NOTATION

Q, heat flux, W; i, evaporation heat, J/kg; k, porosity, %; l, length, m; ρ, density, kg/m^3; A, cross-sectional area of wick, m^2; η, dynamic viscosity, N·sec/m^2; σ, surface-tension coefficient, N/m; θ, wetting angle, deg; r_e, effective radius of pores, m; φ, angle of inclination of pipe to the horizontal, deg; m_v, mass of vapor, kg; d_{in}, inner diameter of pipe jacket, m; δ, thickness of wick wall, m; p, pressure, abs. atm; m_l, mass of liquid, kg.

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INVESTIGATION OF THE MAXIMUM HEAT-TRANSFER CAPACITY OF CLOSED TWO-PHASE THERMOSIPHONS

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UDC 536.27:669.214

We determine the maximum heat fluxes transmitted by thermosiphons, depending on the geometric dimensions, the working pressure, the type of liquid used, and the degree of filling with coolant.

In recent years there has been an increase of interest in heat-transfer devices constructed in the form of wickless heat pipes, i.e., closed two-phase thermosiphons. The advantages of these devices (simplicity of manufacture, reliability of operation, fairly high heat-transfer capacity, etc.) open prospects for their widespread use in various fields of technology [1-3].

There are a good many studies devoted to the investigation of heat-transfer processes in closed evaporative thermosiphons. An absolute majority of these studies are concerned essentially with questions of heat exchange in the evaporation and condensation segments of the heat-transfer device. Limiting regimes of operation of closed two-phase thermosiphons have not been sufficiently investigated. Stoyanov [4] considered a heat-transfer crisis resulting from the complete evaporation of the intermediate coolant. The results of this analysis enable us to determine the maximum working temperature of a heat-transfer device as a function of the degree to which it is filled with coolant. Concerning the maximum heat fluxes transmitted by two-phase thermosiphons, there are only fragmentary data (for example, [5]), from which it is not possible to derive any generalized relationship.

In the present article we give the results of an investigation of the maximum heat fluxes transmitted by vertical two-phase thermosiphons, depending on their geometric, physical, and regime parameters. The range of variation of the parameters is shown in Table 1.

The investigations were carried on an experimental apparatus consisting of experimental thermosiphons with heat-input and heat-removal segments, electrical heating units, systems for the input of cooling water, and measuring devices. The experimental thermosiphons were

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TABLE 1. Range of Variation of Geometric and Regime Parameters

Coolant	$d_{in\,hi}$, mm	l_{hi}, mm	$l_{h\,hi}$, mm	$l_{h\,hi}/d_{in\,hi}$	$d_{in\,hr}$, mm	l_{hr}, mm	P, bars	ε_{hv}, %	ε_{tv}, %
Water	10—50	2000	500—2000	10—196	20, 60, 89	600, 1140	1—10	100—25	2—70
Ethyl alcohol	14—44	2000	120—2000	46—86	89	1140	1—20	100—75	5—10
Freon-12	10—34	2000	700—2000	58—120	89	1140	6—25	100—75	3—10

made of Kh18N10T stainless steel pipes, some with equal and some with unequal diameters in the heat-input and heat-removal segments. The condensation segment of the thermosiphon was made in the form of a "pipe-in-pipe" heat exchanger. For convenience in regulating the pressure in the inner cavity of the thermosiphon, the cooling jacket of the condenser was subdivided and consisted of three parts of different lengths.

The heat flux was supplied to the evaporation segment by passing an electric current through the wall of the pipe from two series-connected ANG-5,000/6 dc generators.

The measuring scheme was used for determining the temperatures at various points of the thermosiphon tested, the pressure in its internal cavity, the flow rate and temperature of the cooling water, and the power input. To check the temperature regime of the thermosiphon, we used Chromel—Alumel thermocouples placed on the outside surface of the pipe in the heat-input zone, as well as in capillaries introduced into the internal cavity of the thermosiphon in the lower part of the heat-input segment, and in the adiabatic zone.

The power generated in the heating zone was determined from the voltage drop across the working segment and also across the shunt connected in series in the heating circuit.

The maximum heat-transfer capacity of the thermosiphon was found by increasing the thermal load stepwise while maintaining constant pressure inside the thermosiphon and achieving a stationary regime. The point at which the operating regime was no longer normal was fixed by observing the sharp increase in the temperature of the pipe wall in the heating zone. At that moment the input source was automatically disconnected in order to avoid burnout of the pipe and damage to the thermocouples.

In the tests we investigated how the heat-transfer capacity of the thermosiphon was affected by the following parameters: the diameter and length of the heat-input segment, the pressure of the intermediate coolant and the degree to which the inner cavity of the thermosiphon was filled with it, the nature of the working liquid, and the dimensions of the condenser. The length of the heating zone was varied by shifting the upper current input terminal used for feeding the electric current into the thermosiphon pipe. In order to determine the effect of the condenser dimensions on the heat-transfer capacity of the thermosiphon, the experiments were conducted for different geometric characteristics.

The operation of a closed two-phase thermosiphon is characterized by different partial processes of heat and mass exchange taking place inside its envelope, and each of these processes can limit the heat-transfer capacity. An analysis of these processes shows that the maximum heat flux in the thermosiphon may be limited by the achievement of a critical heat-flux density when there is boiling on the heat-exchange surface, by a crisis in longitudinal heat and mass transfer, or by a passage of the coolant from the two-phase to the single-phase state when its working temperature changes. In order to clarify the nature of the crisis limiting the operation of thermosiphons in the cases of practical importance for large l/d ratios, we made visual observations on a glass thermosiphon tube with dimensions of $d_{in} = 6$ mm and $l_h = 200$ mm. Freon-12 was used as the intermediate coolant. The heating of the tube was achieved by the use of water heated to a specified temperature and pumped through the annular space between the two glass tubes. The results of the visual observations showed that for a low degree of filling and relatively low heat fluxes on the inner surface of the tube there moves a stable film of liquid washed by a countercurrent of vapor. As the heat flux increases, there is a sharp discharge of liquid from the lower part of the tube into the condensation zone, and the operation of the thermosiphon thereafter is pulsating, since the cessation of the liquid input results in a reduction in the velocity of the vapor, followed by the input of a new dose of liquid into the heating zone. This type of abnormal operation of the thermosiphon indicates that the crisis of heat and mass transfer under

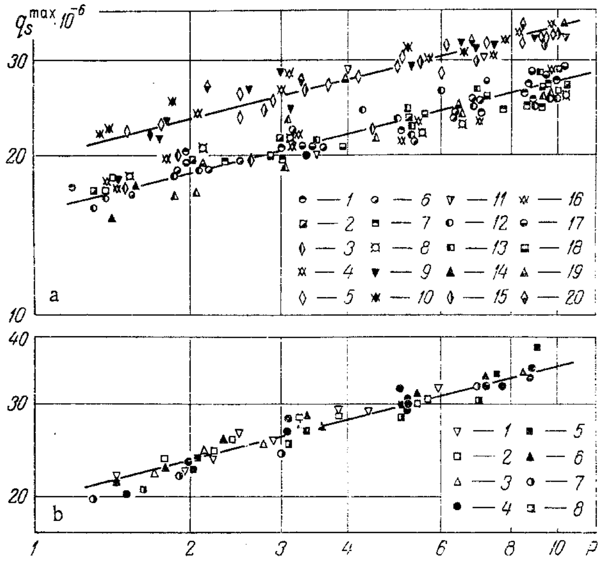

Fig. 1. Maximum allowable heat flux q_s^{max}, W/m², as a function of pressure and degree of filling: a) 1-5: l_h/d_{in} = 100 (1: ε_{hv} = 1.0; 2: 0.75; 3: 0.5; 4: 0.3; 5: 0.2); 6-11: l_h/d_{in} = 75 (6: ε_{hv} = 1.0; 7: 0.75; 8: 0.5; 9: 0.25; 10: 0.2; 11: 0.1); 12-16: l_h/d_{in} = 50 (12: ε_{hv} = 1.0; 13: 0.75; 14: 0.5; 15: 0.2; 16: 0.175); 17-20: l_h/d_{in} = 25 (17: ε_{hv} = 1.0; 18: 0.75; 19: 0.5; 20: 0.25); b) 1-3: l_h/d_{in} = 72.6 (1: ε_{hv} = 1.0; 2: 0.5; 3: 0.25); 4-6: l_h/d_{in} = 49 (4: ε_{hv} = 1.0; 5: 0.5; 6: 0.25); 7, 8: l_h/d_{in} = 25 (7: ε_{hv} = 1.0; 8: 0.5).

these conditions is determined primarily by the interaction of the countercurrents of vapor and liquid at the interface between the phases and by the achievement of some critical velocity in the vapor flow. Analogous phenomena take place when there is annular countercurrent two-phase flow in vertical heat pipes; these are referred to by the term "flooding" [5].

Because of these characteristics of the crisis phenomena in long thermosiphons, when we processed the experimental data, the quantity we chose to calculate was the thermal flux density q_s per unit of cross-sectional area of the channel in the heat-input zone.

Figure 1a shows the results of the experiments for a pipe with an inner diameter of 20 mm, in the form of the variation of maximum heat-flux density q_s^{max} as a function of pressure for various values of the length of the heating zone and the degree of filling of that zone with liquid. As can be seen from the figure, the maximum transferred heat flux depends to a great extent on the pressure in the thermosiphon cavity and, in the range investigated, is independent of the length of the heating zone. The effect of the degree of filling of the heating zone with liquid is seen only in the range of relatively small values of this parameter. As the degree of filling increases to values greater than 25% of the heating zone, there is a decrease in the heat flux q_s^{max} by about 25%. This phenomenon is apparently attributable to the fact that since the pipe diameter and the diameter of formation of vapor bubbles are comparable, the liquid is forced into the upper part of the thermosiphon, and thus the conditions for the stable existence of countercurrent between liquid and vapor become worse.

These phenomena were observed by the authors of [6] in a case in which they were determining the coefficients of heat exchange. They determined the depth of filling of a two-phase thermosiphon (∿18% of its length) at which the power removed by the thermosiphon was a maximum.

The results of experiments for the evaporation segment of a thermosiphon with a diameter of d_{in} = 27 mm (Fig. 1b) showed that in this case the function $q_s^{max} = f(P, l_h)$ is of the

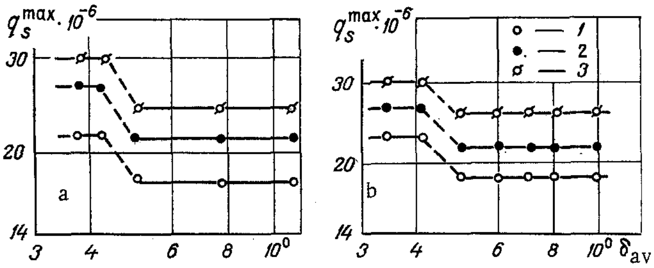

Fig. 2. Maximum allowable heat flux as a function of the amount of intermediate coolant, referred to the inner surface of the thermosiphon: a) condenser d_{in} = 60 mm, l = 600 mm; b) condenser d_{in} = 89 mm, l = 1,140 mm; 1) P = 1.7 bars; 2) 4; 3) 8. δ_{av}, mm; q_s^{max}, W/m².

same kind as for a pipe with d_{in} = 20 mm. However, the previously observed decrease of q_s^{max} as a function of ε was not observed this time. It is probable that in this case the formations of vapor bubbles do not completely cover the entire cross section of the channel and do not force the liquid from the heating zone into the upper part of the thermosiphon.

Experiments conducted with other values for the typical dimensions of the heat-input segment of a thermosiphon with water filling showed that for pipes with a diameter d_{in} < 20 mm the experimental data agree qualitatively and quantitatively with the data for a pipe with d_{in} = 20 mm, while for pipes with d_{in} > 27 mm they agree with the data for a pipe with d_{in} = 27 mm.

In order to determine how the maximum heat-transfer capacity of the thermosiphon is affected by the geometric dimensions of the heat-removal segment, we conducted experiments on three condensers with different dimensions (see Table 1). The pipe diameter in the heating zone was kept constant at d_{in} = 20 mm. As the experiments showed, the geometric dimensions of the condensation segment, in the investigated range of variation, do not affect the maximum heat flux transmitted by the thermosiphon. However, it was found that the onset of differences in experimental data according to the degree of filling of the thermosiphon with liquid occurred for each condenser with a different value of ε, irrespective of which volume (the volume of the evaporation segment or the total thermosiphon volume) the liquid volume was referred to. This brought up the question of defining the relative value ε in such a way that the discontinuous variation of the maximum heat flux would be uniquely characterized irrespective of the dimensions of the individual segments of the thermosiphon. Apparently, the variation of the maximum heat-flux value is related to the variation of the flow regime for the vapor and liquid phases, which, in turn, is determined not by the volume but by the thickness of the liquid layer on the inner surface of the pipe. For a thinner film of liquid the flow is annular in nature, with highly stable countercurrent between liquid and vapor. As the film thickness increases, there is wave formation, separation of the liquid from the surface of the film, and forcing of the excess liquid into the upper part of the thermosiphon. The flow of the liquid and vapor becomes pulsating, as was confirmed by visual observations. These phenomena lead to an increase in the hydraulic resistance when there is relative motion of the phases and, consequently, to a less frequent occurrence of a heat- and mass-transfer crisis.

Figure 2 shows the experimental data for two sets of typical dimensions of the condenser when the volume of filling liquid is referred to the total inner surface of the thermosiphon. The degree of filling defined in this way is the average thickness of the liquid film on the inner surface of the thermosiphon. As can be seen from Fig. 2, the variation of the maximum heat-transfer capacity of the thermosiphon in each case comes at the same value of δ_{av}, which is equal to ∿0.4-0.5 mm, confirming that the above reasoning is correct. Similar experiments were performed on pipes with inner diameters of 10 mm and 14 mm. It was found that the limiting value of δ_{av}, which characterizes the variation in the flow regime, is practically independent of the inner diameter of the evaporation segment of the thermosiphon.

In order to ascertain the effect of the physical properties of the intermediate coolant on the maximum transmitted heat flux, we conducted experiments on liquids with sharply dif-

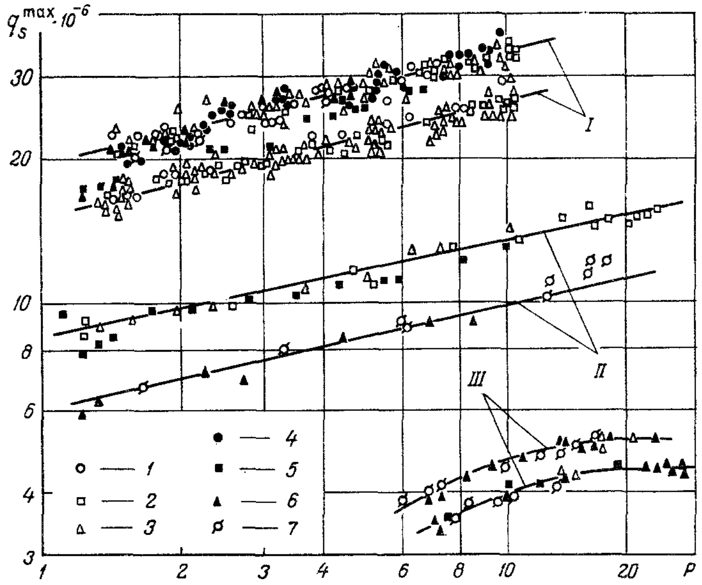

Fig. 3. Experimental data on q_s^{max} for different liquids: I) water; II) ethyl alcohol; III) Freon-12; 1) d_{in} = 50 mm; 2) 44 mm; 3) 34) mm; 4) 27 mm; 5) 20 mm; 6) 14 mm; 7) 10 mm.

fering physical and thermodynamic properties: water, ethyl alcohol, and Freon-12. The results of these experiments for heat-input segments with different geometric dimensions are shown in Fig. 3. It can be seen from the figure that for each coolant the maximum heat flux is independent of the diameter of the heat-input segment and depends only on the regime of liquid–vapor countercurrent, determined by the quantity of liquid in the inner cavity of the thermosiphon per unit of inner surface. In the experiments it was established that the limiting value of the average liquid-film thickness characterizing the transition from one flow regime to another depends only slightly on the type of liquid used and lies in the range of 0.4-0.6 mm.

The existence of two flow regimes of liquid and vapor characterized by different values of maximum heat flux was observed for all the liquids investigated at the following diameters of the evaporation segment: for water at $d_{in} \leqslant 20$ mm, for ethyl alcohol at $d_{in} \leqslant 14$ mm, and for Freon-12 at $d_{in} \leqslant 20$ mm. Thus, the effect of the degree of filling of the thermosiphon with water on q_s^{max} for coolants with different physical properties is observed only for relatively small pipe diameters in the evaporation segment ($d_{in} \leqslant 20$ mm) and for overfilling with liquid, i.e., for $\delta_{av} > \delta_{av\,cr}$. For large values of pipe diameter in the heating zone ($d_{in} >$ 20 mm) the value of q_s^{max} is independent of the degree of filling of the thermosiphon with coolant in the investigated range of variation.

In order to generalize the experimental data obtained, we considered a model of the flow crisis for liquid–vapor countercurrent in which the stability of the liquid flow was disturbed when the vapor reached some critical velocity. Such processes [7] can be generalized in the form of a criterial function

$$K = \Phi\left(\mathrm{Re};\ \mathrm{Ga};\ \mathrm{We};\ \frac{\rho' + \rho''}{\rho'};\ \frac{l_1}{l}\right). \tag{1}$$

The system of criteria (1) must be supplemented with criteria characterizing the boiling crisis for free motion of the liquid. An analysis of these criteria, which are given, for example, in [8], leads to the appearance of the complexes K_t, K_P, and Pr.

Thus, the breakdown of the stability of flow and heat exchange for the case of motion in a vertical pipe with countercurrent between vapor and boiling liquid can be described, in the general case, by the equation

$$K = f\left(\mathrm{Re};\ \mathrm{Ga};\ \mathrm{We};\ K_P;\ K_t;\ \mathrm{Pr};\ \frac{\rho' + \rho''}{\rho'};\ \frac{l_1}{l}\right). \tag{2}$$

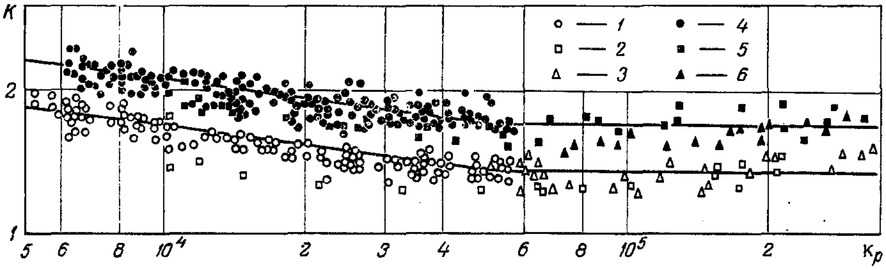

Fig. 4. Generalization of the experimental data in criterial form: 1, 2, 3) experimental data for circular flow regime; 4, 5, 6) for annular flow regime; 1, 4) water; 2, 5) ethyl alcohol; 3, 6) Freon-12.

In the case under investigation there is no forced motion of the coolant, and, in addition, the heat-transfer crises in the investigated range of parameters is independent of the geometric dimensions of the thermosiphon. Therefore, we must eliminate from the system (2) the complexes Re, Ga, We, and l'/l. Because of this the criterial equation becomes

$$K = f\left(\mathrm{K}_P;\ \mathrm{K}_t;\ \mathrm{Pr};\ \frac{\rho' + \rho''}{\rho'}\right). \tag{3}$$

Figure 4 shows the generalization of the experimental data in the form of the function $K = f(K_p)$. As can be seen from the graph, this function satisfactorily generalizes all the experimental data individually for each of the investigated flow regimes of vapor—liquid flow. The criterion K_t was not found to have any effect, apparently because there is almost a one-to-one relationship between K_t and K_p for some groups of substances [9]. The complexes Pr and $(\rho' + \rho'')/\rho'$ for the given conditions are obviously not decisive. It can also be seen from Fig. 4 that starting from a certain value there is a degeneration of the effect of the criterion K_p. To calculate the process, we can use the following relations:

$$\mathrm{K} = C_1 \mathrm{K}_P^n \quad \text{for} \quad \mathrm{K}_P = 5\cdot 10^3 - 5\cdot 10^4; \quad \mathrm{K} = C_2 \quad \text{for} \quad \mathrm{K}_P = 5\cdot 10^4 - 3.5\cdot 10^5,$$

where $C_1 = 5.65$, $n = -0.13$, $C_2 = 1.35$ for $d < 20$ mm, $\delta > \delta_{cr}$; $C_1 = 7.02$, $n = -0.13$, $C_2 = 1.7$ for $d < 20$ mm, $\delta < \delta_{cr}$, and $d_{in} > 20$ mm, $\delta \gtreqless \delta_{cr}$.

NOTATION

q_s^{max}, maximum permissible heat-flux density per cross-sectional area of heat-input segment; $d_{in\,hi}$, $d_{in\,hr}$, inner diameters of heat-input and heat-removal segments, respectively; l_{hi}, l_{hr}, lengths of heat-input and heat-removal segments; $l_{h\,hi}$, length of heated segment of heat-input zone; P, pressure; ε_{hv}; ε_{tv}, degrees of volume filling referred to volume of heating zone and to volume of thermosiphon; $\delta_{av} = V_l/F_{in}$, thickness of liquid film averaged over surface; V_l, volume of liquid at t = 20°C; F_{in}, inner surface of thermosiphon; $K = W_{cr}\sqrt{\rho''}/\sqrt[4]{g\sigma(\rho' + \rho'')}$, stability criterion; $K_p = P\sqrt{\sigma/g(\rho' - \rho'')}/\sigma$, pressure criterion; $Ga = gl^3/\nu^2$, Galileo number; $We = \sigma/g(\rho' - \rho'')l^2$, Weber number; $Pr = \nu/a$, Prandtl number; $Re = Wl/\nu$, Reynolds number; K_t, saturation temperature criterion; σ, ν, a, coefficients of surface tension, kinematic viscosity, and thermal diffusivity; ρ', ρ'', densities of liquid and vapor; W_{cr}, critical velocity of vapor.

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PROBLEM OF CONSIDERING THE EFFECT OF SUPERHEATING OF VAPOR ON INTENSE CONDENSATION PROCESS OF VAPOR FLOW ON A PLATE

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UDC 536.423.4

The effect of superheating of vapor on the intense condensation process of vapor flow on a plate is analyzed.

In the analysis of the process of film condensation of moving vapor the elucidation of the role of such secondary factors as the superheating of the vapor, the impurity of the noncondensing gas, etc., is of certain interest. The numerical solutions given in [1] are devoted to the effect of these factors on the intensity of heat transfer in the case of an isothermal plate.

The object of the present investigation is to analyze the effect of superheating of vapor on the heat transfer from a moving vapor to a plate under conditions of intense condensation at constant heat flux.

As is well known [2, 3], near the front edge of the plate the velocity profiles in the vapor phase under conditions of sufficiently intense condensation at constant heat flux correspond with a good accuracy to the asymptotic profile of Meredith and Griffith for a boundary layer with homogeneous suction [4]. The consideration of the effect of superheating of the vapor under such conditions amounts to the analysis of the temperature field in the dynamic boundary layer of Meredith and Griffith with certain assumptions.

If the compressibility of the vapor and the heat release due to friction are neglected, then the equations for the boundary layer for plane motion are written in the following well-known form:

$$\begin{gathered}\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0,\\ \rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=\mu\frac{\partial^2 u}{\partial y^2},\\ \rho C_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\lambda\frac{\partial^2 T}{\partial y^2}.\end{gathered}\tag{1}$$

We shall assume that the condition of constancy of heat flux at the wall (q = const) corresponds to homogeneous suction of the boundary layer $v=-v_0=\text{const}$ [4],* where in the notation used here v_0 is a positive quantity. System (1) is accordingly simplified:

*This condition is satisfied rigorously for the condensation of the saturated vapor when the heat flux is completely determined by the condensed mass. In the case of superheated vapor this assumption will be strictly correct only in the case when the heat flux caused by the superheating of the vapor will be constant over the length of the plate.

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