows a graph of $\bar{\alpha} = f(q, l, d_{in})$, which is typical for all the working liquids in the range p = 0.3-12 bar.

Heat transfer in the thermosiphon occurs by evaporation, mass transfer, and condensation. To explain the particular features of the hydrodynamics, we carried out visual observations on glass thermosiphons of length 0.5 m and d_{in} = 6, 12, and 24 mm. For q = 160-10,000 W/m² we first observed evaporation of the liquid from the surface of separation of the phases. The condensate formed under the action of the gravitational forces enters the heat-supply zone. In the adiabatic and condensation zones there is a stable film of condensate covering the whole surface. At a distance of $(2\text{-}3)d_{in}$ from the entrance to the heat-supply zone the film disintegrates into separate streams and strands. When the thermal load is increased a sharp contraction of the film occurs into individual streams directly at the entrance to the heat-supply zone. Intense evaporation of the heat carrier from the streams running into pools was observed. When a certain value of the temperature of the walls and the saturation pressure was reached vaporization centers appeared in the streams. The process of vapor generation when boiling occurs in the streams was accompanied by intense spraying of the heat carrier on the uncovered parts both in the immediate vicinity of the places where the bubbles collapse, and on the opposite wall of the tube. A reduction in the temperature of the tube walls compared with the temperature of the wall at the instant preceding the beginning of boiling in the streams was observed. In the developed boiling mode of the heat carrier in the streams the whole of the uncovered surface of the heat-supply zone was uniformly coated with evaporating drops of liquid, whose diameter was 0.5-1.5 mm. When there was a sudden discharge of the load the process occurred in reverse order except that when a film instability occurred at the entrance to the heat-supply zone it moved to the bottom of the element and moistened the whole surface of the heat-supply zone.

Analysis of the experimental data obtained from the point of view of the wetting of the surface of the heat supply using the relations given in [10-12] showed that over the whole range of heat-flux densities q = $(0.16\text{-}50) \cdot 10^3$ W/m² and Reynolds numbers of the film Re = 1-90, the film is unstable, breaks up into separate streams, and dry spots appear.

Hence, there are two regions of heat exchange depending on the heat-flux density in the heat-supply zone of two-phase thermosiphons. The region I-II where evaporation of the heat carrier from the streams occurs, where the heat-supply zone and is approximated by a curve of the form $\bar{\alpha} \sim q^{-0.33}$. The increase in the heat exchange as l and d_{in} decrease is due to the increase in the fraction of the surface which is wetted by the streams compared with the dry surface.

The region II-III is the region where boiling of the heat carrier in the streams and evaporation of drops of liquid ejected on the exposed surface when the drops break up, occurs. The process in this region is self-similar with respect to l and d_{in} and is described by a relation of the form $\bar{\alpha} \sim q^{0.7}$.

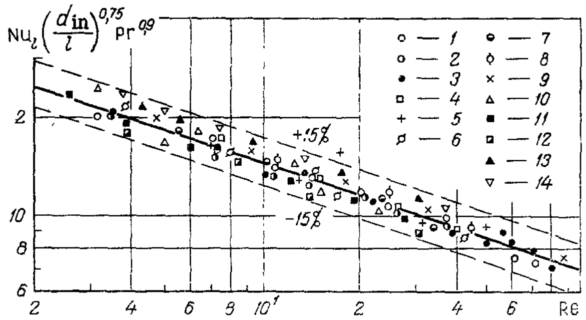

Fig. 2

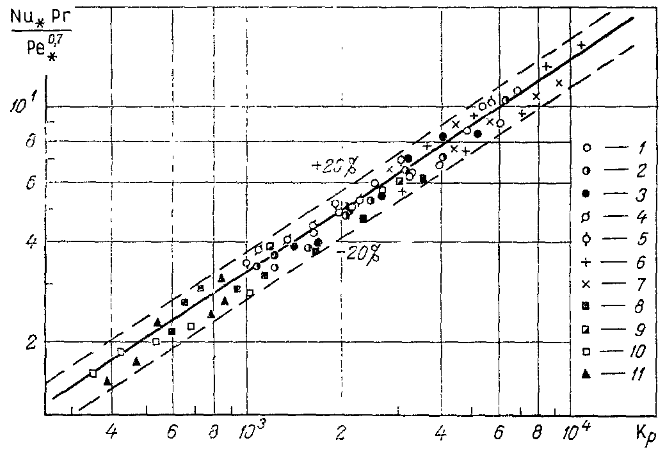

Fig. 3

Fig. 2. Heat transfer in the evaporation region of vertical two-phase thermosiphons: Freon-11: 1) d_{in} = 24 mm, l = 200 mm; 2) 24 and 100; 3) 24 and 50; 4) 12 and 100; 5) 6 and 200; Freon-113; 6) d_{in} = 18 mm, l = 100 mm; 7) 12 and 50; 8) 6 and 350; Freon-142: 9) d_{in} = 18 mm, l = 200 mm; 10) 10 and 50; acetone: 11) d_{in} = 7 mm, l = 350 mm; 12) 12 and 100; ethanol: 13) d_{in} = 12 mm, l = 200 mm; 14) 18 and 50.

Fig. 3. Heat exchange in the boiling region of vertical two-phase thermosiphons: Freon-11: 1) ε = 5%; 2) 20; 3) 40; Freon-113: 4) ε = 30%; 5) 15; Freon-142: 6) ε = 10%; 7) 5; ethanol: 8) ε = 20%; 9) 10; acetone: 10) ε = 1%; 11) 20.

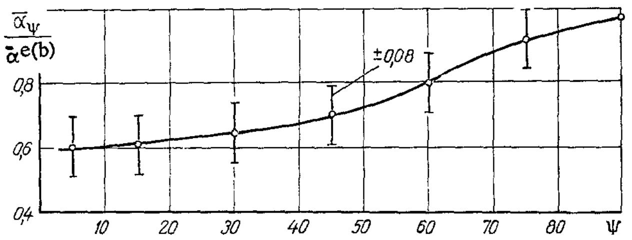

Fig. 4. Heat-transfer coefficient as a function of the angle of inclination of the heat-transmitting element.

The high heat exchange and also the isothermal properties of the heating surface in regions I-II and II-III are due to evaporation and boiling of the heat carrier in the streams, freely varying its position on the heat-exchange surface, and also the high thermal conductivity of the material of the walls of the element. A similar conclusion was reached in [13].

As a result of the generalization of the experimental data on the heat exchange in vertical two-phase thermosiphons under evaporation conditions using Freon-11, 113, and 142, as well as acetone and ethanol in the range q = (0.16-15) · 10^3 W/m^2, p = 0.3-3.5 bar, d_{in} = 6-24 mm, and l = 50-350 mm we obtained the following similarity equation:

$$\mathrm{Nu}_l = 3050\mathrm{Re}^{-0.33}\,\mathrm{Pr}^{-0.9}\,(l/d_{\mathrm{in}})^{0.75}, \tag{1}$$

which describes the experimental data with an error of ±15% (Fig. 2).

A generalization of the experimental data on heat exchange in the boiling mode using Freon-11, 113, and 142, and also acetone and ethanol in the range q = (2-50) · 10^3 W/m^2, p = 0.8-12 bar, d_{in} = 6-24 mm, l = 50-350 mm, and Ψ = 90° is shown in Fig. 3. The similarity equation

$$\mathrm{Nu}_* = 0.0096\mathrm{Pe}_*^{0.7}\mathrm{Pr}^{-1.0}K_p^{0.6} \tag{2}$$

describes the experimental points with an error of ±20%.

Figure 4 shows the heat transfer coefficient as a function of the orientation of the heat-transmitting element in $\{\bar{\alpha}_\Psi/\bar{\alpha},\ \Psi\}$ coordinates, from which it is seen that the heat exchange decreases as Ψ decreases. This can be explained by the fact that when the angle of inclination is reduced the fraction of the surface not sprinkled with streams, and the fraction of the heat flux transferred to the streams by thermal conduction, increase, leading to an increase in the thermal resistance in the heat-supply zone and to a reduction in the coefficient $\bar{\alpha}_\Psi$. We noted an increase in the nonisothermal nature of the heat-exchange surface as Ψ was reduced. The difference in the readings of the thermocouples on the upper and lower generating lines for small values of Ψ = 5-45° reached 4-5°C.

The heat transfer when the orientation of the heat-transmitting element is changed can be represented by the equation

$$\bar{\alpha}_\Psi = \bar{\alpha}\, 0.61 e^{0.5 \sin^4 \Psi} \tag{3}$$

(the continuous curves in Fig. 4), which describes the experimental data over the whole range of variation of the geometrical, physical, and operational parameters investigated with a confidence interval of ±0.08 with a confidence coefficient of 0.9.

NOTATION

$\bar{\alpha}$	is the mean heat-transfer coefficient for a vertical position of the heat-supply zone;
q	is the density of the heat flux supplied;
p	is the pressure;
ρ', ρ''	are the density of the liquid and the vapor;
σ, μ, ν, λ, a	are the surface tension, dynamic and kinematic viscosity, thermal conductivity, and thermal diffusivity;
l	is the length of the supply zone;
d_{in}	is the internal diameter;
Ω, ε	are the degree of volume filling referred to the volume of the thermosiphon and the volume of the heat-supply zone;
r	is the latent heat of vaporization;
$\Gamma = ql/r$	is the mass density of irrigation;
Ψ	is the angle of inclination;
$Nu = (\bar{\alpha}^e/\lambda)l$, $Nu_* = (\bar{\alpha}^b/\lambda)\sqrt{\sigma/g(\rho'-\rho'')}$	are the Nusselt numbers for evaporation and boiling;
$Re = 4\Gamma/\mu$	is the Reynolds number for the film;
$Pe_* = Re_* Pr$	is the Peclet number;
$Re_* = (q/r\rho''\nu)[\sigma/g(\rho'-\rho'')]^{1/2}$	is the Reynolds number for boiling;
$Pr = \nu/a$	is the Prandtl number; and
$K_p = p/\sqrt{\sigma g(\rho'-\rho'')}$	is the pressure number.

The indices are as follows:

e is the evaporation;
b is the boiling;
and Ψ is the arbitrary position of the supply zone.

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