HEAT-EXCHANGE PROCESSES IN THE HEAT-SUPPLY ZONES OF TWO-PHASE THERMOSIPHONS OPERATING ON FREONS 11, 113, AND 142 AND ON WATER AND ETHANOL

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The main heat-exchange laws in the heat-supply zone of a two-phase thermosiphon are established. The heat-exchange intensity during boiling of the heat carrier in the heat-transfer element is determined as a function of the geometrical and operating parameters.

Two-phase thermosiphons, which are simple, reliable, and rather highly efficient heat-transfer devices, can be employed successfully in various branches of engineering [1-5]. A two-phase thermosiphon consists of a closed cavity partially filled with a heat carrier having a phase change in the heat-transfer process. The high efficiency of such a device is explained by the utilization of the latent heat of vaporization. There are well-known reports [6-9] devoted to the investigation of the heat-exchange processes in the heat-supply zones of two-phase thermosiphons. But the results presented in these papers have a particular character, and sometimes they also contradict each other significantly. An attempt to generalize the experimental data was made in [10]. Unfortunately, the generalizing functions obtained are constructed on a formal basis, do not reflect the physics of the processes taking place, and do not allow for the influence of the length of the heat-supply zone.

In the present paper the results of an experimental study of the heat-exchange laws in the heat-supply zone of a two-phase thermosiphon as a function of the geometrical, physical, and operating parameters are presented. The studies were carried out on copper thermosiphons with a diameter $d_{in}$ = 6-24 mm, a length of 250-700 mm, and filled to 20-50% of the internal volume of the thermosiphon with the working liquid. The tests were conducted in the range of heat-flux densities $q = (0.3\text{-}110) \cdot 10^3$ W/m² at angles of inclination $\Psi$ = 5-90°. Freons 11, 113, and 142 served as the working liquids in all the thermosiphons investigated. Distilled water and ethanol

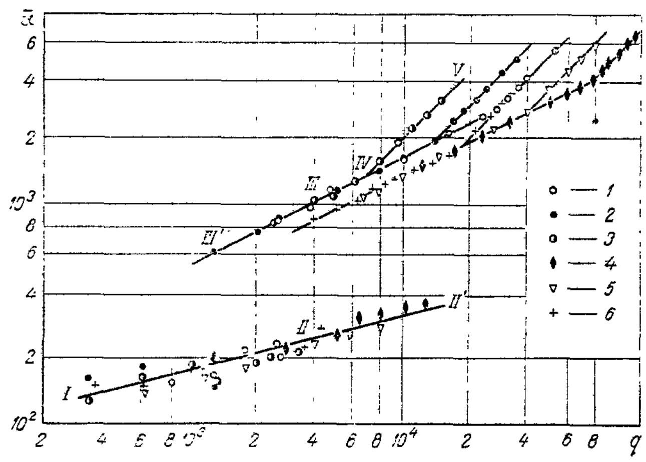


Fig. 1. Dependence of heat-transfer coefficient $\bar{\alpha}$, W/m² · deg, on the heat-flux density q, W/m²: Freon 113, $\Omega$ = 30%, $\Psi$ = 90°, $d_{in}$ = 24 mm; I-II') region of free-convective heat exchange and undeveloped boiling; III'-IV) developed bubble boiling; IV-V) region of hydrodynamic stabilization of the process of developed bubble boiling. Air cooling: 1) $l$ = 50 mm; 2) 100 mm; 3) 200 mm; water cooling: 4) 50 mm; 5) 100 mm; 6) 200 mm.

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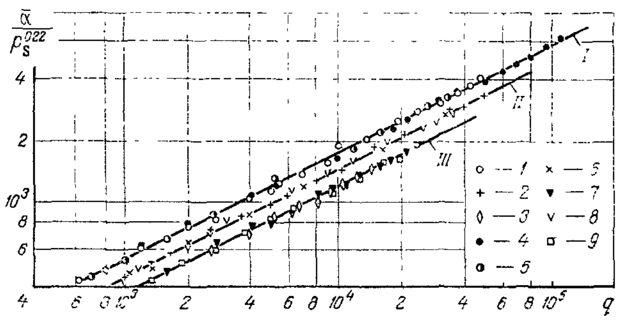

Fig. 2. Dependence of coefficient $\bar{\alpha}$, W/m$^2$ · deg, on heat-flux density q, W/m$^2$, and saturation pressure $p_s$, bar: I) $d_{in}$ = 24 mm; II) 12 mm; III) 6 mm. Air cooling: 1) Freon 11, Ω = 30%, $l$ = 50 mm; 2) Ω = 20%, $l$ = 100 mm; 3) Ω = 25%, $l$ = 200 mm; 4) Freon 113, Ω = 50%, $l$ = 100 mm; 5) Freon 142, Ω = 20%, $l$ = 50 mm. Water cooling: 6) Freon 113, Ω = 20%, $l$ = 100 mm; 7) Ω = 33%, $l$ = 200 mm; 8) Freon 142, Ω = 50%, $l$ = 50 mm; 9) Ω = 20%, $l$ = 200 mm.

were also used in addition to the heat carriers enumerated above in the heat-transfer element with $d_{in}$ = 24 mm. The inner surfaces of the thermosiphons were polished to eliminate the effect of roughness on the heat-exchange laws. The thermosiphons were filled with the working liquid on a special stand which assured the purity and deaeration of the working liquid, as well as the evacuation of the cavity of the element to a residual pressure $p = 10^{-3}$ torr and filling with a measured amount of working liquid with an accuracy of ±5%.

The investigations were performed on an experimental installation with air and water cooling. The velocity of the cooling air was varied in the range of w = 1-60 m/sec and that of the water in the range of w = (2-9) · $10^{-2}$ m/sec. In all the experiments the length of the cooling zone was 200 mm. The pressure in the cavity of the thermosiphon was varied from 0.025 to 15 bar. The heat was supplied with a resistance heater. The load was varied with a voltage regulator connected to the ac power line (220 V, 50 Hz) through a voltage stabilizer. The power supplied was measured with a type D57 wattmeter (accuracy class 0.1). The temperature distribution in the wall of a thermosiphon was measured with copper — Constantan thermocouples (leads 0.16 mm in diameter) caulked in longitudinal channels on the outer surface. Depending on the geometrical dimensions of the thermosiphons and the lengths of the heat-supply zones, the thermocouples were distributed with two to three in a cross section, separated from each other by a distance of 10-15 mm. The saturation temperature was measured with thermocouples introduced into the inner cavity of the element through sleeves soldered into the end plugs. The emf of the thermocouples was measured with an R37-1 dc potentiometer (accuracy class 0.01). The average heat-transfer coefficients $\bar{\alpha}$ were determined for each mode on the basis of the resulting distributions of the temperatures over the length and perimeter of the heat-supply zone and of the saturation temperature in the volume of the heat carrier.

The dependence $\bar{\alpha} = f(q, l)$ for different cooling conditions is presented in Fig. 1. This dependence is typical for different diameters of the heat-transfer elements, degrees of filling, angles of inclination, and working liquids. An analysis of the experimental data showed that, depending on the heat-flux density q, there are three heat-exchange regions, the existence of which was also confirmed by visual observations of the internal processes in glass thermosiphons 0.5 m long and with diameters $d_{in}$ = 6, 12, and 24 mm operating on Freons 11 and 113.

Region I-II corresponds to free convection and undeveloped boiling. At low values of $q < 2000$ W/m$^2$ in the heat-supply zone one observes free-convective heat exchange which is characterized by considerable superheating of the liquid. The free convection is accompanied by the evaporation of the heat carrier from its surface. The existence of the evaporation process is indicated by the runoff of a very thin film of condensate, which is most clearly seen in the adiabatic zone. Undeveloped boiling is observed in the heat-supply zone at $q > 2000$ W/m$^2$. This heat-exchange mode corresponds to a low density of evaporation centers, a decrease in the superheating of the liquid, and a decrease in the wall temperature. Intense convection is observed along with the evaporation. In region I-II the heat-transfer coefficient depends weakly on the length of the supply zone and is approximated by the curve $\bar{\alpha} \propto q^{0.25}$.

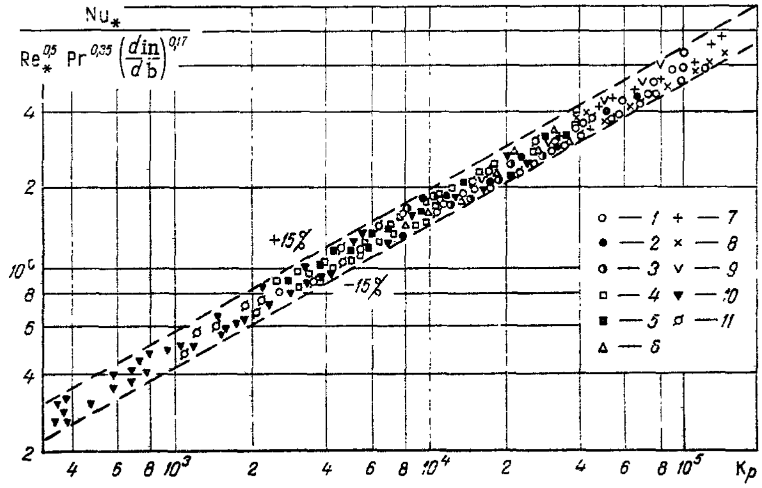


Fig. 3. Generalization of experimental data on boiling in two-phase thermosiphons: 1) Freon 11, $d_{in} = 24$ mm, $\Psi = 90°$; 2) 12 mm, $\Psi = 30°$; 3) 6 mm, $\Psi = 5°$; 4) Freon 113, $d_{in} = 24$ mm, $\Psi = 60°$; 5) 12 mm, $\Psi = 90°$; 6) 6 mm, $\Psi = 30°$; 7) Freon 142, $d_{in} = 24$ mm, $\Psi = 90°$; 8) 12 mm, $\Psi = 30°$; 9) 6 mm, $\Psi = 90°$; 10) water, $d_{in} = 24$ mm, $\Psi = 90°$; 11) ethanol, $d_{in} = 24$ mm, $\Psi = 90°$.

A further increase in the heat load leads to a sharp decrease in the wall temperature, with the liquid temperature approaching the saturation temperature in the process. A transition sets in to the region of developed bubble boiling III-IV, where a set of evaporation centers develops whose number increases with an increase in q. The heat-exchange process is self-similar with respect to $l$ and is described by the function $\bar{\alpha} \propto q^{0.5}$.

The region IV-V corresponds to the mode of hydrodynamic stabilization of the process of developed bubble boiling. The boiling in the heat-supply zone proceeds so vigorously that the entire volume of liquid through which the bubbles float up consists of a saturated two-phase mixture whose volume is considerably greater than the volume of the original filling. The heat-transfer coefficient in the region of hydrodynamic stabilization essentially depends on the length of the supply zone and is approximated by the function $\bar{\alpha} \propto q^{0.98}$.

A comparison of our observations with the results of the investigation of [11] showed that both in a large volume and in two-phase thermosiphons free convection and undeveloped boiling precede developed bubble boiling. Qualitative agreement is observed between the heat-exchange processes in the unstable boiling mode in thermosiphons and in a large volume. In connection with the limited sizes of the heat-supply zone and the phase interface in two-phase thermosiphons, the heat-exchange laws in the region of developed boiling differ essentially from the well-known concepts on the mechanism of the process under the conditions of a large volume. This difference is expressed in the fact that the vapor bubbles which have separated from the surface, continuously colliding with each other, change the directions of their motion, and this in turn leads to a decrease in the floating-up velocity and hinders their emergence at the free surface.

The concentration of the bubbles and the degree of their interaction increase as the evaporation mirror is approached. The limited size of the phase interface and the facts presented above are the reasons why not all the bubbles are able to emerge at the free surface and be destroyed, which leads to the appearance of a two-phase layer with a high concentration of the vapor phase. The chaotic nature of the bubble motion and the thickness of the layer saturated by the vapor phase increase with an increase in the heat load. Intense turbulization of the boundary layer and the volume of the heat carrier by chaotically moving bubbles is observed in the entire heat-supply zone.

Water cooling of the condensation zone increases the intensity of heat removal, which leads to a decrease in the saturation pressure in the cavity of the thermosiphon. The decrease in the pressure $p_s$ does not cause a qualitative or quantitative change in the intensity of heat transfer in region I-II, but the region of free-convective heat exchange and undeveloped boiling (I-II') increases somewhat in the process. In any case, the start of the boiling zone III'-IV depends on the kind of liquid and the pressure in the system, since the critical radius of a vapor bubble is determined by the thermophysical properties of the liquid and by the saturation temperature $t_s$. The extent of the region of unstable boiling and the start of developed boiling are characterized by the

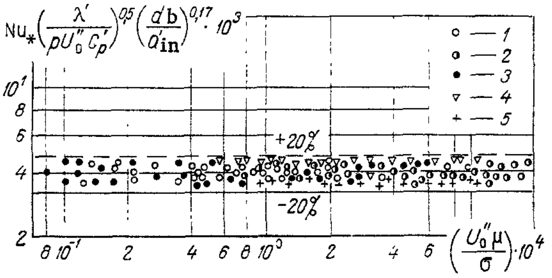


Fig. 4. Generalization of experimental data on boiling in two-phase thermosiphons in the coordinates $\{Nu_*(\lambda'/pU''_0c'_p)^{0.5}(d_b/d_{in})^{0.17};\ U''_0\mu'/\sigma\}$: 1) Freon 11; 2) Freon 113; 3) Freon 142; 4) water; 5) ethanol.

combined action of the saturation pressure $p_s$ and the supplied heat-flux density q, on which the number of active evaporation centers depends.

The heat-exchange intensity during developed bubble boiling and the transition to the region of hydrodynamic stabilization also depend on $p_s$ and q. In this region an increase of 1.5 times in the pressure $p_s$ with q = const causes an increase of 20-25% in $\bar{\alpha}$. The experiments showed that with a twofold decrease in the pressure $p_s$, upon a threefold increase in q the boiling process chantes to the region of hydrodynamic stabilization.

Although the region of convection and undeveloped bubble boiling is of definite interest in an investigation of the heat-transfer capacity of two-phase thermosiphons, since it characterizes the start-up and the beginning of their efficient operation, in the present work more attention was paid to regions III'-IV and IV-V, since they are more characteristic in the stable operation of thermosiphons.

In Fig. 2 it is shown that the heat-transfer coefficient $\bar{\alpha}$ does not depend on the length of the heat-supply zone or the degree of filling, although it is affected by the inner diameter of the thermosiphon. With a decrease in the diameter the escape of the vapor phase from the heat-supply zone is hindered, and this leads to a decrease in the heat-exchange intensity.

The empirical dependence of $\bar{\alpha}$ on the heat-flux density and the saturation pressure in regions III'-IV and IV-V has the form

$$\bar{\alpha} = Cq^{0.5}p_s^{0.22}, \tag{1}$$

where C = 11.7 for $d_{in}$ = 6 mm, C = 14.1 for $d_{in}$ = 12 mm, and C = 16.5 for $d_{in}$ = 24 mm.

Adopting as the basis a similarity equation of the form

$$Nu_* = C\,Re_*^n\,Pr^{0.35}\,K_p^{0.65}, \tag{2}$$

recommended by S. S. Kutateladze for generalized heat exchange during boiling, as a result of the generalization of our test data on the boiling of Freons 11, 113, and 142 as well as water and ethanol under the conditions of a closed thermosiphon in the range of q = (0.6-110) · $10^3$ W/m², $p_s$ = (0.1-15) bar, and $d_{in}$ = 6-24 mm, we obtained the equation

$$Nu_* = 0.0123\,Re_*^{0.5}\,Pr^{0.35}\,K_p^{0.54}\left(\frac{d_{in}}{d_b}\right)^{0.17}, \tag{3}$$

which describes the experimental data with an error of ±15% (see Fig. 3). The results of the generalization show that during developed boiling of the heat carrier in a two-phase thermosiphon, its orientation in space in a wide range of variation of the angle of inclination $\Psi$ = 5-90° does not affect the heat transfer.

A comparison of Eq. (3) with similar functions obtained for the conditions of liquid boiling in a large volume [11, 12] shows that the higher heat-exchange intensity in a thermosiphon is reflected with the help of the appropriate values of the constant and of the exponents to $Re_*$ and $K_p$ and by the introduction of the simplex $d_{in}/d_b$, which in sum characterizes the turbulization of the boundary layer by the separating and moving vapor bubbles.

The results of the experimental investigation were analyzed from the point of view of the analogy of the heat-exchange process during boiling on solid surfaces and of the bubbling of a liquid by a gas through a perforated wall [13, 14]. It was found that the heat-exchange intensity in two-phase thermosiphons is proportional to the bubbling velocity $U_0'' = q/r\rho''$ to the power $n = 1/2$. The analysis of the results made it possible to assume that the number $Nu_*$ is proportional to the criterion $K_*^{0.5}$. The experimental data obtained for all the investigated liquids were generalized (see Fig. 4) with a scatter of ± 20% in the form of the equation

$$Nu_* = 4.05 \cdot 10^{-3} K_*^{0.5} \left(\frac{d_b}{d_{in}}\right)^{0.17}. \tag{4}$$

As one would expect, the constant coefficient in Eq. (4) has a considerably higher value, i.e., 2.7 times higher, than that in Eq. (12) of [14], which describes the boiling process in a large volume.

## NOTATION

$\bar{\alpha}$, average heat-transfer coefficient in the heat-supply zone; q, supplied heat-flux density normalized to surface of heating zone; p, pressure; $\rho'$, $\rho''$, densities of liquid and vapor; $\sigma$, $\mu$, $\nu$, $a$, $\lambda$, coefficients of surface tension, dynamic and kinematic viscosity, thermal diffusivity, and thermal conductivity; $c_p$, specific heat capacity at constant pressure; $a_*$, velocity of sound; $l$, length of supply zone; $d_{in}$, inner diameter; $\Omega$, degree of volume filling, normalized to volume of thermosiphon; r, latent heat of vaporization; $U_0'' = q/r\rho''$, reduced velocity of vapor; $\Psi$, angle of inclination; $Re_* = q\sqrt{\sigma/g(\rho' - \rho'')}/r\rho''\nu$, Reynolds number; $Pr = \nu/a$, Prandtl number; $K_p = p/\sqrt{\sigma g(\rho' - \rho'')}$, pressure number; $K_* = Pe_*/M_*^2 = (pU_0'' c_p'/\lambda')$; $Pe_* = U_0''\sqrt{\sigma/g(\rho' - \rho'')}/a_*$, Peclet number; $M_*^2 = \rho'\sqrt{\sigma/g(\rho' - \rho'')}/p$, Mach number.

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