VAPOR CONDENSATION INSIDE HORIZONTAL PIPES

V. G. Rifert

UDC 536.24:536.420.4

The investigation of heat exchange during film condensation of vapor inside horizontal pipes is as timely a problem as ever. The need to conduct further investigations of this process is due to two main factors: First, the contradictory nature and the absence of fundamental limits on the use of the calculating functions for heat transfer from vapor to wall, and second, expansion of the area of application of the process of condensation inside horizontal pipes in evaporators and condensers of distilling, drying, power, and refrigerating equipment [1-4].

It must be noted that neither an analysis nor recommendations on the calculation of heat exchange during vapor condensation inside horizontal pipes are encountered in the published monographs or textbooks devoted to processes of heat exchange.

Hydrodynamics of Two-Phase Flow with Condensation inside Horizontal Pipes. In contrast to vapor condensation inside vertical pipes and on the outer surface of a horizontal pipe, where the flows of vapor and condensate film are parallel and most often comoving, inside a horizontal pipe the gravity vector is perpendicular to the direction of vapor motion, which considerably complicates the hydrodynamics of the two-phase flow. Important aspects of the hydrodynamics here are the modes of flow of the phases with respect to structure and character, determination of the zone of the pipe cross section flooded with condensate, and the frictional resistance at the phase interface.

Extremely little attention has been paid to these questions in the literature in application to vapor condensation inside horizontal pipes.

Baker's chart of modes, which is presented in a number of monographs, [5] in particular, is classical for the determination of modes of flow of a two-phase stream inside a horizontal pipe. Constructed on the basis of experimental data on the flow of an adiabatic stream (a mixture of water and air at atmospheric pressure), this chart and modifications of it only approximately characterize the stream structure when a two-phase mixture is supplied to the pipe entrance. As noted in [5], the hydrodynamics of the stream essentially depends on the conditions of entrance of the phases, and therefore we are interested in analyzing the characteristics of modes for our concrete case with vapor condensation.

For condensation at the entrance with $x \geqslant 1$ (saturated or superheated vapor) the entrance cross section is occupied only by vapor. The subsequent stream structure will depend on the ratio of the force of gravity and vapor friction, which act on the condensate film which forms, and on the ratios of volumes of liquid and vapor. Almost the only estimate of the modes of flow of a two-phase stream with condensation inside a horizontal pipe is given in [6]. The authors distinguish the four most probable and experimentally observed modes of flow (Fig. 1) and characterize them by two forces or the pressure gradient: the longitudinal force due to vapor friction,

$$F_{\text{fr}} = \frac{dP_{\text{fr}}}{dL} = \frac{2C_f G_{\text{v}}^2}{D\rho_{\text{v}}}, \tag{1}$$

and the radial force due to gravity,

$$F_{\text{pr}} = \frac{dP_{\text{pr}}}{dD} = g(\rho - \rho_{\text{v}}). \tag{2}$$

Then the authors of [6] construct the ratio of forces (1) and (2), which is one of the complexes for characterizing the modes:

Fiftieth Anniversary of the Great October Socialist Revolution Polytechnic Institute, Kiev. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 44, No. 6, pp. 1017-1029, June, 1983. Original article submitted February 15, 1982.

 0022-0841/83/4406-0700$07.50

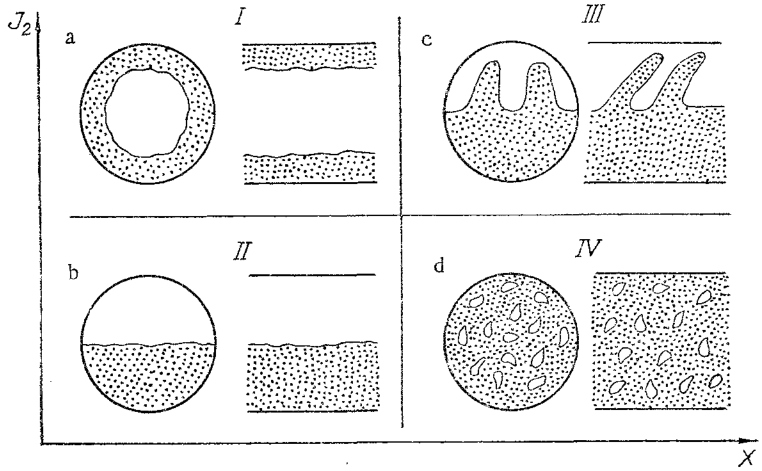


Fig. 1. Diagram of modes of flow of phases in a horizontal pipe: a) region I, annular and annular misted mode; b) region II, wave and stratified mode; c) region III, intermittent mode; d) region IV, bubble mode.

$$J_2 = \left[\left(\frac{F_{fr}}{F_{pr}}\right)\left(\frac{1}{2C_f}\right)\right]^{0.5} = \frac{G_{mix}}{[Dg\rho_v(\rho-\rho_v)]^{0.5}}, \quad (3)$$

where $G_{mix} = G_v + G_c$ is the total mass flow rate of vapor and condensate, $kg/m^2 \cdot sec$. The coefficient of friction C_f, which is difficult to determine, especially for separate flow of the phases, was thereby eliminated from (1) and (2). The second complex, laid out along the other axis of the chart of modes, characterizes the volumetric vapor content [6]:

$$X = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho}\right)^{0.5}\left(\frac{\mu}{\mu_v}\right)^{0.1} \quad (4)$$

A simplified method of estimating the stream structure is proposed in [6] in application to the calculation of heat transfer during vapor condensation inside a horizontal pipe: For $J_2 > 1.5$ and $X < 1.0$ one assumes an annular mode; for $J_2 < 0.5$ and $X < 1.0$ a wave or stratified mode; for $J_2 < 1.5$ and $X > 1.5$ a plug (intermittent) mode; for $J_2 > 1.5$ and $X > 1.5$ a bubble mode. There is a transitional zone between regions I and II and between II and III.

This calculating scheme of modes is in satisfactory agreement with the experimentally observed (visually) stream structure for the condensation of vapors of various liquids (water vapor, freon) inside pipes with D = 4.8-50.8 mm and L = 0.61-6.7 m [6].

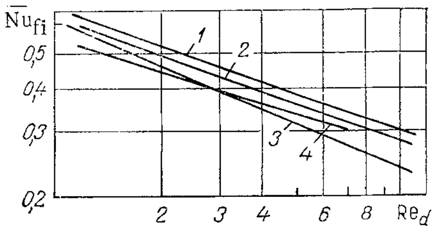
It must be mentioned that visual determination of modes of flow is very subjective. Moreover, the approach to the determination of modes of flow adopted in [6] does not allow for the thickness and viscosity of the condensate. The balance of shear stresses acting on the condensate film and due to friction at the phase interface and gravity was analyzed in [2]. Their relation has the form

$$\Pi = \frac{C_f\rho_v W_v^2}{2\rho g\delta}, \quad (5)$$

where the film thickness δ in annular flow under the action of frictional forces can be estimated from well-known functions of [7].

For $\Pi \geqslant 10$ one observes predominantly annular flow of the condensate film, while for $\Pi \leqslant 0.1$ it flows down along the perimeter of the pipe under the action of gravity. Both forces operate in a wide range of variation of the mode parameters, i.e., over a large part of the pipe length. To estimate the ratio (5) one must know C_f. Experimental data on C_f for vapor condensation are confined to one report [8], for pressures $P > 1.23$ MN/m^2 and an annular mode of phase flow.

The next important parameter is the fraction of a pipe cross section occupied by condensate in the stratified mode of phase flow. The depth h of the liquid layer inside a horizon-



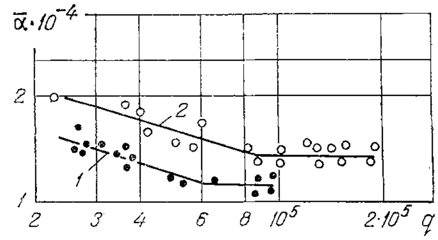


Fig. 2 Fig. 3

Fig. 2. Heat exchange during vapor condensation under the conditions of the stratified mode of phase flow: 1) calculation from (10), test data; 2) [22]; 3) [23]; 4) [2, 24]; $\overline{\mathrm{Nu}}_{fi} = \frac{\bar{\alpha}}{\lambda}\left(\frac{\nu^2}{g}\right)^{1/3}$, $\mathrm{Re}_D = qd/r\mu$.

Fig. 3. Heat transfer with complete condensation of water vapor inside a pipe: 1) P = 0.015 MN/m²; 2) 0.11.

tal pipe was determined in [5, 9-16]. The case of liquid flow under the action of a constant pressure gradient along the pipe axis for an adiabatic stream was considered in [5]. No functions for engineering calculations of h are given in [5], and only the absence of convergence between the solutions obtained and experiment is indicated.

The flow of a stream in an inclined pipe under the action of the gravity component was studied in [14], so that from the overall dependence for the angle of stream overflow it was found that $\varphi_0 = \infty$ for a horizontal pipe. In fact, as the author of [14] himself notes, free flow of liquid from the end of the pipe is also observed in a horizontal pipe owing to the slope of the free surface of the stream at the end of the pipe. Kutateladze [13] experimentally investigated water flow at the bottom of a semicircular horizontal trough. Chaddo [9] and Chato [10] derived an equation for calculating the level of condensate using the theory developed in the hydraulics of open channels for zero-head flow, i.e., without allowance for the intensity of condensation or the velocity of the condensing vapor.

V. V. Konsetov [14] generalized the data of [9, 13] by the dependence

$$1 - \cos\varphi_0 = 4.2\,\mathrm{Fr}^{0.33}\left(\frac{\sigma}{\rho g D^3}\right)^{0.25}, \tag{6}$$

where $\mathrm{Fr} = qL/r\rho(gD^3)^{0.5}$; φ_0 is the overflow angle at the end of the pipe.

The case of complete vapor condensation is analyzed in [11] and it is assumed that at the end of the pipe, where the vapor content is x = 0, the liquid entirely fills the pipe cross section. Here the conditions at the pipe outlet are not considered.

In [15, 16] it is noted that in the condensation of vapor inside a horizontal pipe it is possible both for the entire pipe cross section to fill with condensate at the end, when vapor condensation is complete, and for only part of the pipe to fill with a stream of condensate.

In [15, 16] a detailed analysis is made of differential equations, obtained in [9-11], describing the variation of the angle φ_0 of the condensate stream in the stratified mode of flow. Functions for engineering calculations of the parameters of the condensate stream are absent from [15, 16]. Therefore, this report, just like [11], is of interest only on the theoretical level and for the analysis of contradictory data on heat transfer. In practice, condensation inside pipes is organized so that there is some blow-through of vapor, and the mode of free runoff of condensate is observed in this case. Then the overflow angle in each section of a pipe with a length L can be determined from the function (6).

For a theoretical calculation of heat transfer during vapor condensation one must know the coefficient of frictional drag at the phase interface. The assumption that one can take the same C_f as for adiabatic flow, as is done in many reports [12, 13, 17-20], does not agree with well-known experimental data (on the flow of two-phase streams) [21]. The results of [8] indicate a considerable discrepancy (up to 100% or more) between the drag for vapor condensation and calculations based on a homogeneous theory with adiabatic flow.

Thus, investigations of hydraulic drag during vapor condensation also comprise one of the urgent problems, the solution of which is needed for the development of a method of calculating heat exchange.

Heat Exchange during Condensation. An analysis of the hydrodynamics of two-phase flow with vapor condensation inside a horizontal pipe shows that modes of flow of the phases which differ in structure can exist in different sections of the pipe: annular, stratified, and intermittent flow. The mechanism of heat exchange and its laws in these zones will be different. As shown by an analysis of reports, the results of which are given below, there is considerable qualitative disagreement in the laws of variation and a large scatter in the values of the average coefficients of heat transfer obtained by different authors.

First of all, let us consider the qualitative disagreement in the laws of heat exchange. In [2, 9, 10, 22-25] the average coefficient of heat transfer $\bar{\alpha}$ decreases with an increase in the heat load and, as is shown in Fig. 2, is close quantitatively to the coefficient of heat transfer during condensation on the outer surface of a horizontal pipe. In many reports [3, 13, 17-20, 26-32] the average heat transfer increases with an increase in the heat load q, with the exponent to q of different authors ranging from 0.4 [13] to 1.7 [32]. In some reports [2, 14, 17] it is noted that $\bar{\alpha}$ is practically independent of q in a certain range of variation of q. The dependence of $\bar{\alpha}$ on q has the same character for $q > 5\cdot10^4$ W/m² in Kutateladze's experiments [13] for a low-pressure vapor, although the author does generalize the experimental data for all q with one function where $\bar{\alpha} \sim q^{0.4}$. Without yet dwelling on the models and the assumptions (often disputable) adopted by various authors in their construction (this will be done below), with the help of which empirical and semiempirical equations were obtained for calculating the average heat transfer during vapor condensation, let us consider the possible reasons for this disagreement in the qualitative laws of variation of $\bar{\alpha}$. The increase in $\bar{\alpha}$ with q in the entire range of variation of q obtained in [3, 26, 27] is explained by the authors by the fact that the velocity of the vapor and condensate increases with an increase in q. However, this would be correct if the annular mode of phase flow were predominantly observed in the entire length of the pipe.

An estimate of the forces from (3)-(5) and the visual observations of these authors indicate the presence of the stratified mode in these experiments in a wide range of variation of q along most of the pipe.

In [14] the author explains the different degrees of the dependence of $\bar{\alpha}$ on q obtained in [13, 25, 26] by the influence of heat transfer from the vapor to the wall on the average experimental coefficient and by the difference in the ratios of the coefficients of heat transfer in the condensate stream ($\bar{\alpha}_p$, the zone with φ_0), in the section of the pipe where the condensate flows under the action of gravity ($\bar{\alpha}_n$, the zone with $\pi-\varphi_0$), and on the cooling side ($\bar{\alpha}_2$), as well as the law of variation of $\bar{\alpha}_2$. Thus, the author assumes that the stratified mode of phase flow always exists. The analysis in [14] was made without allowance for the temperature distribution in the wall, a calculation of the fraction of the pipe cross section occupied by the condensate stream, or a determination of the mode of phase flow. In addition, the dependence for the average temperature drop, which serves as the basis for the building of subsequent arguments in [14], was found incorrectly. All this allows us to assert that the causes given in [14] for the different character of the influence of q on $\bar{\alpha}$ obtained in [13, 25, 26] are unconvincing.

The main reasons for the different character of the influence of q on $\bar{\alpha}$ obtained in the experiments of different authors are the following. The first reason is that the integral-mean coefficients of heat transfer for the entire pipe were measured in all the reports. However, the modes of phase flow as a function of pipe diameter and length, the physical properties of the vapor and condensate, and the degree (complete or incomplete) of vapor condensation vary considerably over the length of the pipe, and each individual zone of the pipe (containing the annular, intermittent, or stratified modes of phase flow) has its own law of heat exchange, since it is determined by the interaction of different forces.

Furthermore, the state of the two-phase stream at the end of the pipe with complete vapor condensation was not monitored at all in many reports [3, 17, 19, 32]. And since a greater or lesser part of the pipe may be entirely filled with condensate in this case, according to [15, 16], the condensation process did not take place in it. In this case the working length of the pipe and the experimental coefficient of heat transfer grew with an increase in q.

And finally, even with an annular mode of phase flow in the entire length of the pipe, the degree of influence of q on $\bar{\alpha}$ can differ depending on the character of the mode of flow of the condensate film and the degree of vapor condensation. For example, with laminar flow of the condensate film under the action of interphase friction the local heat transfer can be calculated from a function known from [7],

$$\alpha = \frac{\lambda}{2}\left[\frac{rC_f W_V^2 \rho_V(\rho - \rho_V)}{\mu q Z}\right]^{0.5}, \tag{7}$$

where the average vapor velocity W_V in any pipe cross section Z is determined in generas as

$$W_V = W_{bl} + \frac{4q(L - Z)}{rD\rho_V}. \tag{8}$$

With partial vapor condensation $\bar{\alpha}$ will grow as $\sim q^{0.4}$ with an increase in q for turbulent vapor flow, if one assumes that $C_f \sim Re_V^{-0.2}$. For laminar vapor flow α may hardly depend on q. With a blow-through velocity $W_{bl} > 0$, depending on its magnitude, a situation may develop such that an increase in the thickness of the condensate film results in a decrease in α with an increase in q.

Now let us examine the models and methods of calculation of heat transfer during vapor condensation inside a horizontal pipe, with allowance for their consistency with the mode of phase flow, proposed by different authors.

First of all, we consider situations when stratified or annular modes of phase flow occur over most of the pipe. In the stratified mode of phase flow, when the level of the condensate stream is $\varphi_0 \ll \pi$ over most of the pipe, the average heat transfer should agree qualitatively and quantitatively with that calculated from the Nusselt equation for the case of vapor condensation on the outer surface of a horizontal pipe. With allowance for φ_0 the function for the local (average for a specific pipe cross section) $\bar{\alpha}$ will have the form

$$\bar{\alpha} = \bar{\alpha}_{Nu}\left(1 - \frac{\varphi_0}{\pi}\right), \tag{9}$$

where

$$\bar{\alpha}_{Nu} = \left(\frac{\lambda^3 \rho(\rho - \rho_V) g r}{4\mu D \Delta T}\right)^{0.25}. \tag{10}$$

To obtain $\bar{\alpha}$ over the entire length of the pipe with the stratified mode of phase flow one must integrate (6) over Z, where Z is the coordinate along the pipe. Equation (6) is inconvenient for such integration. Therefore, a simpler relation obtained by Kutateladze [13] can be used for this purpose.

$$\frac{U}{\pi D} = 19.27\left(\frac{V\nu^3}{g^2 D^7}\right)^{0.12}, \tag{11}$$

where U is the perimeter of the lower part of the pipe wetted by the condensate stream; V is the volumetric flow rate of liquid, $V = q\pi DZ/r\rho$.

The integral-mean value is

$$\frac{\bar{U}}{\pi D} = 20.1\left(\frac{q\nu^3 L}{r\rho g^2 D^6}\right)^{0.12}. \tag{12}$$

Then with a constant ΔT over the length of the pipe, the average coefficient of heat transfer is defined as

$$\bar{\alpha} = \bar{\alpha}_{Nu}\left[1 - 20.1\left(\frac{q\nu^3 L}{r\rho g^2 D^6}\right)^{0.12}\right]. \tag{13}$$

Using (10), we represent (13) in the dimensionless form

$$\overline{\mathrm{Nu}}_{\mathrm{fi}} = 0.627\,\mathrm{Re}_D^{-0.33}\left[1 - 20{,}1\left(\frac{g\nu^3 L}{r\rho g^2 D^6}\right)^{0.12}\right]. \tag{14}$$

The experimental data of different authors [2, 22-24] who investigated the condensation of freons, methyl alcohol, and water under conditions when the stratified mode of phase flow occurred over most of the length of the pipe are compared in Fig. 2 in the coordinates $\overline{\mathrm{Nu}}_{fi} = f(\mathrm{Re}_D)$. Our data [2, 24] on the condensation of water vapor at a pressure P = 0.15 and 1.1 bar were obtained inside copper and brass pipes with D = 20 mm and L = 1.5 m in the range of variation of heat fluxes $q = 2\cdot10^4 - 10^5$ W/m² (Fig. 3).

In our experiments the outer surface of the pipe was cooled by high-velocity water circulation, so that the water temperature varied along the length of the pipe within limits of no more than 4-5°C.

The vapor condensation was organized with a slight (2-3%) evaporation to avoid filling the pipe cross section with condensate. In each test this mode was monitored visually through a viewing window mounted in the exit chamber behind the pipe. When complete vapor condensation (without blowthrough) was organized we noted a sharp decrease in the vapor pressure and temperature at the exit from the pipe and in the average coefficient of heat transfer, which increased with an increase in q. Heat exchange of this character is explained by the appearance of uncondensed gases at the end of the pipe and the exclusion of part of the pipe from operation. An increase in q naturally decreases the zone of the pipe occupied by uncondensed gases, which results in an increase in $\bar{\alpha}$ for the entire pipe. The influence of q on $\bar{\alpha}$ is the same in the case of filling of part of the pipe with condensate during complete vapor condensation. For $q > 7\cdot10^4$ W/m² the initial section of pipe with an annular mode of phase flow occupied a large part of the pipe, as a result of which the law of variation of the average $\bar{\alpha}$ with q was changed. Thus, in the region of the layered mode the relatively scanty data of various authors are in satisfactory qualitative and quantitative agreement with each other and for small q they are close to the line calculated from Eq. (10).

The separation of the data of different authors into layers can be explained by the different degrees of influence of the condensate stream. With allowance for φ_0 from (14) all the data agree very well with each other, as the calculations showed. A law of heat exchange close to that for vapor condensation on the outer surface of a horizontal pipe, i.e., characteristic of the stratified mode of phase flow, also occurred in the experiments whose results are presented in [9, 10, 25]. When the level of the condensate stream is high and the stratified mode of phase flow is conserved, one must allow for vapor condensation in the stream, which can have decisive importance for $\bar{\alpha}$. We note, however, that larger φ_0 are possible in long and small-diameter pipes and for relatively large q and small r. On the other hand, all these factors (an increase in L and q and a decrease in D and r) promote the maintenance of the annular mode of phase flow over most of the pipe, where $\bar{\alpha}$ most often grows with an increase in q. To clarify what has the greater influence on $\bar{\alpha}$, the section with an annular mode at the start of the pipe or the section with a deep stream at the end of the pipe requires a local analysis of the sections of the pipe.

For annular two-phase flow one must distinguish between laminar and turbulent flow of the condensate film. The vapor flow is almost always turbulent in the annular mode, since Re_v is always larger than 10^4 here. Concerning the critical value of Re_{fi} for a film when it flows in a moving vapor, there is an absence in the literature of sufficiently objective research yielding a clear answer about the critical value Re_{fi_c}. It is only known that with an increase in the gas (vapor) velocity in comoving phase flow Re_{fi_c} is smaller than for a stationary vapor. In [7] it is indicated that $\mathrm{Re}_{fi_c} = 60$. A wide range of Re_{fi_c} is given in various reports, from 60 to 500, where the film Reynolds number for condensation is written as $\mathrm{Re}_{fi} = qL/r\mu$.

In all reports on heat exchange during vapor condensation inside a horizontal pipe under the conditions of the annular mode of phase flow, the turbulent—turbulent flow of both phases is analyzed.

Such a problem was first solved by Kutateladze [13] within the framework of a semiempirical theory of turbulent heat exchange. In later reports [17, 18, 30, 31] the problem was solved on the basis of the Reynolds theory of hydrodynamic analogy.

In all these reports the coefficient of friction C_f between vapor and film, appearing in the dependence for the shear stress, is taken as equal to C_f for one-phase turbulent fluid flow. It is well known [21], however, that C_f for a two-phase stream can differ considerably (severalfold) from C_f for a one-phase stream.

Kutateladze's equation [13] for calculating the average heat transfer during complete vapor condensation in a pipe has the form

$$\overline{\mathrm{Nu}}_D = 0.04\,\mathrm{Re}_{\mathrm{fi}}^{0.8}\mathrm{Pr}^{0.4}\left(\frac{\rho}{\rho_\mathrm{v}}\right)^{0.5}\left(\frac{\mu_\mathrm{v}}{\mu}\right)^{0.2}, \qquad (15)$$

where the constant is determined in [13] from experimental data.

A comparison of (15) with experimental data presented in [13] shows satisfactory qualitative agreement with theory only in the region of $\mathrm{Re}_{\mathrm{fi}} > 10^3$. The scatter of the experimental data relative to the line (14) is up to ±100%.

The decrease in the average heat transfer with an increase in pressure with the other parameters (q, L, D) constant, obtained in the experiments of [13], attracts attention. A dependence very close to (15) for heat transfer during turbulent flow of a condensate film and with the dominant influence of the vapor velocity was obtained in [17]:

$$\overline{\mathrm{Nu}}_D = 0.0417\,\mathrm{Re}_{\mathrm{fi}}^{0.8}\mathrm{Pr}^{0.4}\left(\frac{\rho}{\rho_\mathrm{v}}\right)^{0.5}\left(\frac{\mu_\mathrm{v}}{\mu}\right)^{0.1}. \qquad (16)$$

The functions (15) and (16) are valid for complete condensation of the moving vapor.

A different approach to the solution of the problem from that in [13] was used in [18, 31]. The authors presume the presence of intensive removal of liquid from the film surface into the vapor stream, and in this connection they assume that the liquid flow rate G_c in the film is much less than the flow rate of the mixture, $G_{mix} = G_v + G_c$, where G_v is the vapor flow rate. Therefore, a homogeneous model of the flow of a two-phase stream is used.

The heat-transfer function in [18, 31] is determined to within a constant c, which is found from experiment,

$$\overline{\mathrm{Nu}}_D = c\,\mathrm{Re}_{l\mathrm{v}}^{0.8}\mathrm{Pr}^{0.43}\frac{1}{2}\left\{\left[1 + x_1\left(\frac{\rho}{\rho_\mathrm{v}} - 1\right)\right]^{0.5} + \left[1 + x_2\left(\frac{\rho}{\rho_\mathrm{v}} - 1\right)\right]^{0.5}\right\}, \qquad (17)$$

where x_1 and x_2 are the vapor content at the pipe entrance and exit, $\mathrm{Re}_{lv} = 4G_{mix}/\pi D\mu$, while G_{mix} is the flow rate of the mixture, assumed by the authors to be close to the vapor flow rate. Thus, $\mathrm{Re}_{lv} = 4qL/r\mu$ for $x_2 = 0$. The proportionality constant for a copper pipe is c = 0.032, and for a stainless steel pipe c = 0.024. There is no explanation of the influence of the material on heat transfer in these reports.

In [29] the authors assume the following semiempirical function for calculating heat transfer in the case of vapor condensation with turbulent flow of both phases:

$$\overline{\mathrm{Nu}}_D = 1.136\,\mathrm{Re}_{\mathrm{fi}}^{0.8}\mathrm{Pr}^{0.33}(1 + \mathrm{Pr}\,K)\left(\frac{D}{L}\right)^{0.72}. \qquad (18)$$

Thus, according to this report, heat transfer is influenced by the simplex D/L and by the product of the Prandtl and Kutateladze numbers. We note that in [29] the appearance of these complexes in (12) is not justified from the standpoint of the hydrodynamic analogy.

The function (18) is valid in the following ranges of variation of the dimensionless compleses: $1 \leqslant \mathrm{Pr} \leqslant 5$; $1.3 \leqslant K \leqslant 24$; $90 \leqslant L/D \leqslant 400$; $875 \leqslant \mathrm{Re}_{\mathrm{fi}} \leqslant 3.75 \cdot 10^4$. As the authors state, the function (18) generalizes the array of experimental data better than Eqs. (15) and (17). However, both (15) and (17) are written with an error in [29]: in (15), the large value of the constant in front of $\mathrm{Re}_{\mathrm{fi}}$; while in (17), the exponent 0.5 is on the ratio ρ/ρ_v. Moreover, in [29] they distort Kutateladze's data [13], which do not depend on the parameter L/D in the treatment (15), and this actually agrees with the data of [13].

The parameter L/D also appears in a generalized function obtained by Volkov, who investigated the complete condensation of water vapor inside copper, brass, and steel pipes which

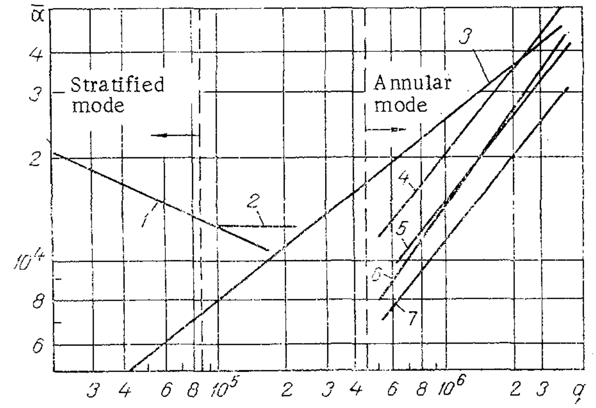


Fig. 4. Comparison of different functions for the case of condensation of water vapor inside a pipe, L = 1.5 m, D = 20 mm, P = 0.11 MN/m²: 1, 2) author's experimental data [2, 24]; 3) calculation from (19); 4) from (17); 5) (18); 6) (20); 7) (15), (16), and (23).

are practically horizontal (inclined 2° to the horizontal) [3]:

$$\frac{\bar{\alpha}}{\lambda}\left(\frac{\nu a}{g}\right)^{0,33} = 0.031\,\mathrm{Re}_{\mathrm{fi}}^{0,5}\left(\frac{D}{L}\right)^{0.2}. \tag{19}$$

In [3] nothing is said about monitoring the state of the flow at the end of the pipe, while for low heat fluxes, when the stratified mode of phase flow should exist along the entire pipe, according to (3)-(5), the experimental $\bar{\alpha}$ are considerably lower than those calculated from (14) (Fig. 4).

Vapor condensation for the annular mode of phase flow in structure and turbulent–turbulent in character was also analyzed in [19]. The authors stated that with vapor condensation inside horizontal pipes, annular flow occurs along most of the pipe length. Starting from this assumption, the authors of [19] use the function

$$\frac{\alpha\mu}{\lambda\rho^{0,5}} = c\mathrm{Pr}^{0,65}F_0^{0,5} \tag{20}$$

to calculate local heat transfer during vapor condensation inside a horizontal pipe, where F_0 is the sum of the forces (friction, gravity, and the impulse due to vapor condensation) acting on the condensate film. The dimensionality of F_0 according to [19] is kg/m·sec².

In [19] F_0 is defined under a number of assumptions: the absence of slippage between vapor and liquid at the vapor–liquid interface, equality of the pressure losses to friction with condensation to those for an adiabatic stream, which is determined from the function of Lockhart and Martinelli [5, 21].

The exponent to Pr and the constant c were found experimentally in [19]. Although the authors of [19] consider that their function satisfactorily generalizes the experimental data of reports which are absent from library collections of the USSR, their scatter of ±100% relative to the line calculated from (20) is very large. Therefore, it cannot be considered better than the function (15), which is a lot simpler.

Finally, let us consider two more reports [30, 33] where the condensation of a moving vapor inside a horizontal pipe is also investigated. A distinctive feature of these reports consists in the introduction of the ratio of the current and critical pressures P and P_{cr} into the empirical function for calculating heat transfer.

In [30] the function for the local α has the form

$$\alpha = \alpha_0\left[(1-x)^{0.8} + \frac{3{,}8x^{0.76}(1-x)^{0.04}}{P_0}\right], \tag{21}$$

where $P_0 = P/P_{cr}$ is found from equations for one-phase turbulent flow of a fluid.

Let us determine $\bar{\alpha}$ for a pipe with a length L:

$$\bar{\alpha} = \frac{1}{L}\int_0^L \alpha dL. \tag{22}$$

For complete condensation (21) takes the form

$$\bar{\alpha} = \alpha_0 \left(0.55 + \frac{2.09}{P_0^{0.38}}\right). \tag{23}$$

We note that the function (21) and (23) were obtained purely empirically without any physical grounding from the analogy, adopted by the author of [30], between the mechanisms of heat transfer during the film condensation of a moving vapor and boiling with forced motion.

In [30] it is shown that the functions (21) and (23) generalize, to within ±16%, numerous data of different authors on condensation in the annular mode in a wide range of variation of saturation temperature, heat fluxes, pipe diameter, Re_{fi} and Pr, and the physical properties of the liquid.

Somewhat earlier than in [30], Borishanskii et al. [33] used the ratio P/P_{cr} in the calculating function to analyze data on vapor condensation inside pipes, including horizontal ones.

The appearance of the relative pressure P_0 in the calculating equation emerges in [33] in connection with the application of the law of corresponding states. The form of the function $f(P_0)$ is chosen empirically for different P_0. Then one assigns the empirical relations between $\bar{\alpha}$ and q, L, $f(P_0)$, the critical pressure, the temperature, and the molecular weight of the condensate. These functions are different for a stationary and for a slowly and rapidly moving vapor. But these concepts were not made specific in [33].

Thus, in contrast to the stratified mode of phase flow, for which there is a certain clarity in the calculation of the average heat transfer, for vapor condensation under the conditions of the annular mode of flow presumed by the authors there are considerably different recommendations on the calculation of $\bar{\alpha}$. To illustrate this statement, in Fig. 4 we present lines calculated from different equations for the case of the complete condensation of water vapor with P = 0.11 MN/m² inside a pipe with L = 1.5 m and D = 20 mm. The presentation of the calculated results in the explicit form $\bar{\alpha} = f(q)$ is explained by the different structures of Eqs. (14)-(20) and (23). Lines corresponding to the limits of existence of the stratified or annular modes of phase flow over most of the pipe length, from (3) and (4), are also plotted in the figure.

As is seen in Fig. 4, the calculated and experimental lines 1 [of Eq. (14)], 2 and 3 of (19) for $q > 2 \cdot 10^5$ W/m², and 4 of (17) correlate best with the mode of phase flow. Equations (15), (16), (18), (23) disagree considerably with each other. With a change in pressure either way from P = 0.11 MN/m² one observes progressive disagreement in the $\bar{\alpha}$ calculated from (15) and (23). The same disagreement in the $\bar{\alpha}$ calculated from different functions as in Fig. 4 occurs for other pressures and the case of partial vapor condensation. It is noteworthy that in the absence of vapor condensation, when $x_2 = 1$, only Eq. (17) yields the function sought for the flow of a one-phase liquid to within a constant. The correlations (15) and (16) yield considerably lower $\bar{\alpha}$, while the functions (18) and (23) cannot be used at all in this case, since they contain dimensionless complexes not characteristic for the description of one-phase flow.

It also follows from Fig. 4 that the behavior of heat exchange is least clear in the region where the influence of the forces of interphase friction and gravity on the hydrodynamics of the condensate film is comparable. In [2] and [6] it is proposed to find $\bar{\alpha}$ in this region through a linear interpolation over J_2 between the values of $\bar{\alpha}$ calculated from (14) and (17). This suggestion requires experimental justification, however.

In conclusion, let us define the most urgent tasks in the investigation of vapor condensation inside horizontal pipes. For this the local thicknesses of the condensate film and the heat transfer in a specific pipe cross section and along the pipe perimeter must be investigated. Together with viewing of the process and measurement of the fraction of liquid removed with the vapor and the hydraulic resistance, this permits a strict identification of the mode of phase flow and a basis for the use of functions for the stratified, annular, or intermittent modes of phase flow.

NOTATION

L, D, length and inside diameter of pipe; λ, a, ν, μ, σ, r, thermal conductivity, thermal diffusivity, coefficients of kinematic and dynamic viscosity, surface tension, and heat of vaporization of liquid, respectively; ρ_v, μ_v, density and coefficient of dynamic viscosity of vapor; W_v, G_v, linear and mass velocities of vapor; x, vapor content by weight; C_f, coefficient of friction; ΔT, temperature drop; q, heat flux density; $Nu_{fi} = \frac{\alpha}{\lambda}\left[\frac{\nu^2}{g\left(1-\frac{\rho_v}{\rho}\right)}\right]^{0.33}$; $Nu_D = \frac{\alpha D}{\lambda}$; $Re_D = \frac{qD}{r\mu}$; $Re_v = \frac{W_v D \rho_v}{\mu_v}$; $Re_{fi} = \frac{qL}{r\mu}$; $K = \frac{r}{C_p \Delta T}$.

LITERATURE CITED

1. L. S. Fletcher, V. Sernas, and L. S. Galowin, "Evaporation from thin water films on horizontal tubes," Ind. Eng. Chem. Process Des. Develop., 13, No. 3, 265-269 (1974).
2. V. G. Rifert and V. Yu. Zadiraka, "Condensation of water vapor inside smooth and profiles horizontal pipes," Teploenergetika, No. 8, 77-80 (1978).
3. D. I. Volkov, "Generalization of experimental data on heat transfer during vapor condensation inside a horizontal pipe," Tr. Tsentr. Kotloturb, Inst., No. 101, pp. 295-305 (1970).
4. G. N. Danilova et al., in: Heat Exchangers for Refrigerating Installations [in Russian], G. N. Danilova, S. N. Bogdanov, O. P. Ivanov, et al., Mashinostroenie, Moscow (1973), p. 356.
5. D. Butterworth and G. F. Hewitt (eds.), Two-Phase Flow and Heat Transfer, Oxford Univ. Press, Oxford—New York (1977).
6. G. Breber, J. W. P. Palen, and J. Taborek, "Prediction of horizontal tubeside condensation of pure components using flow regime criteria," Trans. ASME, J. Heat Transfer, 102, No. 3, 471-476 (1980).
7. S. S. Kutateladze, Fundamentals of the Theory of Heat Exchange [in Russian], Nauka, Novosibirsk (1970).
8. L. D. Boiko, "Hydraulic resistance during vapor condensation in a horizontal pipe," Teploenergetika, No. 1, 41-46 (1968).
9. K. J. Chaddo, "Film condensation of vapors in horizontal tubes," Refrig. Eng., 65, No. 4, 90-95 (1957).
10. J. C. Chato, "Laminar condensation inside horizontal and inclined tubes," ASHRAE J., 4, 52-60 (1962).
11. I. Rather and S. Kesios, "Analysis of a two-phase, one-component, stratified stream of condensate," Trans. ASME, J. Heat Transfer, 88, No. 3 (1966).
12. P. K. Sarma, A. S. P. Sarma, and K. V. Apparao, "Predictions of flow-level angles in two-phase one-component stratified flow," Trans. ASME, J. Heat Transfer, 93, No. 2, 254-256 (1971).
13. S. S. Kutateladze, "Heat transfer during film condensation of vapor inside a horizontal pipe," in: Problems of the Heat Transfer and Hydraulics of Two-Phase Media [in Russian], Gos. Energet. Izd., Moscow (1961), pp. 138-156.
14. V. V. Konsetov, "On heat transfer during vapor condensation inside horizontal pipes," Izv. Vyssh. Uchebn. Zaved., Energ., No. 12, 68-75 (1961).
15. V. B. Khabenskii, V. S. Grabovskii, and P. A. Morozov, "Condensation during stratified flow of a two-phase stream inside horizontal and inclined pipes," in: Heat Exchange and Hydrodynamics [in Russian], Nauka, Leningrad (1977), pp. 102-114.
16. V. B. Khabenskii, V. S. Grabovskii, and P. A. Morozov, "Condensation levels in horizontal and inclined pipes in the stratified mode of flow," in: Heat Exchange and Hydrodynamics [in Russian], Nauka, Leningrad (1977), pp. 180-189.
17. V. V. Konsetov, "An experimental investigation of heat transfer during condensation of water vapor inside horizontal and slightly inclined pipes," Teploenergetika, No. 12, 67-71 (1960).
18. L. D. Boiko and G. N. Kruzhilin, "Heat transfer during vapor condensation in a pipe," Izv. Akad. Nauk SSSR, Energ. Transp., No. 5 (1966).
19. M. Solimen, J. R. Schuster, and P. J. Berenson, "General heat transfer correlation for annular flow condensation," ASME Pap. 67-WA/HT 12 (1968).
20. G. P. Verkhiver, G. F. Smirnov, and S. D. Tetel'baum, "Analytical determination of the coefficient of heat transfer during vapor condensation inside pipes under large weight loads," Teploenergetika, No. 11, 74-79 (1967).
21. L. S. Tong, Boiling Heat Transfer and Two-Phase Flow, Wiley, New York (1965).
22. P. N. Romanenko and A. B. Levin, "Heat transfer during the condensation of Freon-12 inside a pipe," Kholod. Tekh., No. 6, 22-26 (1969).

23. J. A. Myers and H. F. Rosson, "Condensing coefficient inside a horizontal tube near atmospheric pressure," Chem. Eng. Prog., Symp. Ser., 57, No. 32, 150-155 (1961).
24. J. V. Putilin, V. L. Podbereznev, V. G. Rifert, et al., Proceedings of the Seventh International Symposium on Fresh Water from the Sea, Vol. 1 (1980), pp. 241-252.
25. D. M. Ioffe, "Condensers with air cooling for small refrigerators," Kholod. Tekh., No. 5 (1958).
26. S. A. Gorodinskaya, On the Generalization of Test Data on Heat Transfer during Vapor Condensation inside Horizontal Pipes [in Russian], Kiev. Politekh. Inst., Kiev (1955), Part 18.
27. V. D. Yusufova, "An experimental investigation of the process of heat transfer during benzene and toluene condensation inside horizontal condensers," Izv. Akad. Nauk Azerb. SSR, No. 2, 22-25 (1953).
28. Z. L. Miropol'skii, "Heat transfer during the condensation of high-pressure vapor inside pipes," Teploenergetika, No. 3, 79-83 (1962).
29. V. Narayana and P. K. Sarma, "Condensation heat transfer inside horizontal tubes," Can. J. Chem. Eng., 50, No. 18, 547-549 (1972).
30. M. M. Shah, "A general correlation for heat transfer during film condensation inside pipes," Int. J. Heat Mass Transfer, 22, 547-556 (1979).
31. P. Ananiev, L. D. Boyko, and G. N. Kruzhilin, "Heat transfer in the presence of steam condensation in a horizontal tube," Int. Dev. Heat Transfer, Part 2, 290 (1961).
32. D. Razavi and E. K. Clutterbuck, "Total condensation inside a horizontal tube," Chem. Ind., 2, 205-206 (1974).
33. V. M. Borishanskii, D. I. Volkov, N. I. Ivashchenko, and O. P. Krektunov, "The use of the method of thermodynamic similarity to generalize test data on film condensation," in: The Temperature Regime of a Vapor-Generating Surface [in Russian], Tr. Tsentr. Kotloturb. Inst., No. 131 (1975).