HEAT LIBERATION DURING VAPOR CONDENSATION IN A THERMOSIPHON

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A method is described for calculation of the heat-transfer coefficient during condensation of vapor of single-component and binary heat-transfer agents in a thermosiphon. Results of processing experimental data are presented.

The design of heat-exchange apparatus using thermosiphons requires a thorough examination of operating regimes, dependent on heat supply and removal conditions. With concentrated heat supply to the boiling section and convective heat exchange with the external medium in the exhaust segment, the intensity of heat transfer accompanying condensation is of major importance. A number of studies have been dedicated to this question [1-5]. These studies, which considered vertical thermosiphons, concluded that the major propositions of Nusselt theory for condensation processes are valid. At the same time, the divergence of concrete experimental results on heat transfer during condensation from theoretically expected values has been explained in differing ways.

Thus, the experimental data of [2] on heat transfer produced values several times larger than the heat-transfer coefficients calculated from the Nusselt equation, which was explained as being due to breakoff of the condensate film from the thermosiphon wall. In [3] it was found that in condensation of water vapor in a vacuum the coefficient α_{0v} initially increases, then decreases with increase in thermal flux. The increase in heat-liberation coefficient with increased thermal loading was explained in [4] as due to the presence of uncondensed gases (air) in the thermosiphon cavity. Questions of heat transfer during condensation were considered in greater detail in [5]. In accordance with [6, 7], the experimental data were processed in the form of functions

$$A = f(\mathrm{Re_c}), \tag{1}$$

$$\frac{\bar{\alpha}_{0v}}{\bar{\alpha}_{Nu}} = f(\varphi). \tag{2}$$

The parameters introduced in Eqs. (1), (2) are defined by the following expressions:

$$A = \frac{\bar{\alpha}_{0v}}{\lambda_c}\left[\frac{\nu_c^2}{g\left(1-\frac{\rho_v}{\rho_c}\right)}\right]^{1/3}, \tag{3}$$

$$\mathrm{Re_c} = \frac{q_c l_{c.s}}{r\mu_c}, \tag{4}$$

$$\varphi = \frac{\rho_v}{\rho_c}\left[\frac{\nu_v}{\nu_c}\right]^2 \frac{\mathrm{Re}^2}{\mathrm{Ga}^{2/3}}\mathrm{Re_c}^{-0,28}\varepsilon_t. \tag{5}$$

When the experimental data of [5] are processed with Eqs. (1), (2), they give satisfactory agreement with the Nusselt solution at small values of $\mathrm{Re_c}$ = 7-10 and φ = 0.1-0.6. At larger values of $\mathrm{Re_c}$ and the parameter φ there is a significant increase in the heat-liberation coefficient, which in the opinion of the authors of [5] is produced by changes in the presence of the heat-transfer agent, the flow regimes of the condensate film, and dynamic interaction between the vapor flow and the condensate film.

The cause for this divergence in the growth of heat liberation during condensation with increase in thermal loading must be sought in the method used for processing the experimental results, and the physics of the process must be considered from a somewhat different viewpoint.

In thermosiphons the vapor volume is limited, and the vapor motion is directional. Usually, the condenser section is cooled intensely, so that at its input, i.e., at the boundary of the condenser and adiabatic sections, the axial vapor

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velocity can achieve large values. The axial vapor velocity then decreases rapidly due to condensation, and in some sections of the condenser becomes close to zero.

Vapor and condensate flow regimes depend on the thermal loading, thermophysical characteristics of the heat-exchange agent used, length of the condenser section, and internal diameter of the thermosiphon. As a rule, vapor and condensate flow in thermosiphons is laminar. In order that the vapor flow become turbulent, significant thermal loading must be supplied. For example, for a thermosiphon with internal diameter d_{in} = 0.02 m (heat-exchange agent, water at P_c = 0.1 MPa, condensation temperature T_c = 100°C) a turbulent regime can be achieved at thermal loading greater than 5 kW. Considering the construction features of the thermosiphons, it is hardly possible to attain values above 2kW for the under-water heat load. Beginning at the boundary of the adiabatic and condenser sections, the axial vapor velocity decreases because of condensation and becomes equal to zero at some section, while the radial vapor motion regime in the direction toward the cooling surface will be laminar over the entire extent of the condensation zone. Consequently, in thermosiphons there is no reason to expect any sharp divergence from a regime of condensation of a slowly moving vapor with a condensate film in laminar flow.

The divergence between experiment and theory is brought about by two facts: 1) when the experimental data are processed with the well-known formula

$$\bar{\alpha}_{0v} = \frac{q_c}{\overline{\Delta T}} \tag{6}$$

the thermal load removed is referred to the entire surface of the thermosiphon condenser section; 2) the mean difference between the saturation temperature and the wall temperature is also calculated over the entire length of the condensation section. In this case the heat-liberation coefficient will always increase with increase in thermal loading. In fact, if heat-removal conditions are maintained unchanged, for example, the cooling water flow rate and its temperature at the entrance, with increase in thermal load both the saturation temperature and the wall temperature of the condensation section will increase, the latter more rapidly.

The mean temperature difference $\Delta T = T_{sat} - T_w$ will fall, while the heat-liberation coefficient α_{0v} will increase, its numerical value tending to infinity, since $\Delta T \to 0$. Such a conclusion contradicts condensation theory and demands a different treatment of the problem.

Experiments with transparent thermosiphon models with various heat-exchange agents and mixtures [8] have shown that with usual methods of heat removal by forced convection of water or air, the liquid film occupies only a portion of the length of the condenser section. The length of the condensation zone, like the vapor and condensate film flow regimes, depends on the thermal load supplied to the boiling section, the length of the adiabatic section, and the inner diameter of the thermosiphon. The method of heat removal from the condenser section has a certain effect on the length of the condensate film.

Thus, the thermal load removed from the thermosiphon is not extracted from the full surface of the condensation section but from some part of that surface, corresponding to the length of the liquid film. The calculated temperature difference must also be found for this concrete condensation zone.

The variable condensate film length l_{ci} of the i-th condensation zone will define a heat-liberation surface $f_{ci} = \pi l_{ci} d_{in}$, and a thermal flux density for the heat removed through the given surface, $q_{ci} = Q_c/f_{ci}$. We find the experimental values of the heat-liberation coefficient from the expression

$$\bar{\alpha}_{0vi} = q_{ci}/\overline{\Delta T}_i. \tag{7}$$

In this case, to generalize the experimental data we may use Eq. (1), but in Eqs. (3), (4) it is necessary to substitute the values of $\bar{\alpha}_{0vi}$, q_{ci}, l_{ci}. Then (4) can be written in the form

$$\mathrm{Re}_{ci} = \frac{q_{ci} l_{ci}}{r\mu_c}. \tag{8}$$

In this method of processing and generalizing experimental data the main complication is determining the size of the condensate film (condensation zone length) l_{ci}, which is the defining dimension in the criterion Re_{ci}. If we consider the specifics of thermosiphon operation and construction, commencing from the fact that the condensation zone length depends on the thermal load supplied and the thermophysical characteristics of the heat-exchange agent, the values of l_{ci} can be determined from an expression obtained through analysis of the equations of vapor motion and energy of a variable mass flow:

$$l_{ci} = \frac{\omega_v^2}{g} \frac{(Q_s/Q_e)(\mu_v/\mu_c)\,\mathrm{Pr}^{0.65}}{(\rho_c/\rho_v)^{0.5}(l_a/d_{in})^{0.5}}. \tag{9}$$

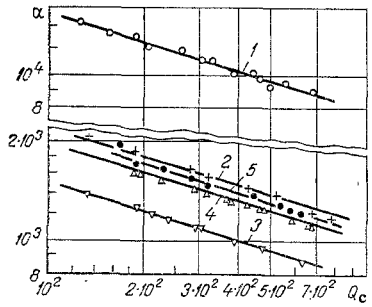

Fig. 1. Heat-liberation coefficient during condensation ($W/m^2 \cdot deg\ K$) versus thermal load removed (W): 1) water; 2) acetone; 3) benzol; 4) ethanol; 5) 50% ethanol +50% acetone.

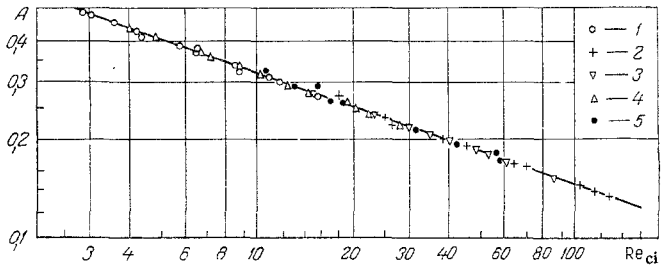

Fig. 2. Generalization of experimental data on various heat-exchange agents ($A = 0.695\ Re_{ci}^{-0.34}$): 1-5) see Fig. 1.

The vapor velocity w_v is calculated from

$$w = \frac{Q_e/r}{F_{v.c.}\ \rho_v}. \tag{10}$$

The ratio Q_s/Q_e characterizes the limiting condensation zone length, dependent on the sonic limit of thermal power transferable and the value of the thermal loading to the thermosiphon. The sonic limit of thermal loading can be found from the expression

$$Q_s = \sqrt{\frac{c_p}{c_v} RT_{sat} F_{v.c.}\ \rho_v r}. \tag{11}$$

The ratios μ_v/μ_c and ρ_v/ρ_c reflect the viscous and dynamic interaction of the heat-exchange agent at the phase boundary. The Prandtl number characterizes the physical properties of the condensate. An important quantity is the ratio l_a/d_{in}, the scale coefficient which considers the structural dimensions of the thermosiphon, the change in cross section and elongation of the adiabatic section due to increase in condensate film thickness as it moves downward, and the decrease in liquid level in the boiling zone. This quantity will appear in the equation for determination of ΔT_i. In accordance with the Nusselt solution for vapor condensation on a vertical wall and [9], the unknown temperature head will be given by

$$\Delta T_i = 1{,}08\, l_{ci} \left(\frac{q_{ci}}{A_1}\right)^{4/3} \frac{l_a}{d_{in}}, \tag{12}$$

where $A_1 = \sqrt[4]{\lambda_c^3 rg(\rho_c - \rho_v)/\nu_c}$. Since ΔT_i is related to l_{ci} and q_{ci}, which are calculated with consideration of l_a/d_{in}, in Eq. (12) this ratio will be a constant, found by approximation of the experimental data.

In the proposed method all unknown quantities are functions of two main parameters: the thermal load Q_e applied to the boiling section of the thermosiphon and the thermal power removed from the condenser Q_c. Both parameters Q_e and Q_c can be measured simply and accurately.

Data on condensation of vapor of various heat-exchange agents and mixtures were studied and generalized. Experiments were performed with thermosiphons having a condenser section length $l_{c.s.}$ = 190 and 250 mm and ratio l_a/d_{in} = 1-1.35. Heat-exchange agents used were: acetone, benzol, water, ethanol, and mixtures of these in the concentration ratio 15, 25, 50, 75%. A D-552 wattmeter was used to measure thermal load supplied, and the heat extracted was measured

from the flow rate and heating of the cooling fluid (water or air) surrounding the condenser.

All possible measures were taken in the experiments to eliminate uncondensed gases, including provision of a vacuum pump and three-way valve with manometer. Boiling and condensation temperatures were measured with thermocouples located along the thermosiphon axis within the cavity, while wall temperatures were determined using Chromel–Copel thermocouples and the compensation method. Moreover, the condensation temperature was determined from the pressure in the thermosiphon cavity, measured by a reference manometer.

For all heat-exchange agents and mixtures studied, the thermal load supplied was varied over the range 150-1050 W. The measured thermal power extracted from the condenser section comprised Q_c = 120-900 W. Values of the temperature difference ΔT_i, calculated with Eq. (12) for a given condensation zone were compared with the experimentally measured temperature difference.

In calculating the values of $\bar{\alpha}_{ovi}$, all physical constants of the heat-exchange agent were taken at saturation temperature. For the binary agents, mixtures of normal (nonassociating) liquids acetone–ethanol, water–acetone, the physical constants were calculated with the empirical formulas of [10] for the measured saturation temperature. For other mixtures the physical constants were determined experimentally, and also calculated with the formulas of [10].

Figure 1 presents the processing of experimental data in the form $\bar{\alpha}_{ovi} = f(Q_c)$ for the heat-exchange agents studied. For comparison, heat-liberation values for one of the binary mixtures studied are also shown (curve 5). As is evident from the figure, the experimental values of the heat-liberation coefficient are described by

$$\bar{\alpha}_{ovi} = C\,(Q_c)^{-0,3}, \tag{13}$$

corresponding to the well-known relationship for the process of vapor condensation.

The experimental points for the binary agent lie between the curves describing heat liberation of the single component heat-exchange agents acetone and ethanol. This indicates that at comparable values of heat liberation, for vapor condensation within a thermosiphon and convective heat liberation, the binary heat-exchange agent permits improvement of thermosiphon characteristics.

The results of generalizing the experimental data in the form $A = f(Re_{ci})$ are shown in Fig. 2. Values of the parameter A were calculated with Eq. (3), while Re_{ci} was calculated with Eq. (8). The physical constants were determined at the saturation temperature. The range of the criterion Re_{ci} in the experiments comprised 2.8-130.

By processing the experimental data an equation was obtained, which describes the process of condensation of the vapors of various heat-exchange agents in vertical thermosiphons:

$$A = 0,695\,Re_{ci}^{-0,34}. \tag{14}$$

It is evident from the figure that for the various heat-exchange agents, both single-component and binary (mixtures of normal, nonassociating liquids), the experimental data are described to a high degree of accuracy by Eq. (14), which can be used to calculate vapor condensation not only in thermosiphons but also in thermal tubes.

NOTATION

Re, Ga, Pr, Reynolds, Galileo, and Prandtl numbers; α, heat-liberation coefficient; λ, thermal conductivity coefficient; ν, μ, kinematic and dynamic viscosity coefficients; g, acceleration of gravity; ρ, density; r, heat of vaporization; T, temperature; q, thermal flux density; Q, thermal load; c_p, c_v, isobaric and isochoric heat capacities; R, ideal gas constant; ϵ_t, correction factor allowing for variation of physical parameters; l, length of boiling, condensation, or adiabatic sections; f, condensation zone surface; F, vapor channel surface; d, diameter; w, velocity. Subscripts: c, condensate; c.s., condensation section; e, evaporation; s, sound; v.c., vapor channel; in, inner; sat, saturation; a, adiabatic section; Nu, Nusselt.

LITERATURE CITED

1. V. L. Ivanov, "Study of heat exchange in a closed channel under natural convection conditions with change in the aggregate state of the heat-exchange agent," Izv. Vyssh. Uchebn. Zaved., Mashinostr., No. 1, 117-129 (1963).
2. B. S. Larkin, "An experimental study of the two-phase thermosiphon tube," Eng. J. (Can), No. 4, 8-9 (1971).
3. S. P. Andreev, "Study of heat exchange during liquid phase conversions in a closed channel," Teploénergetika, No. 7, 65-69 (1972).
4. Z. R. Gorbis and G. A. Savchenko, "Study of the effect of uncondensed impurities on efficiency of heat transfer by an evaporative thermosiphon," Teploénergetika, No. 10, 70-73 (1973).
5. A. N. Alabovskii, M. K. Bezrodnyi, and V. F. Moklyak, "Study of heat exchange during vapor condensation in vertical thermosiphons," Izv. Vyssh. Uchebn. Zaved., Énerget., No. 7, 61-67 (1979).

6. V. P. Isachenko, Heat Exchange during Condensation [in Russian], Énergiya, Moscow (1977).
7. S. S. Kutateladze, Fundamentals of Heat Exchange Theory [in Russian], Atomizdat, Moscow (1979).
8. N. I. Mirmov and I. G. Belyakova, "Some results of a study of thermosiphons with binary heat-exchange agents," Inzh.-Fiz. Zh., 38, No. 5, 934 (1980).
9. E. M. Novokhatskii and A. N. Gorovoi, "Internal thermal resistance of a thermosiphon," Izv. Vyssh. Uchebn. Zaved., Énerget., No. 5, 87-92 (1978).
10. S. Bretshnaider, Properties of Gases and Liquids [in Russian], Khimiya, Leningrad (1966).

LOCAL HEAT EXCHANGE IN THE FILM CONDENSATION OF A STATIONARY VAPOR ON A VERTICAL SURFACE

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A method is described for calculating the distributions of the heat-transfer coefficient and heat flux over the height of a condensation surface under all regimes of condensate film flow. Data on these distributions are also presented.

In designing condensers and calculating vapor condensation, one is usually dealing with mean values of the heat-transfer coefficient and heat flux over the entire height of the condensation surface. However, in certain cases the designer may be able to use such information as the distribution of local heat flux over the height of the condenser. These data could be valuable, for example, in organizing efficient delivery of the vapor to the condensation surface.

In the film condensation of vapor, local heat flux can be expressed as follows:

$$q_c = (T_s - T_f) \Big/ \left(\frac{1}{\alpha} + R_c + \frac{1}{\alpha_1} \right). \tag{1}$$

The change in R_c and $1/\alpha_1$ over the height of the condensation surface can often be ignored in a first approximation. Thus, the change in q_c depends primarily on the change in T_f and α. For T_f, we can write

$$T_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_0^{x_s} \varepsilon_q dx_s, \tag{2}$$

if the heat-removing medium and condensation film constitute a forward flow. On the other hand,

$$T_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_1^{x_s} \varepsilon_q dx_s, \tag{3}$$

if there is a counter flow.

Since $\int_0^1 T_f dx_s = \overline{T}_f$, then Eqs. (2) and (3) respectively yield

$$\overline{T}_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_0^1 dx_s \int_0^{x_s} \varepsilon_q dx_s, \tag{4}$$

or

$$\overline{T}_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_0^1 dx_s \int_1^{x_s} \varepsilon_q dx_s. \tag{5}$$

According to [1], the heat-transfer coefficient for a nonisothermal surface should be averaged in conformity with the law

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