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REVIEW

HEAT TRANSFER IN CONDENSATION OF VAPOR MOVING INSIDE VERTICAL TUBES

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A review of works devoted to the study of heat transfer in condensation of moving vapor in cocurrent flow of vapor and film is presented. Generalization of a wide range of experimental data obtained by different authors showed that in wave regimes of film flow conditions take place under which an increase in vapor velocity does not lead to enhancement of heat transfer, as compared to heat transfer in condensation of motionless vapor. In turbulent film flow, an intense entrainment of the film from the crests of waves into the vapor core begins when $We > We_{cr}$*, thus leading to considerable enhancement of heat transfer.*

Heat exchangers with vapor, which condenses inside tubes, being a heating medium, are widely used in the oil, chemical, refrigeration, and food branches of industry and in power engineering. A principal advantage of this structure of a heat exchanger is as high as is wished pressure inside the tubes without increasing the thickness of the walls of the apparatus casing. A review of works on heat transfer in condensation inside tubes has been detailed in the monographs [1–3] and papers generalizing experimental investigations [4–6]. We consider only the problem of cocurrent vapor and liquid flow in condensation of vapor without noncondensable admixtures.

According to the relation between gravity forces and forces of friction, six flow modes are considered in [1]:

1) laminar film flow with a prevailing effect of gravity forces (g; lam);

2) laminar film flow with a commensurable effect of gravity forces and forces of interphase friction (g; U_v; lam);

3) laminar film flow with a prevailing effect of forces of interphase friction (U_v; lam);

4) turbulent film flow with a prevailing effect of gravity forces (g; t);

5) turbulent film flow with a commensuarable effect of gravity forces and forces of interphase friction (g; U_v; t);

6) turbulent film flow with a prevailing effect of forces of interphase friction (U_v; t).

However, as shown by the studies of the hydrodynamics of draining films, several regimes have been omitted here, viz.:

7) wave film flow with a prevailing effect of gravity forces, (g, b);

8) wave film flow with a commensuarable effect of gravity forces and forces of interphase friction (g; U_v; b);

9) wave film flow with a prevailing effect of forces of interphase friction (U_v, b).

Most difficult in this gradation is the determination of the parameter at which one mode is replaced by another. As will be shown below, laminar-to-turbulent mode transition and replacement of a laminar film flow by a wave and a wave flow by a turbulent flow are functions of the crossflow of substance, which is determined by friction on the vapor–film phase interface. Crossflow is variable (even in the experiments of the same author) due to variation of the specific heat flux over the tube height and vapor velocity.

In the most general formulation, account of the effect of vapor flow is reduced to determination of friction on the vapor–film phase interface. If the film surface is assumed to be smooth, friction and heat transfer can be determined by combined solution of the equations of motion, energy, and continuity for the liquid and vapor phases with conjugate boundary conditions.

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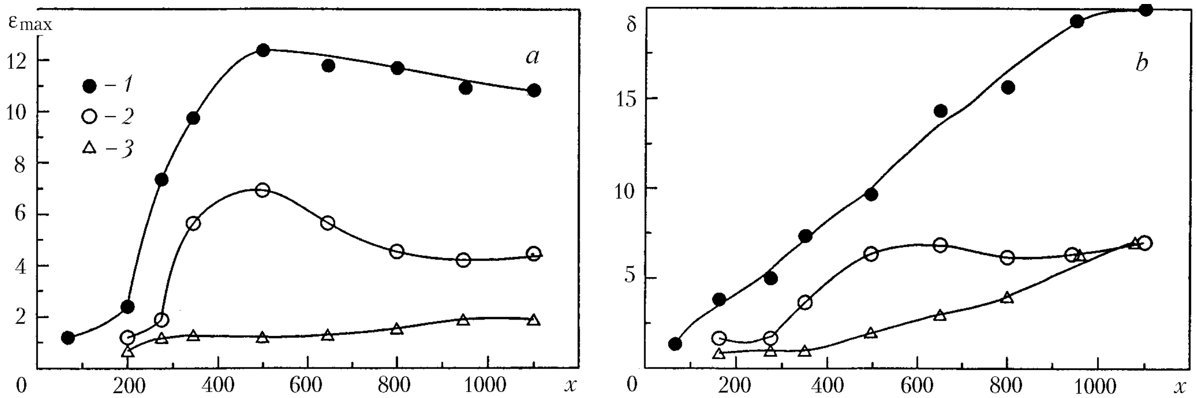

Fig. 1. Change in the cross-section maximum value of intensity of velocity fluctuations ε_{max} (a) and change in the boundary-layer thickness δ over the plate length (b) [8]: 1) without suction; 2) in uniform suction $S_q = 5 \cdot 10^{-3}$; 3) in stepwise suction and at a total value of $S_q = 5 \cdot 10^{-3}$. ε_{max}, %; δ, x, mm.

If the film surface is wave, the mechanism of interaction between the vapor flow and film becomes extremely complex and resists theoretical analysis. Even the knowledge of local values of interface friction does not suffice to solve the problem. Additional information about the effect of the process of condensation and vapor flow on critical parameters of film flow is necessary. Moreover, some facility with estimating a fraction of liquid entrained from the surface of the film condensate into the vapor flow is desirable.

The problem of condensation of moving vapor is similar to the problem of gas flow past a permeable surface. The presence of crossflow of a substance tangibly changes the laws governing a flow past a body, i.e., in our case, the laws governing a vapor flow past a film. Schlichting [7] gave the solution of the Navier–Stokes equations with an asymptotic profile of suction in a laminar flow past a plate. It is shown that the tangential stress on the wall is equal to

$$\tau_w = \mu_v \left(\frac{\partial U}{\partial y} \right) = \rho_v (-\vartheta)\, U_v \tag{1}$$

and, consequently, is independent of the viscosity of the vapor. Suction changes laminar-to-turbulent layer transition considerably. In suction, the critical Re number for vapor which corresponds to laminar-to-turbulent layer transition can increase by more than two orders. Figure 1 presents results of the experimental investigation of laminar-to-turbulent transition in the presence of suction on the plate [8]. It is shown that the intensity of velocity fluctuations and the boundary-layer thickness are substantially dependent on suction and the way of its organization. In stepwise suction, its coefficient was the largest at the beginning of the plate.

Curiously, the process of suction in condensation qualitatively resembles stepwise suction in the experiment presented. In condensation inside a tube, a maximum value of heat flux is observed in the initial section where the liquid film is minimum.

Thus, the presence of crossflow of a substance $S_q = \dfrac{q}{r\rho_v U_v}$ leads to considerable changes in the critical Reynolds numbers for a vapor and a film, which characterize transition regimes. The critical Re number for vapor corresponding to laminar-to-turbulent layer transition can increase by two orders [8]. For liquid, the Re number for a film characterizing laminar-to-wave transition increases to about tenfold by the data of the authors of [9, 10] and to about 100-fold by the data of [11].

Dependences Generalizing Experimental Data. The formulas suggested by different author for determining the coefficient of heat transfer in condensation inside tubes are presented in [1, 3, 5]. Thus, for example, the well-known Boiko–Kruzhilin dependence in complete condensation of vapor has the form [12]

$$\overline{\mathrm{Nu}} = 0.035 \mathrm{Re}_{\mathrm{liq}}^{0.8} \mathrm{Pr}_{\mathrm{liq}}^{0.4} \left[1 + 0.315 \left(\frac{\rho_v}{\rho_{\mathrm{liq}}} \right)^{0.67} \right]. \tag{2}$$

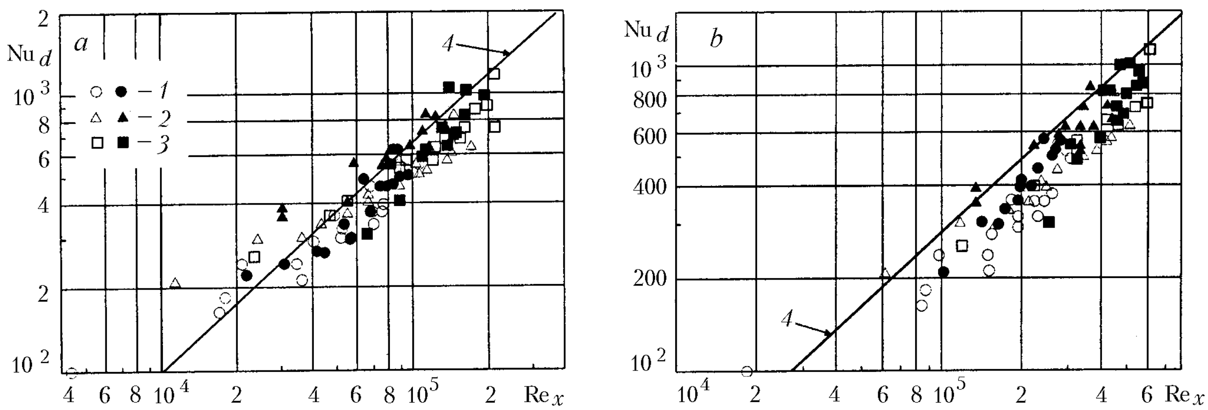

Fig. 2. Dimensionless coefficient of heat transfer in condensation of high-speed vapor flow [19]: 1) $35 \cdot 10^4$; 2) $45 \cdot 10^4$; 3) $55 \cdot 10^4$ kg/(m^2·h); 4) calculation by the Cavallini–Zeechin relation [16] (a); 4) calculation by the Boiko–Kruzhilin relation [12] (b), $\mathrm{Re}_x = \frac{md}{F\mu_{\mathrm{liq}}}\left\{(1-x_2)+x_2\frac{\rho_{\mathrm{liq}}}{\rho_{\mathrm{v}}}\right\}^{0.625}$. Open symbols — horizontal tube, filled symbols — vertical tube. Pr = 2.0.

Expressions from [5, 13–18] do not differ principally from (2). In [3], these are given in Table 3.1. These dependences are obtained under the following assumptions:

a) wave formation is absent on the phase interface;

b) a universal velocity profile is used as for a one-phase liquid;

c) equality of tangential stresses on the wall and phase interface is assumed;

d) the coefficient of resistance on the phase interface is taken as in a one-phase flow or with account of roughness, where the mean film thickness is taken to be the roughness element;

e) stall of droplets from the film surface by vapor flow is disregarded.

In the formulas mentioned, physical special features related to the effect of crossflow discussed above are not taken into account. This means that each special feature can be considered as approximate and suitable for describing the process within the studied range of parameters. However, the validity of this assumption is not specified, as a rule, by any limit, which is incorrect in principle. By the value of exponents at determining criteria we can assume that dependences of the type of (2) describe heat transfer only in a turbulent mode of condensate film flow.

In incomplete condensation of vapor, expression (2) takes on the form

$$\frac{\overline{\mathrm{Nu}}_{x_2}}{\overline{\mathrm{Nu}}_{x_2=0}} = \frac{\left(\frac{\rho_{\mathrm{liq}}}{\rho_{\mathrm{v}}}\right)^{1/2} + \left[1 + x_2\left(\frac{\rho_{\mathrm{liq}}}{\rho_{\mathrm{v}}} - 1\right)\right]^{1/2}}{(1-x_2)^{0.8}\left[\left(\frac{\rho_{\mathrm{liq}}}{\rho_{\mathrm{v}}}\right)^{1/2} + 1\right]} . \tag{3}$$

Here the dimensionless formula which describes heat transfer in a turbulent one-phase liquid flow serves as a scale. Comparison is made at $\mathrm{Re}_{\mathrm{eq}}$ = idem. Here $\mathrm{Re}_{\mathrm{eq}} = \overline{U}_{\mathrm{liq}} d_{\mathrm{eq}}/\nu = 4\mathrm{Re}_{\mathrm{liq}}$. In Fig. 2, taken from [19], experimental data in condensation of water vapor in the tube with d = 14 mm is compared with the calculation by the Boiko–Kruzhilin [12] and Cavallini–Zeechin [16] formulas. We can note that in high-rate condensation of vapor the results of the experiments in vertical and horizontal tubes coincide and the calculation by the Cavallini dependence describes experimental data satisfactorily.

The empirical relation of Borishanskii et al. [20] allows, in the present author's opinion, description of heat transfer for all modes of film flow. This relation has the form

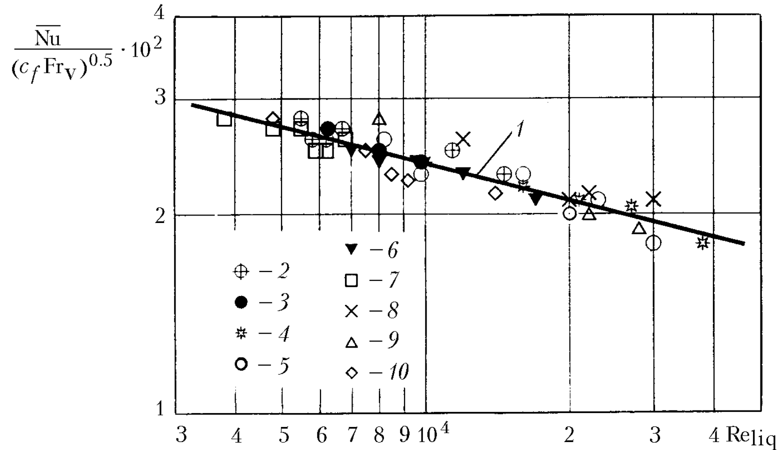

Fig. 3. Heat transfer in complete condensation of vapor under conditions of the prevailing effect of interphase friction [11]: 1) calculation by (7); 2) P = 4.9; 3) 2.9; 4) 8.7; 5) 7.0; 6) 5.3; 7) 3.0; 8) 8.8; 9) 5.9; 10) 2.45 MPa [2 and 3) vertical tube, d = 10 mm; L = 3.0 m [24]; 4–7) horizontal tube; d = 10 mm, L = 4.0 m [23]; 8 and 9) horizontal tube, d = 13 mm, L = 12 m [22]; 10) horizontal tube, d = 10 mm, L = 2.5 m [22]].

$$\overline{\mathrm{Nu}}\,\mathrm{Fr}_0 = f\left(\mathrm{Fr}_0;\ \mathrm{Ga}^{1/3};\ \frac{\mathrm{Pr_{liq}}}{\mathrm{Pr_v}}\right) \tag{4}$$

or

$$\frac{\alpha}{\lambda}\frac{U_{\mathrm{liq}}^2}{g} = f\left(\frac{U_{\mathrm{liq}}^2}{(g\nu)^{2/3}}\ \frac{\mathrm{Pr_{liq}}}{\mathrm{Pr_v}}\right).$$

Here $U_{\mathrm{liq}} = 4qL/(r\rho_{\mathrm{liq}}d)$ is the velocity of condensate at the tube outlet.

Processing of the experiments in the coordinates (4) allowed the authors of [20] to obtain the computation formula

$$\overline{\mathrm{Nu}}\,\mathrm{Fr}_0 = 0.1\,\sqrt{7A^{1.7} + 0.2A^{2.8}}\,, \tag{5}$$

where $A = \mathrm{Fr}_0\,\mathrm{Ga}^{1/3}(\mathrm{Pr_{liq}}/\mathrm{Pr_v})$.

Equation (5) holds when $0.3 \le A \le 400$. Borishanskii et al. [20] state that this expression allows description of experimental points with a scatter of ±30%, including laminar-to-turbulent film-flow transition. The dependence presented in [20] has no clear physical interpretation as yet.

Another approach to generalization of data on condensation of vapor moving at a high velocity was suggested by Rifert [11]. A fundamental difference of this approach is the determination of friction on the phase interface. The coefficient of friction is determined with account of suction in the boundary layer. According to Kutateladze and Leont'ev, for a turbulent boundary layer with suction we have [21]

$$c_f = c_{\mathrm{fr}}\,\frac{(1 - 0.25b)^2}{(1 + 0.25b)^{0.2}}\,. \tag{6}$$

For the prevailing effect of interphase friction over the greatest length of the tube the experimental data for complete condensation of vapor inside horizontal [22, 23] and vertical [24] tubes are described by the dependence [11]

$$\mathrm{Nu} = 0.117\,(c_f\,\mathrm{Fr_v})^{0.5}\,\mathrm{Re_{liq}^{-0.17}}\,. \tag{7}$$

In Fig. 3, taken from [11], the dimensionless formula (7) generalizes experimental data satisfactorily.

The Froude number and the coefficient of friction are determined by the inlet velocity of vapor and the Re number for a film at $x = L$. In [11] it is stated that due to the strong effect of interphase friction on condensate-film flow and entrainment of liquid by vapor, the law governing laminar film condensation holds within a wide range of Re numbers, up to $\mathrm{Re_{liq}} = 4{\cdot}10^4$.

At such Reynolds numbers the film is likely to be called pseudolaminar, where the role of waves is leveled due to entrainment and the effect of suction.

The problem of heat transfer in condensation of moving vapor on a vertical surface with a laminar regime of film flow was solved by Shekriladze [25, 26] and Fujii and Uehara [27] most properly. The following assumptions were made in [27]:

a) inertia and convective terms in the equation for a laminar liquid film can be omitted;

b) physical properties of the film are taken at the saturation temperature; thermal resistance on the phase interphase can be disregarded;

c) temperature of the cooled surface is uniform, i.e., $t_{\mathrm{w}} = \mathrm{const}$;

d) a laminar condensate film is streamlined by vapor flow with a laminar boundary layer.

The initial system of equations of motion, energy, and continuity for vapor and a film with the conjugate boundary conditions is solved numerically. The numerical solutions are approximated by the following dimensionless relations:

1. The case of motionless vapor: a local value of the Nusselt number is determined as

$$\mathrm{Nu}_x = \left(\frac{\mathrm{Ga}_x \mathrm{Pr_{liq}} \mathrm{K}}{4}\right)^{1/4} ; \tag{8}$$

the Nusselt formula for describing mean heat transfer in a laminar film flow has the form

$$\overline{\mathrm{Nu}} = \frac{4}{3}\left(\frac{\mathrm{Ga}_L \mathrm{Pr_{liq}} \mathrm{K}}{4}\right)^{1/4} . \tag{9}$$

2. It follows at $g = 0$ and a high velocity of vapor when $\mathrm{R}/(\mathrm{Pr_{liq}K}) << 1.0$ (small Δt, dense vapor) that

$$\frac{\mathrm{Nu}_x}{\sqrt{\mathrm{Re}_x}} = 0.45\left(\frac{\mathrm{R}}{\mathrm{Pr_{liq}K}}\right)^{1/3} . \tag{10}$$

For $\mathrm{R}/(\mathrm{Pr_{liq}K}) << 10$ (small vapor densities, large Δt) we have

$$\frac{\mathrm{Nu}_x}{\sqrt{\mathrm{Re}_x}} = 0.5 . \tag{11}$$

A similar result was obtained much earlier by Cess [28].

When $\mathrm{R}/(\mathrm{Pr_{liq}K}) << 10$, a general expression for calculation of the local Nusselt number Nu_x

$$\frac{\mathrm{Nu}_x}{\sqrt{\mathrm{Re}_x}} = 0.45\left(1.2 + \frac{\mathrm{Pr_{liq}K}}{\mathrm{R}}\right)^{1/3} . \tag{12}$$

can be obtained. The mean Nusselt number is determined as

$$\frac{\overline{\mathrm{Nu}}}{\sqrt{\mathrm{Re}'_{\mathrm{v}}}} = 0.9\left(1.2 + \frac{\mathrm{Pr_{liq}K}}{\mathrm{R}}\right)^{1/3} . \tag{13}$$

3. The case of combined effect of gravity forces and forced convection:

$$\frac{\mathrm{Nu}_x}{\sqrt{\mathrm{Re}_x}} = M\left(1 + \frac{Z\mathrm{Pr}_{\mathrm{liq}}\mathrm{K}}{4M^4}\right)^{1/4}, \tag{14}$$

where $M = 0.45\left(1.2 + \frac{\mathrm{Pr}_{\mathrm{liq}}\mathrm{K}}{\mathrm{R}}\right)^{1/3}$ is the dimensionless complex.

The averaged Nusselt number is found as

$$\overline{\mathrm{Nu}} = \left\{0.656\left(1.2 + \frac{\mathrm{Pr}_{\mathrm{liq}}\mathrm{K}}{\mathrm{R}}\right)^{4/3} \mathrm{Re}_{\mathrm{v}}^{\prime 2} + 0.79\mathrm{Ga}_{\mathrm{liq}}\mathrm{Pr}_{\mathrm{liq}}\mathrm{K}\right\}^{1/4}. \tag{15}$$

If we take $\chi = \left(1.2 + \frac{\mathrm{Pr}_{\mathrm{liq}}\mathrm{K}}{\mathrm{R}}\right)^{1/3}$, then

$$\overline{\mathrm{Nu}} = \left(0.656\chi^4 \mathrm{Re}_{\mathrm{v}}^{\prime 2} + 0.79\mathrm{Ga}_{\mathrm{liq}}\mathrm{Pr}_{\mathrm{liq}}\mathrm{K}\right)^{1/4}. \tag{16}$$

Then, in [27], experimental data of different authors obtained in condensation of water vapor and organic liquids are compared with the values of the coefficients of heat transfer calculated by the suggested relations.

At some calculated parameters the coincidence of calculation and experiment is $\pm$20%. However, when $\overline{\mathrm{Nu}} > 2 \cdot 10^4$, the difference between experimental and calculated data is up to 70%, which, in the opinion of Fujii and Uehara [27], is due to turbulence in a liquid film.

Heat Transfer in Condensation of Motionless Vapor on a Vertical Surface. In vapor condensation inside tubes the regimes where vapor can be assumed motionless are absent. However, a small velocity of vapor cannot exert a tangible effect on heat transfer.

Analysis of the data on condensation of motionless vapor is of importance in principle, since:

1) it indicates minimum values of $\overline{\mathrm{Nu}}$ numbers in condensation of this substance on a vertical surface;

2) in processing of experiments on heat transfer in condensation of moving vapor in the coordinates

$$\mathrm{Nu}^*/\mathrm{Nu}_0^* = f(\mathrm{Re}_{\mathrm{liq}}) \tag{17}$$

it allows one to reveal incorrect measurements or calculations in condensation of pure vapor, which appear to lie below unity. (Nu_0^* is the experimentally found Nusselt number in condensation of motionless vapor on a vertical surface.)

In concurrent flow of a film and vapor, heat transfer can be enhanced to a certain extent compared to the case of motionless vapor. The number Nu_0^* is the scale by which, provided $\mathrm{Re}_{\mathrm{liq}}$ = idem and $\mathrm{Pr}_{\mathrm{liq}}$ = idem, one can judge the possibility of heat-transfer enhancement due to friction on the phase interface.

Results of the experimental investigations of heat transfer in condensation of motionless R21 vapor ($\mathrm{Pr}_{\mathrm{liq}}$ = 3.5) and water vapor ($\mathrm{Pr}_{\mathrm{liq}}$ = 1.75, $\mathrm{Pr}_{\mathrm{liq}}$ = 1.12) on a vertical surface are given in [29] and [30], respectively. The data obtained in these experiments are shown in Fig. 4. We should note that in [29] results of the measurements in condensation of R21 are compared with the data of different studies of condensation of other refrigerants and are in good agreement with them; at small Re numbers these data agree with the Nusselt theory.

In [31], the experiments of Kutateladze and Shrentsel' on condensation of water vapor, which were published in [30], are compared with the data of different authors and also show good agreement with them.

The following special features engage our attention in analysis of the data presented in Fig. 4. As the Reynolds number for film changes within $100 \le \mathrm{Re} \le 1000$ the Nusselt number Nu^* remains virtually constant (wave regime of film flow). In a turbulent regime of film flow, distinct stratification of data on the Prandtl numbers for liquid and enhancement of heat transfer with an increase in the Reynolds number are observed. The same figure gives comparison of the experimental data with those calculated by the relation

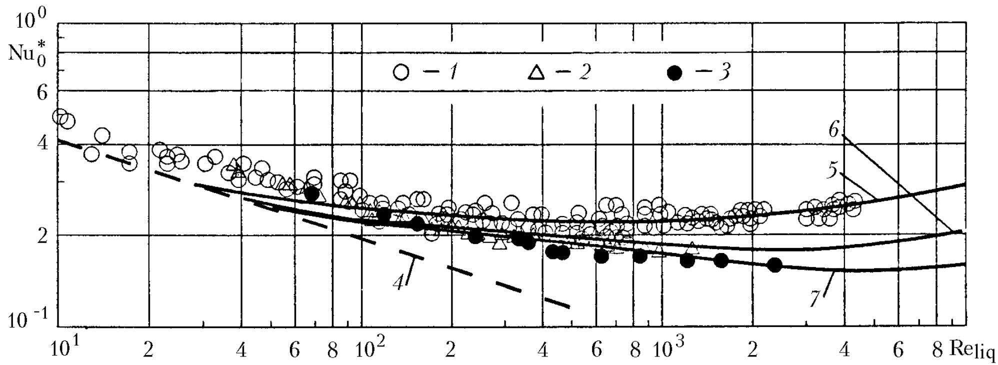

Fig. 4. Heat transfer in condensation of motionless vapor on a vertical surface: 1) Pr = 3.5, Re = 10–4250 [29]; 2) Re = 25–1270; 3) Re = 25–2340 [30]; 4) calculation by the Nusselt theory; 5–7) calculation by (18) [5] Pr = 3.5; 6) 1.75; 7) 1.12]; 1) R21; 2 and 3) water vapor.

$$\overline{\mathrm{Nu}^*} = \overline{\mathrm{Nu}^*_{\mathrm{lam}}}\,\frac{\mathrm{Re}^*}{\mathrm{Re}} + \overline{\mathrm{Nu}}_{\mathrm{lam.wave}}\,\frac{\mathrm{Re}_{\mathrm{cr}} - \mathrm{Re}^*}{\mathrm{Re}} + \overline{\mathrm{Nu}_{\mathrm{t}}}\,\frac{\mathrm{Re} - \mathrm{Re}_{\mathrm{cr}}}{\mathrm{Re}}\,. \tag{18}$$

Here Re^* is the boundary of the laminar-to-wave flow transition; $\mathrm{Re}_{\mathrm{cr}}$ is the critical Reynolds number for a film which corresponds to the wave-to-turbulent transition of film flow;

$$\overline{\mathrm{Nu}^*_{\mathrm{lam}}} = 0.925\mathrm{Re}^{-1/3}\,; \tag{19}$$

the Nusselt formula for describing averaged heat transfer in laminar film flow on the vertical surface is

$$\overline{\mathrm{Nu}}_{\mathrm{lam.wave}} = 0.527\mathrm{Ka}^{-1/15}\,. \tag{20}$$

Expression (20) is obtained assuming the thermal resistance of the film in the wave mode of flow to be determined by the "residual" film thickness.

In [32], the dependence of Re^* on liquid properties

$$\mathrm{Re}^* = 2.3\mathrm{Ka}^{1/15}\,. \tag{21}$$

was found. Here it was found that in the region $\mathrm{Re}^* \le \mathrm{Re} \le \mathrm{Re}_{\mathrm{cr}}$ the "residual" film thickness is virtually independent of the Re number for a film. The wave-to-laminar flow transition is determined as

$$\mathrm{Re}_{\mathrm{cr}} = 35\mathrm{Ka}^{1/5}\,. \tag{22}$$

In order to describe heat transfer in turbulent film flow, Kutateladze [33] used the model where eddy viscosity is piecewise approximated by the relations

$$0 \le \eta \le 6.8\,, \quad \mu_{\mathrm{t}} = 0\,;$$

$$6.8 \le \eta \le 0.2\,(\eta_\delta - 6.8)\,, \quad \overline{\mu_{\mathrm{t}}} = 0.4\,(\eta - 6.8)\sqrt{1 - \frac{\eta}{\eta_\delta}}\,;$$

$$0.2\,(\eta_\delta - 6.8) < \eta < \eta_\delta\,, \quad \overline{\mu_{\mathrm{t}}} = 0.08\,(\eta_\delta - 6.8)\sqrt{1 - \frac{\eta}{\eta_\delta}}\,.$$

The dimensionless film thickness is related to the Re number as:

$$\mathrm{Re} = \frac{1}{\eta_\delta} \int_0^{\eta_\delta} \varphi(\eta)\, d\eta = \frac{1}{\eta_\delta} \int_0^{\eta_\delta} \frac{(\eta_\delta - \eta)^2}{1 + \mu_t(\eta)}\, d\eta ,$$

where η = 6.8 is the conventional boundary of the viscous sublayer correlating with the Prandtl–Kármán constant κ = 0.4. Local and mean Nusselt numbers in turbulent film draining are calculated by the relations

$$\mathrm{Nu}_t^* = \eta_\delta^{1/3} \left(\int_0^{\eta_\delta} \frac{d\eta}{1 + \mathrm{Pr}\mu_t} \right)^{-1} , \tag{23}$$

$$\overline{\mathrm{Nu}_t^*} = \left(\eta_\delta - \eta_{\delta_{cr}} \right)^{-1} \int_{\eta_{\delta_{cr}}}^{\eta_\delta} \mathrm{Nu}_t^* d\eta_t . \tag{24}$$

In [33], formulas (23) and (24) are compared with experimental data on condensation of vapor of water, refrigerants R12, R21, and R22, ethanol, and nitrogen.

Analysis of Experimental Data on Vapor Condensation in a Tube. This analysis requires a number of general comments:

1. If the authors give only the values of mean heat flux and mean (over the tube length) temperature head and the experiments are conducted in complete condensation of vapor, the data are not amenable to strict analysis. All the above-mentioned regimes of film and vapor flow can take place in such experiments. A distinct boundary between these regimes is unknown; consequently comparison of them with the computational models is usually incorrect with the probability of those assumptions which are made in the computational model of the author of the work under analysis.

2. In condensation of vapor, especially water vapor, researchers often do not give an initial concentration of air in vapor. In vapor condensation it can be small in the outlet section of the tube. This will pronouncedly distort the determination of the heat flux and temperature head, since there appears diffusion resistance which is in no way allowed for in the models mentioned. In [34], it is shown that a noticeable effect of air in condensation of moving vapor manifests itself at a concentration of the latter higher than 0.01%. In analysis of experiments on condensation of water vapor one must keep in mind the possible influence of it on heat transfer.

3. At high velocities of vapor a noticeable entrainment of liquid from the film to the vapor flow takes place. In [35], it is shown that heretofore correctly determined dimensionless parameters, which could estimate the effect of this phenomenon on heat transfer, are unknown. Thus, in [27], the deviation of experimental data from calculated data at some parameters of the experiment can be explained, in the authors' opinion, by the origination of turbulence in the film. However, with the same probability this can be related to entrainment of a portion of the liquid to the vapor flow.

Heat transfer in condensation of moving vapor inside vertical tubes was experimentally studied in [14–16, 22, 24, 36–50]. The experiments were conducted on condensation of vapor of different substances, at different pressures, and in complete and partial condensation of vapor in tubes of different lengths and diameters. The works mentioned above can conventionally be divided into four groups:

a) works which virtually are inaccessible to readers in Russia, e.g., [36, 37];

b) works which do not give any objective information, except for empirical dimensionless relations by which the experiments are processed, e.g., [38, 39];

c) works which do not present all determining parameters (pressure or diameter of the experimental tube, etc.) [40–43];

d) works where results of the measurements are given (in full volume or partially) in the form of tables or graphs which are amenable to further analysis [16, 24, 44–50].

Table 1 lists the works where the results of measurements are amenable to analysis.

TABLE 1. Main Parameters in Condensation of Vapor Moving inside Tubes

Ref.	Working body	Tube characteristics		Characteristics of parameters	P, MPa	$q \cdot 10^{-3}$, W/m^2	U_v, m/sec	Re_{liq}
		d, mm	L, m					
[44]	Water vapor	40	1.2	Averaged	0.1	47–233	10–80	67–400
[45]	Benzene	18.9	0.907	»	0.1	106–147	19–32	75–104
[46]	Water vapor	10	2.2; 3.2	»	1–9	100–1000	11–50	700–8000
[47]	»	9.88	1.026	Local	0.11	70–400	20–95	100–550
[16]	R11	20	1.7	Averaged	0.11–0.15	14–100	11.5–43	320–2500
[48]	Propane	8.0	1.0	»	0.5–0.9	4–35	0.26–4.8	87–940
[24]	Water vapor	10; 20	1.5; 3,0	»	0.8–7.0	30–680	0.7–31	190–10 000
[49]	»	30	3.0	Local	0.1	10–400	10.4–15.1	50–2000
[50]	Water vapor and isopropanol–water mixture	30	3.0	»	0.1	4.5–430	4.8–15.1	20–2300

We must note that the works given in this table were performed according to different techniques. Thus, in [24, 45, 46, 48], complete or almost complete condensation of vapor took place in the experimental tube. In these publications, its own value of velocity at the inlet and virtually zero velocity of vapor at the tube outlet correspond to each point on the graph or in the table. Heat-flux variation results in changes in the Reynolds number for a film at the tube outlet and vapor flow at its inlet. The graphs or tables of the cited papers give the length-mean values of specific heat flux and heat-transfer coefficient. If the velocity of vapor was not indicated, its value was determined by the relation

$$U_v = \frac{4qL}{\rho_v r d} . \tag{25}$$

In [16, 44], the velocity of vapor at the inlet remained constant for each series of experiments and values of the specific heat flux and Re number for a film varied due to variation of the vapor velocity at the tube outlet. In these works, the authors give averaged values of the parameters in the experiment.

In [47, 49, 50], local values of specific heat flux and the heat-transfer coefficient for a short tube section are presented. However, in the experiments by Isachenko et al. [47], complete condensation of vapor occurred and values of specific heat fluxes and velocities of vapor changed greatly at the inlet to each local section of the tube. In the experiments of [49, 50], the velocity of vapor at the inlet to the lower part of a 200-mm-long tube, where local heat flux and the heat-transfer coefficient were determined in each series of experiments, was kept constant. The Reynolds number for a film increased due to variation of the reflux density in the pre-inserted uncooled section of the tube.

Most experiments were conducted in condensation of water vapor. In [20, 24], experiments were made with substantial variation of pressure (P = 0.76–6.86 MPa) in tubes of different lengths and diameters. The authors of these works gave the results of the experiments in the form of tables where all the experimental parameters are listed.

Figure 5 presents the results of processing of these data in coordinates (17) (experiments were conducted on complete condensation of vapor in the experimental tube). Its own value of vapor velocity at the inlet to the experimental tube corresponds to each point in the figure. The inlet velocity of vapor increases with increase in the Reynolds number for a film.

The results of experimental data processing given in Fig. 5a cast some doubt upon their authenticity. In the overwhelming majority of experiments conducted in tubes with d = 20 mm, the ratio Nu^* / Nu_0^* is less than unity and is virtually independent of the velocity of vapor at the inlet to the experimental section. Such a result was obtained in one series of experiments on condensation in a tube with d = 10 mm and P = 6.86 MPa. These results are given in the same figure.

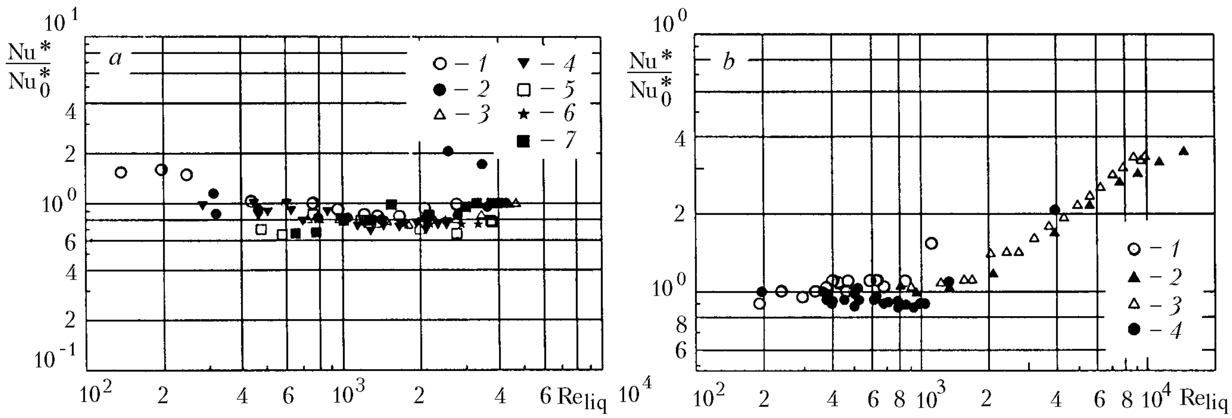

Fig. 5. Relative variation of heat transfer as a function of Re_{liq} in condensation of moving water vapor in the experiments of [24]: a: 1) P = 2.94 MPa; 2) 4.9; 3) 6.86 (d = 19.3 mm, L = 1.5 m); 4) 2.94; 5) 4.51; 6) 6.87 (d = 20 mm, L = 3 m); 7) 6.86 (d = 10 mm, L = 1.5 m); b: 1) P = 0.79 MPa; 2) 2.94 (d = 10 mm, L = 1.5 m); 3) 2.94; 4) 4.9 (d = 10 mm, L = 3 m).

TABLE 2. Local Nusselt Numbers in Condensation of Moving [50] and Motionless [30] Vapor

U_v, m/sec	Re_{liq}			
	200	300	500	1000
4.8 [50]	0.195	0.19	0.18	0.19
10.4 [50]	0.2	0.195	0.185	0.19
15.1 [50]	0.205	0.2	0.19	0.2
0 [30]	0.21	0.2	0.195	0.19

It is seen from the analysis of the data presented in Fig. 5b that a rather large number of experiments were conducted under conditions virtually without the effect of the vapor velocity, since the ratio of the Nusselt numbers in condensation of moving and motionless vapor is close to unity ($Re_{liq} \le 2 \cdot 10^3$). Only the results of measurements at a high velocity of vapor at the tube inlet indicate considerable enhancement of the heat-transfer process due to the effect of vapor on the condensate film (friction on the phase interface and possible entrainment of film to the vapor flow). Further processing of the results of measurements [20, 24] was done only for the experiments shown in Fig. 5b.

In analysis of the data of [50], we should focus our attention on the absence of influence of the vapor velocity on the local Nusselt number with variation of the Reynolds number for a film within the range $200 \le Re_{liq} \le 10^3$. Table 2 presents the values of local Nusselt numbers for different velocities of vapor which changed from 4.8 to 15.1 m/sec at Re_{liq} numbers from 200 to 1000. Here, the Nusselt numbers in condensation of motionless vapor [30] are given for comparison. It is seen from the table that Nusselt numbers obtained in the experiments of [50] only slightly depend on the velocity of vapor, at the inlet to the tube and practically do not differ from the Nusselt numbers in condensation of motionless vapor given in [30].

If we assume that in the wave regime of film flow the "residual" thickness is thermal resistance and heat transfer fully depends on it only, then the result given in Table 2 cannot be taken as unexpected. This only means that the "residual" thickness of the film does not change with the velocity of vapor within the indicated range and, consequently, the thermal resistance of the film measured in the experiment does not depend on the velocity of vapor. This assumption allows one to compare the local Nusselt number obtained in [50] with the averaged Nusselt number from the Kutateladze–Shrentsel' experiments [30] on condensation of motionless vapor on the vertical surface.

For comparison, the results of such processing of the experiments of Cavallini on condensation of R11 in the same coordinates (17) are presented in Fig. 6. The Prandtl numbers in the experiments on condensation of motionless R21 [29] and moving R11 vapors [16] are very close. Our experimental data on condensation of R21 served as a scale in this processing. The comparison was made at Re_{liq} = idem. It is obvious from the data presented that the ratio of Nusselt numbers increases with the velocity of vapor. For a specified velocity of vapor this ratio virtually does not de-

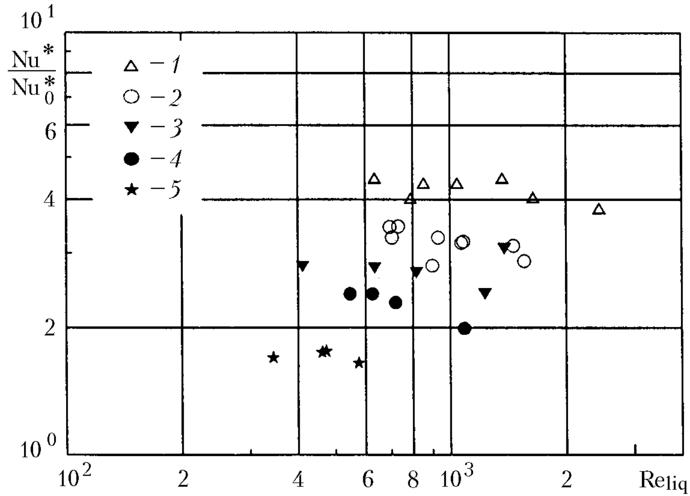

Fig. 6. Relative variation of heat transfer as a function of Re_{liq} in condensation of moving R11 vapor [16]: 1) U_v = 36; 2) 27; 3) 21; 4) 17; 5) 12 m/sec.

pend on the Re number for a film. For all the experiments presented, the ratio Nu^*/Nu_0^* is higher than unity, which indicates the effect of vapor velocity on the process of heat transfer in vapor condensation in the tube.

Thus, the data presented in Figs. 5b and 6 and Table 2 show the existence of two ranges of parameters at which:

a) heat transfer virtually does not depend on the velocity of vapor at the inlet to the tube;

b) considerable enhancement of heat transfer with increasing vapor velocity is observed.

Of the works listed in Table 1, results of two studies — [44] and [48] — are worthy of note. The first work gives the results of two series of experiments conducted at different times. The systematic difference of them at the same conditions reaches 20–25%. Then two series of experiments were processed. However, the systematic difference of the data is pronouncedly seen in their processing, given in Fig. 7.

In our opinion, the experiments on condensation of propane described in [48] were conducted with a systematic error. At velocities of vapor when it can be assumed motionless, results of these measurements lie about 30% above the data obtained in condensation of motionless vapor. The heat fluxes at which measurements were made were very small. Most likely it is very difficult to measure them with proper accuracy; therefore, the data of [48] are not presented in Fig. 7.

Influence of Entrainment on Heat Transfer. Hewitt and Hall-Taylor [35] state that in an annular flow mode the surface waves act as pumps in the film, which pump liquid from the film to the vapor core. Moreover, droplets move toward the film surface. Dynamic equilibrium, at which the velocity of supply of droplets to the gas core from the film by disturbance waves is counterbalanced by deposition of droplets, is reached between these processes. At present, there exist a large number of empirical and semi-empirical dependences for determining the vapor velocity at which entrainment begins. A review and analysis of these works is given in [35]. Results on entrainment of liquid from the film surface blown by an isoteric gas flow, which, in our opinion, are of importance, are presented in more recent works [51–55]. The experimental value of the Weber number

$$We_{cr}^{1/2} = \frac{U_v \rho_v^{1/2} \delta^{1/2}}{\sigma^{1/2}} > 2 \tag{26}$$

is given in [53] as the parameter characterizing the onset of film entrainment.

The drawback of expression (26) is the complexity of calculation of the mean film thickness, which ambiguously depends on the Re number for a film (flow mode). In laminar film flow, the film thickness is determined from the Nusselt theory:

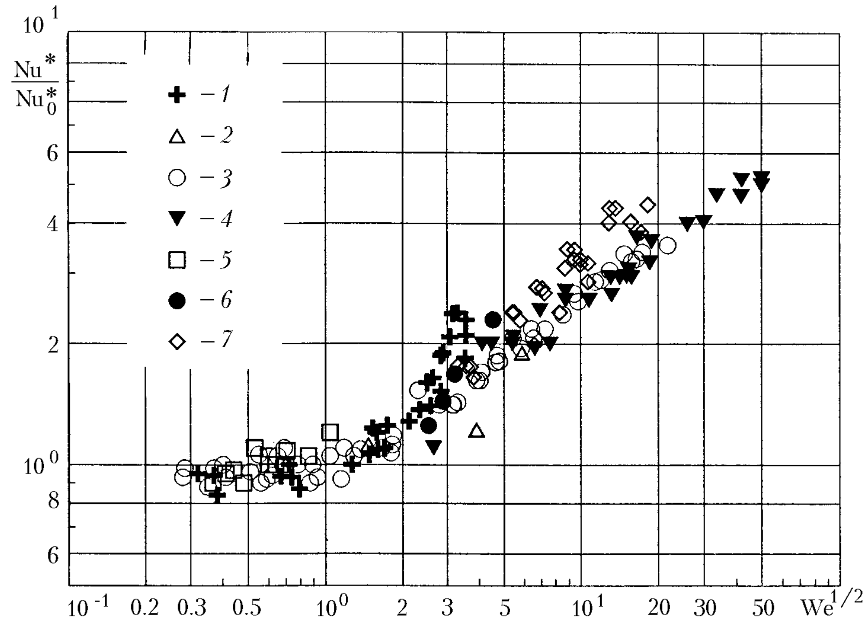

Fig. 7. Influence of entrainment on heat transfer in condensation of moving vapor: 1–5) water vapor [1) by the data of [44]; 2) [47]; 3) [24]; 4) [46]; 5) [49, 50]]; 6) benzene, by the data of [45]; 7) R11, by the data of [16].

$$\delta = \left(\frac{3\nu^2}{g}\right)^{1/3} Re_{liq}^{1/3}, \tag{27}$$

where $Re_{liq} \le Re^*$.

In [56], in the wave mode of film flow, it is suggested that the mean film thickness be found by the empirical Fulford dependence

$$\delta = 0.883 \left(\frac{3\nu^2}{g}\right)^{1/3} Re_{liq}^{0.337}, \tag{28}$$

here $Re^* \le Re_{liq} \le Re_{cr}$.

According to [32], in the turbulent film flow, when $Re_{liq} > Re_{cr}$ the mean film thickness is determined as

$$\delta = Re_{cr}^{-0.2} \left(\frac{3\nu^2}{g}\right)^{1/3} Re_{liq}^{8/15}. \tag{29}$$

Here the Reynolds numbers for a film, Re^* and Re_{cr}, correspond to laminar-to-wave and wave-to-turbulent mode transitions and are calculated by the expressions (21) and (22) given earlier.

The dependences presented for determining the film thickness allow one to write

$$\delta \approx \left(\frac{\nu^2}{g}\right)^{1/3} Re_{liq}^{n}. \tag{30}$$

Having substituted (30) into (26), we obtain

$$We^{1/2} = Fr_v^{1/2} \left(\frac{\rho_v}{\rho_{liq}}\right)^{1/2} \frac{Ga^{1/6} Re_{liq}^{n_1/2}}{Ka^{1/6}}. \tag{31}$$

Here the Froude number is determined by the velocity of vapor at the tube inlet, and the Reynolds number for a film — by reflex density at the tube outlet.

Allowing for the fact that in relations (27)–(29) the exponent at $\mathrm{Re}_{\mathrm{liq}}$ is variable and lies within $1/3 \le n \le 1/2$, the exponent at $\mathrm{Re}_{\mathrm{liq}}$ in expression (31) will vary within $1/6 \le n_1 \le 1/4$ with changes in the mode of film flow.

The expression for determining the critical velocity of vapor (31), at which film entrainment begins, vividly shows that it depends on the ratio of the densities of vapor and liquid, the regime of film flow, and the physical properties of the condensate film.

In Fig. 7, the results of processing of the experiments on condensation of moving vapor of water [24, 44, 47, 49, 50], benzene [45], and refrigerant R11 [16] in tubes of different lengths and diameters are presented in the coordinates

$$\frac{\mathrm{Nu}^*}{\mathrm{Nu}_0^*} = f\left(\frac{U_{\mathrm{v}}\, \rho_{\mathrm{v}}^{1/2} \delta^{1/2}}{\sigma^{1/2}}\right) \equiv \mathrm{Fr}_{\mathrm{v}}^{1/2} \left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{liq}}}\right)^{1/2} \frac{\mathrm{Ga}^{1/6} \mathrm{Re}_{\mathrm{liq}}^{n_1}}{\mathrm{Ka}^{1/6}} \,. \tag{32}$$

Comparison is made at $\mathrm{Re}_{\mathrm{liq}}$ = idem and Pr = idem. In processing the experiments of [49, 50] for turbulent film flow modes the local Nusselt number Nu_0^* was calculated by (23).

Dependence (32) is, naturally, empirical, since it is based on the empirical expression for calculation of the critical value of the Weber number (26). Moreover, the film thickness for wave and turbulent film-flow modes is determined by empirical formulas (28) and (29). The determining criteria in expression (32) have two linear scales — the viscosity-gravity constant $l_{\mathrm{v}} = (\nu^2/g)^{1/3}$ and tube length L — and do not depend on the tube diameter. Experimental data shown in Fig. 7 are obtained in condensation in tubes the diameter of which changed multiply.

A principal drawback of this processing is the absence of objective information on the effect of crossflow of a substance on the Re numbers for a film which characterize laminar-to-wave and wave-to-turbulent mode transitions. However, the results of processing of the experiments in condensation of vapor of different liquids show (Fig. 7) that in the coordinates (32) the experimental data are generalized satisfactorily.

Processing of the experiments allows one to state that entrainment of wave crests into the vapor flow is one of the main mechanisms of enhancement of heat transfer in condensation of vapor moving inside tubes. When $\mathrm{We} > \mathrm{We}_{\mathrm{cr}}$, heat transfer is enhanced manifold compared to the case of motionless vapor.

When $\mathrm{We} < \mathrm{We}_{\mathrm{cr}}$, the effect of entrainment is not observed and intensities of heat transfer in condensation of moving and motionless vapor coincide. In our opinion, this is due to the fact that in wave film flow the "residual" thickness does not change under the effect of vapor flow.

NOTATION

a, thermal diffusivity, m^2/sec; α_{x_2}, coefficient of heat transfer in incomplete condensation of vapor, W/(m^2·deg); α_{conv}, coefficient of heat transfer in a one-phase liquid flow, W/(m^2·deg); α_0 and α, coefficients of heat transfer in condensation of motionless and moving vapor, respectively, W/(m^2·deg); C_p, heat capacity of liquid, J/(kg·$^{\mathrm{o}}$C); d and L, inner diameter of the tube and its length, m; F, cross section of the tube, m^2; g, free-fall acceleration, m^2/sec; l_{v}, viscosity-gravity constant, m; m, mass velocity of liquid, kg/sec; P, pressure, N/m^2; q, specific heat flux, W/m^2; Q_{in} and Q_{out}, power at the inlet to and outlet from the experimental section, W; r, latent heat of vaporization, J/kg; t_{w} and t'', wall temperature and temperature of saturated vapor, deg; U_{v} and U_{liq}, velocities of vapor and liquid, m/sec; $\overline{U}_{\mathrm{liq}}$, mean velocity of liquid, m/sec; $\Delta t = t'' - t_{\mathrm{w}}$, "vapor–wall" temperature head, deg; δ and δ_{cr}, thickness and critical thickness, m; $\vartheta^* = (\tau_{\mathrm{w}}\rho)^{1/2}$, rate of tangential stress, (m/sec)$^{1/2}$; $\vartheta = q/(r\rho_{\mathrm{v}})$, transverse component of vapor velocity, m/sec; λ, thermal conductivity, W/(m·deg); μ, dynamic viscosity, Pa·sec; ν, kinematic viscosity,

m^2/sec; ρ, density, kg/m^3; σ, surface tension, N/m; $\tau = \frac{qU_v}{r}$ and τ_w, friction on the phase interface and the wall, kg/(m·sec^2); $b = \frac{2q}{r\rho_v U_v c_{fr}}$, relative coefficient of friction; c_{fr}, local coefficient of friction for a one-phase flow; c_f, coefficient of friction in a turbulent boundary layer; $Fr_v = \frac{U_v^2}{gL}$, Froude number in the Fujii theory and correlation (32); $Fr_v = \frac{U_v^2}{gd}$, Froude number in formula (7); $Fr_0 = \frac{\overline{U}_{liq}^2}{gd}$, Froude number in formulas (4) and (5) determined by the mean velocity of liquid at the tube inlet; Ga, Galileo number; $Ga_x = \frac{x^3 g}{\nu_{liq}^2}$ and $Ga_L = \frac{L^3 g}{\nu_{liq}^2}$, Galileo numbers — local over x and determined by L, respectively; $K = \frac{r}{C_p \Delta t}$, Kutateladze criterion; $Ka = \frac{\sigma^3}{\nu_{liq}^4 \rho_{liq}^3 g}$, Kapitsa number; n and n_1, variable exponents in (31) and (32); Nu, Nusselt number; $Nu_0^* = \frac{\alpha_0}{\lambda}\left(\frac{\nu_{liq}^2}{g}\right)^{1/3}$ and $Nu^* = \frac{\alpha}{\lambda}\left(\frac{\nu_{liq}^2}{\lambda}\right)^{1/3}$, Nusselt numbers in condensation of motionless and moving vapor constructed by the viscosity-gravity constant; $Nu_d = \frac{\alpha d}{\lambda}$, Nusselt number in condensation of moving vapor calculated by the tube diameter; $Nu_x = \frac{\alpha x}{\lambda}$ and $\overline{Nu} = \frac{\alpha L}{\lambda}$, local and averaged Nusselt numbers constructed by the coordinate x and the tube length L; $\overline{Nu}_{x_2} = \frac{\alpha_{x_2} L}{\lambda}$, Nusselt number in incomplete condensation of vapor in (3); $\overline{Nu}_{x_2=0} = \frac{\alpha_{conv} L}{\lambda}$, Nusselt number in convective heat transfer of a one-phase liquid in (3); $Pr_{liq} = \frac{\nu_{liq}}{a_{liq}}$ and $Pr_v = \frac{\nu_v}{a_v}$, Prandtl numbers for liquid and vapor; $R = \left(\frac{\rho_{liq}\mu_{liq}}{\rho_v \mu_v}\right)^{1/2}$, dimensionless complex; Re, Reynolds number; $Re_v' = \frac{U_v L}{\nu_v}\frac{\nu_v}{\nu_{liq}} = \frac{U_v L}{\nu_{liq}}$, Reynolds number in the Fujii theory; $Re_v = \frac{U_v L}{\nu_v}$, Reynolds number for a vapor flow; $Re_{liq} = \frac{\langle U_{liq}\rangle \langle \delta \rangle}{\nu_{liq}} = \frac{\bar{q}L}{\mu r}$, Reynolds number for a film; $Re_{liq} = \frac{\langle U_{liq}\rangle d}{\nu_{liq}} = 4Re_{liq}$, equivalent Reynolds number for a film ($d_{eq} = 4\delta$); $Re_x = \frac{U_v x}{\nu_v}$, Reynolds number for the vapor flow in the coordinate x along the tube length; $S_q = \frac{q}{r\rho_v U_v}$, dimensionless transverse component of velocity; $We = \frac{U_v^2 \rho_v \delta}{\sigma}$, Weber criterion; x_1 and x_2, vapor content at the inlet to and outlet from the tube ($x_2 = Q_{out}/Q_{in}$); x, coordinate; $Z = x/L$, current dimensionless coordinate; $\bar{\mu}_t = \mu_t/\mu$, dimensionless eddy viscosity; κ, Prandtl–Kármán constant; $\tau_w^* = (\tau_w/\rho_{liq})/(g\nu_{liq})^{2/3}$, dimensionless friction; $\eta = \vartheta^* y/\nu$ and $\eta_\delta = \vartheta^*\delta/\nu$, dimensionless distance from the wall and the film thickness, respectively; ε_{max}, cross-section maximum value of intensity of velocity fluctuations. Indices: wave, wave; in, inlet; out, outlet; liq, liquid; cr, critical; lam, laminar; v, vapor; w, wall; t, turbulent; fr, friction; eq, equivalent; conv, convective; max, maximum; overbar denotes the mean value.

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