HEAT TRANSFER DURING CONDENSATION IN VERTICAL CLOSED THERMOSIPHONS

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The results are presented of an experimental investigation of the hydrodynamic characteristics of a two-phase layer and of the heat-transfer behavior under characteristic operating conditions of the condensing section of a thermosiphon.

A considerable number of papers have dealt with the investigation of heat transfer during the condensation of vapors of various heat transfer media under the conditions of a closed thermosiphon [1-5]. An analysis of these papers shows that in spite of a consensus of opinion on the complexity of the phenomenon being considered, the investigations which have been made have been carried out in most cases without taking into account the changes of the hydrodynamic characteristics of the two-phase systems and the phase flow regimes in the closed cavity of a thermosiphon. In addition, it is known that because of the possible existence of different phase flow regimes during the condensation of a moving vapor, relationships for the intensity of heat transfer can be obtained on the basis of isolating the characteristic hydrodynamic regimes. Otherwise the results obtained from experimental investigations of the mean heat transfer can be used for calculating only identical processes [6]. This circumstance explains the inconsistencies and known contradictions between the results of different papers dealing with heat transfer during condensation in thermosiphons [1, 5].

Among the characteristic features of the condensation of vapor under the conditions of the closed cavity of a thermosiphon, the following should be mentioned: the effects on the heat transfer behavior of residual noncondensable gases which become separated; the possible interaction of the phases at the interface under conditions of countercurrent flow of the liquid film and vapor, changes of the physical level of the two-phase mixture, and possible submergence by the two-phase mixture under conditions of surface condensation; and the mutual effects of the hydrodynamic phenomena in the boiling and condensing sections accompanied by droplet entrainment of liquid into the condensing section. An analysis of these features makes it possible to conclude that reproducible results on the intensity of heat transfer during condensation in thermosiphons can be obtained only under the conditions of controlled characteristic operating regimes of the condenser, which can be isolated on the basis of a preliminary study of the hydrodynamic characteristics of the two-phase system of a thermosiphon. As the main (limiting) operating regimes of the condensing section of a thermosiphon, the following can be isolated: 1) film condensation of the pure vapor of the intermediate heat transfer medium under the conditions of the countercurrent flow of a descenting liquid film and an ascending vapor stream; 2) condensation of vapor from a dynamic two-phase layer (DTL) of the heat transfer medium under the conditions of the total submergence of the condenser surface by the two-phase mixture.

In order to isolate these regimes of heat transfer, a detailed investigation was carried out into the relationship for the change of the physical level of the two-phase mixture in the thermosiphon which characterizes the boundary between the zones of undivided flow (vapor—liquid mixture) and divided flow (film of liquid — core of vapor) of the phases of the heat transfer medium. An analysis of the literature data showed that this problem is completely unstudied with respect to the conditions in thermosiphons, and that the use of known results for bubbling devices does not appear to be possible in view of the specifics of the process hydrodynamics in the two-phase layer of the thermosiphon (lateral blowing of the gas phase, small dimensions of the cross section of the layer compared with its height, etc.).

The investigations were carried out in thermosiphons made of Kh18N10T stainless steel tubes with lateral imput of heat by means of the passage of a dc current directly through the walls of the tube. The height of the dynamic two-phase layer (DTL) was determined by means of a glass insert in the adiabatic section of the thermosiphon. The mean volumetric

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TABLE 1. Ranges of the Parameters in the Investigations

Heat-transfer medium	d_{in}, mm	l_c, m	l_a, m	P, MPa	$q_0 \times 10^{-4}$, W/m	W''_0, m/sec	l_T, m
Water	18;36;66	0,39—1,335	0,3—1,3	0,1—0,7	1—12,0	0,1—13,5	2,0;3,2
Ethyl alcohol	18;36	0,39—1,335	0,3—1,3	0,05—1,0	1—10,0	0,1—3,5	2,2;3,2
Freon-11	18;36	0,39—1,335	0,3—1,3	0,1—1,0	1—10,0	0,1—3,5	2,2;3,2

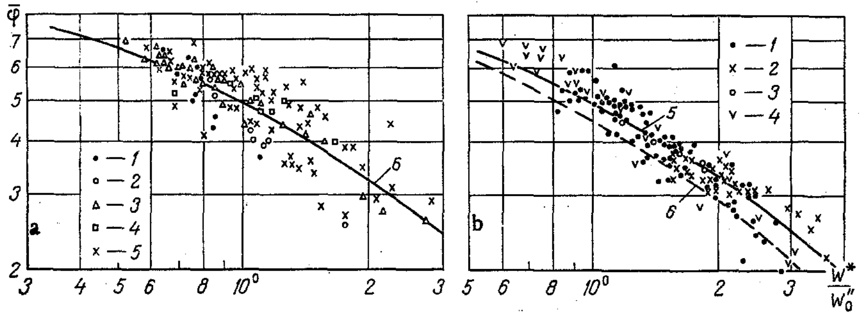

Fig. 1. Generalization of the experimental data on the mean vapor contents of DTLs for the slug regime (a) and the emulsion regime (b) of the vapor flow: a): 1) Freon-11; d_{in} = 14 mm; 2) ethanol; d_{in} = 14 mm; 3) water; d_{in} = 36 mm; 4) water; d_{in} = 20 mm; 5) Freon-11; d_{in} = 20 mm; b): 1) Freon-11; d_{in} = 36 mm; 2) Freon-11; d_{in} = 66 mm; 3) water; d_{in} = 66 mm; 4) ethanol; d_{in} = 36 mm; 5) according to equation (4); 6) according to the data of [7] for a bubbling layer, water.

content in the separate sections and in the entire two-phase layer

$$\varphi_i = \frac{\Delta h_i}{h_i}, \tag{1}$$

$$\bar{\varphi} = 1 - \frac{H_{wt}}{H_{phys}} \tag{2}$$

were taken as the dimensionless quantitative characteristics. The experiments were carried out over the ranges of the parameters shown in Table 1.

Visual observations showed that the structures of the DTLs of the thermosiphons which were studied took the form of highly turbulized vapor—liquid mixtures with droplet entrainment at various heights. The height of the layer depended on the heat loading and the amplitude of the fluctuations of the layer, which was governed by the nature of the liquid, the pressure, and the diameter of the thermosiphon. Two main regimes of motion of the vapor phase were established: slug flow and emulsion flow. The slug regime was observed in thermosiphons of small diameter at relatively low pressures; it was unstable, and was accompanied by considerable pulsations. As the thermosiphon diameter and the pressure were increased the motion of the vapor phase stabilized and underwent a transition to the emulsion regime with a relatively uniform distribution of vapor bubbles.

An analysis of the results which were obtained showed that the important parameters which influence the vapor content of the two-phase layer are the reduced velocity of the vapor phase, the diameter of the thermosiphon, and the nature and pressure of the intermediate heat transfer medium. This made it possible to obtain the following system of generalized variables for describing the way in which the unknown hydrodynamic characteristic $\bar{\varphi}$ varies:

$$\bar{\varphi} = f(\mathrm{Fr},\ \mathrm{Ar}, \mathrm{Bo},\ K_p). \tag{3}$$

According to the visual observations and the evaluation of the experimental data on $\bar{\varphi}$ as a function of the Bond number Bo, it was established that the range of Bond numbers Bo $\leqslant 18$

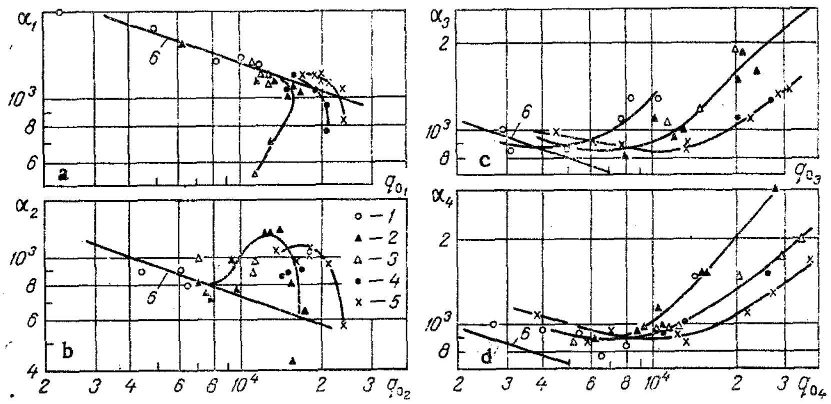

Fig. 2. Effect of the adjacent vapor stream on the distribution of the heat-transfer intensity over the height of the condenser (a, b, c, d: uppermost and successive sections of the condenser, respectively): 1) P = 0.1 MPa; 2) P = 0.205 MPa; 3) P = 0.3 MPa; 4) P = 0.4 MPa; 5) P = 0.5 MPa; 6) according to the Nusselt formula.

corresponds to the slug regime of vapor motion, and the range $Bo \geqslant 30$ to the emulsion regime. The range of Bond numbers $18 \leqslant Bo \leqslant 30$ can be assigned to a transitional regime of motion of the vapor phase.

The kinematic model of a bubbling layer by D. A. Labuntsov et al. [7] was used in order to obtain a generalized design relationship. According to this model, the value of φ in the stabilized section of the distribution curve for the vapor content over the height of the layer can be determined from

$$\varphi = \frac{1}{1 + W^*/W_0''}, \tag{4}$$

where $W^* = W^\circ\psi$ is the group velocity of rise of the vapor bubbles; W° is the rate of rise of a single bubble; ψ is the bubble interaction factor. By using as the scale for the quantity W° [8] the critical velocity of coarse liquid and vapor particles in continuous liquid and gas phases satisfying the equation

$$K^\circ = CK_p^n, \tag{5}$$

where $C = 8.2$, $n = -0.17$ when $K_p \leqslant 4 \cdot 10^4$, and $C = 1.35$, $n = 0$, when $K_p \geqslant 4 \cdot 10^4$, Eq. (4) was in agreement with the experimental data on φ for the DTLs of the thermosiphons. The following relationship was then obtained for the bubble interaction factor:

$$\psi = 110\,Ar^{-0.25} \tag{6}$$

for the slug regime, and $\psi = 3.78$ for the emulsion regime of the vapor phase motion. The results of the generalization are shown in Fig. 1. Equations (4)-(6) together with the relationship (2) make it possible to calculate the true level of the two-phase mixture at a given value of H_{wt} or the degree of filling of the thermosiphon which is based on the optimum value of the physical level for various actual conditions.

Investigations of heat transfer during condensation were carried out in an apparatus which has been described in detail in [9, 10].

In order to determine local heat-transfer coefficients the condensation zone was built in the form of a four-section recuperative heat exchanger of the concentric tube type which was coupled to the adiabatic section. The wall temperature in each section of the heat exchanger was measured by means of a copper resistance thermometer. The temperature and flow rate of the cooling water were determined by means of thermocouples and RS-3 and RS-5 rotameters. Here a regime of condensation of pure vapor was ensured by selecting the necessary length of the adiabatic section and the degree of filling of the thermosiphon with the heat-transfer medium to guarantee the condition of nonsubmersion of the condenser surface with the two-phase mixture, by removing the noncondensable gases through a vent valve in the upper

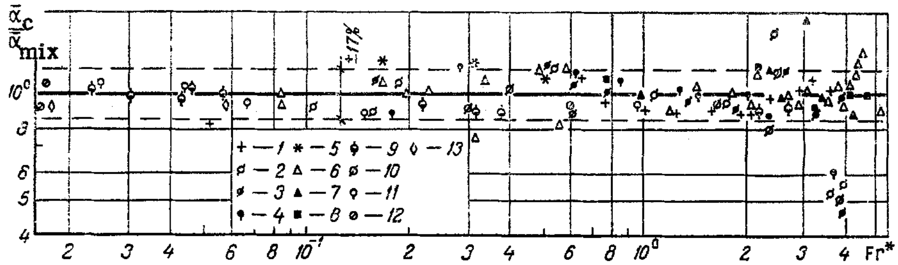

Fig. 3. Generalization of the experimental data on the condensation of pure vapors in thermosiphons: 1-5) Freon-11, d_{in} = 18 mm [1) P = 0.1 MPa; 2) P = 0.3 MPa; 3) P = 0.4 MPa; 4) P = 0.5 MPa; 5) P = 0.6 MPa; 6-8) Freon-113, d_{in} = 18 mm] [6) P = 0.15 MPa; 7) P = 0.4 MPa; 8) P = 0.7 MPa); 9, 10) Freon-11, d_{in} = 36 mm; [9) P = 0.1 MPa; 10) P = 0.5 MPa]; 11, 12) ethanol, d_{in} = 23 mm; [11) P = 0.25 MPa; 12) P = 0.5 MPa]; 13) water, d_{in} = 66 mm; P = 0.15 MPa.

part of the thermosiphon, and by preventing the entrainment of the liquid phase of the heat transfer medium by means of an insert for droplet separation. As a result, the effect of the countercurrent flow of the vapor on the intensity of film condensation could be studied.

Representation of the experimental data which were obtained in terms of the coordinates

$$\mathrm{Nu}^* = f(\mathrm{Re}_c) \tag{7}$$

showed that when different heat transfer media were used in the thermosiphon it was possible for different flow regimes of the condensate film to occur, varying from pure laminar flow to mixed flow. It was established here that the mean values of the heat-transfer coefficient deviated from the relationship for the laminar-wavy regime [6] at a value of the Reynolds number Re_K which is considerably smaller than the value Re_K = 400 [6] which is most often suggested for the condensation of a stationary vapor on a vertical surface. In order to explain the effect on the intensity of heat transfer of the adjacent vapor stream ($W_0'' = 4q_0 l_c/(r\rho'' d_{in})$), the local discrete heat-transfer coefficients in the different sections over the height of the condenser were studied. The characteristic dependences $\alpha_c = f(q_0)$ expressing the effect of the value of W_0'' for the four sections of the condenser are shown in Fig. 2. It can be seen from the figure that beginning at some definite value of q_0, the heat-transfer coefficients over the height of the condensation section are characterized by a sharply erratic behavior which is caused by changes in the interaction conditions between the adjacent streams of liquid and vapor. However, the intensification of heat transfer which is observed at the inlet of the vapor into the condenser is compensated by a reduction in the value of α in the upper part as a result of the entrainment of liquid and the increase in thickness of the condensate film. Comparison of the data which have been obtained on the mean heat transfer with known relationships [11] for the condensation of a stationary vapor (Fig. 3):

$$\overline{\mathrm{Nu}}^* = 1{,}18\mathrm{Re}_c^{-\frac{1}{3}}, \tag{8}$$

$$\overline{\mathrm{Nu}}^* = \frac{0{,}054\mathrm{Pr}^{0,4}\,\mathrm{Re}_c}{\mathrm{Re}_c^{5/6} - 47 + 21\mathrm{Pr}^{0,4}} \tag{9}$$

in terms of the parameters $\mathrm{Nu}^*_{exp}/\mathrm{Nu}^* = f(\mathrm{Fr}^*)$ shows a very satisfactory agreement between the experimental values and the values calculated by Eqs. (8) and (9) for $\mathrm{Re}_K > 100$ up to the heat exchange crisis, which determines the upper limit of the flooding regime for two-phase flow under conditions in a thermosiphon [12].

The results of the investigations of the intensity of heat transfer under conditions in which the condensing section of the thermosiphon was submerged in the two-phase mixture are given in Fig. 4. The experimental data were determined in thermosiphons which did not have transport sections and which had degrees of liquid filling of the heating zone ε_h = 60-100%. The ranges of variation of remaining parameters in the investigation are given in Table 1.

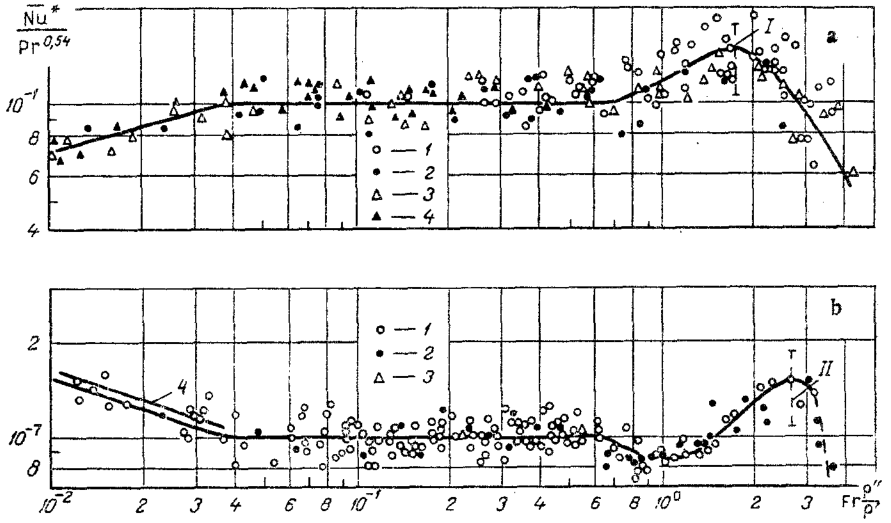

Fig. 4. Generalization of the experimental data on the condensation of vapor from a dynamic two-phase layer for the slug regime (a) and the emulsion regime (b) of the flow of the vapor phase: a): 1–3) d_{in} = 18 mm; 4) d_{in} = 36 mm; 1) Freon-11; 2) ethanol; 3, 4) water; b): 1, 2) d_{in} = 36 mm; 3) d_{in} = 66 mm; 1) ethanol; 2) Freon-11; 3) water; 4) according to Nusselt formula for stationary vapor.

From Fig. 4 it can be seen that on the whole condensation under the conditions of a dynamic two-phase layer leads to an increase in the intensity of heat transfer compared with the Nusselt solution for film condensation of stationary vapor. The intensification of heat transfer is caused by the hydrodynamic features of the process during condensation in a dynamic two-phase layer. The transverse motion of the gas phase and its swirl, the high reduced velocity of the vapor stream, and the growth of the interfacial surface area of the layer cause the high intensity of condensation.

An analysis of the results of the investigations showed that the heat transfer behavior during condensation depends on the regime of motion of the vapor phase. In this connection, generalizations of the experimental data were carried out for the slug regime and the emulsion regime in agreement with the established boundaries of these regimes (Bo < 18 for the slug regime, Bo > 30 for the emulsion regime), and are represented in Fig. 4 in the form of the relationship

$$\frac{\overline{\mathrm{Nu}}^*}{\mathrm{Pr}^{0,54}} = f(\mathrm{Fr}^*). \tag{10}$$

As follows from the generalizations which have been presented, when $\mathrm{Fr}^* < 4\cdot 10^{-2}$ the heat-transfer relationships in the individual vapor flow regimes are different in nature, which is caused by the differences in the hydrodynamic conditions in the cooling zone in the slug and emulsion regimes. In the first case the condensation zone is filled almost completely with alternating vapor and liquid slugs, and in the second case the condenser is only partly filled with the two-phase mixture (the bubble regime of bubbling). As a result of these features, the heat transfer coefficients in the bubble regime in this range of the parameter Fr^* obey in practice the relationship given by Nusselt's equation. In the slug regime a considerable decrease in the intensity of heat transfer is observed. In this case, the mean values of the heat transfer coefficient correspond to the equation

$$\overline{\mathrm{Nu}}^* = 0{,}21\mathrm{Pr}^{0,54}\mathrm{Fr}^{0,24}. \tag{11}$$

In the range of conditions $\mathrm{Fr}^* = 4\cdot 10^{-2} - 6\cdot 10^{-1}$ the relationships for heat transfer during condensation in both the slug regime and the bubbling regime are similar and can be calculated from the equation

$$\overline{\mathrm{Nu}}^* = 0{,}1\mathrm{Pr}^{0,54}. \tag{12}$$

Some of the intensification of heat transfer observed at $Fr^* > 0.6$ is caused by the active interaction of the ascending (in the core of the flow) and descending (at the wall) streams of the heat transfer medium. Comparison of the results which have been obtained with the data of the hydrodynamic investigations of the countercurrent motion of the phases under the conditions of a closed two-phase thermosiphon showed that the beginning of the increase of the intensity of heat transfer corresponds to the beginning of the flooding regime, which is characterized by the beginning of a sharp increase in the friction factor of the vapor flow. The lines I and II in Fig. 4 correspond to the flow crisis of the two-phase stream in the thermosiphons which is related to the maximum of the friction factor [12] for the corresponding flow regimes. The maximum of the relationship $\overline{Nu}^* = f(Fr^*)$ was observed at a value of the group Fr^* corresponding to the reduced velocity of the vapor stream at the entry to the condensing section at the moment when the liquid phase of the heat transfer medium is held up. A further increase in the reduced vapor velocity contributes to an accumulation of the liquid phase in the condensing section, which leads to a sharp reduction in the intensity of heat transfer (as also in the case of condensation of a pure vapor) and to the cessation of the normal operation of the thermosiphon.

NOTATION

φ, true volumetric vapor content; Δh_i, level difference in pressure gauge; h_i, height of i-th section of two-phase layer; H_{wt}, H_{phys}, weight and physical levels of the two-phase layer; $Fr \equiv W''^2/g\delta$, Froude number; $Ar = g\delta^3/\nu'^2 \cdot (\rho' - \rho'')/\rho''$, Archimedes number; $Bo \equiv d/\delta$, Bond number; $K_p \equiv P\delta/\sigma$, pressure group; $K^\circ \equiv W^\circ\sqrt{\rho'}/\sqrt[4]{\sigma/g(\rho' - \rho'')}$, stability group; $Fr^* = Fr \cdot \rho''/\rho'$; $\delta = \sqrt{\sigma/g(\rho' - \rho'')}$, Laplace constant; $Pr \equiv \nu'/a'$, Prandtl number; $Re = q_0 L_c/r\mu'$; Reynolds number; $Nu^* = \alpha_c/\lambda'[\nu'^2/g(1 - \rho'')/\rho']$, Nusselt number; q_0, heat flux density through condenser surface; W'', reduced velocity of vapor stream; r, latent heat of vaporization; μ, ν, dynamic and kinematic viscosities; ρ', ρ'', densities of liquid and vapor; σ, surface tension; α_c, heat-transfer coefficient during condensation; l_c, height of condenser. Subscripts: mix, mixed; c, referring to condensation; wt, weight; exp, experimental; h, referring to heating zone; z.h., heating zone; in, inside; T, thermosiphon.

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