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HIGH-RATE CONDENSATION PROCESS THEORY OF VAPOUR FLOW INSIDE A VERTICAL CYLINDER

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Abstract—On the basis of von Kármán integral relation a process of film condensation of moving vapour inside a vertical cylinder is considered. Relations are obtained for determination of interphase friction and pressure gradient in a condensation process in a tube. Heat transfer process is analysed for the cases of laminar and turbulent condensate film with regard for both the dynamic effect of a vapour flow and the effect of gravity forces and pressure gradient. The results of the analysis are compared with the available experimental data.

NOMENCLATURE

x,	longitudinal coordinate;
y,	transverse coordinate;
U_0,	vapour velocity at the cylinder entrance;
U,	vapour velocity outside the boundary layer;
U_x,	local velocity;
$\overline{U}'_x$, $\overline{U}''_x$,	average velocities of liquid and vapour in the cross-section x, respectively;
D, L,	diameter and length of the cylinder, respectively;
L_0,	length of starting section;
τ_s,	tangential stress at the phase interface;
τ_w,	tangential stress at the surface of a cylinder;
t,	temperature;
Δt,	difference between the vapour saturation temperature and temperature of cooling surface;
P,	static vapour pressure;
ΔP,	pressure drop in a cylinder;
q,	specific heat flux;
α_x,	local heat transfer coefficient;
M_x,	mass of flow rate vapour in the x cross-section;
δ_x,	local thickness of condensate film;
ξ_x,	friction resistance coefficient in the cross-section x;
λ, a, μ,	thermal conductivity, thermal diffusivity and dynamic viscosity of liquid;
ν', ν'',	kinematic viscosity of liquid and vapour, respectively;
ρ', ρ'',	densities of liquid and vapour, respectively;
g,	acceleration of gravity;
δ,	boundary layer thickness;
ϑ,	momentum loss thickness;
δ^*,	displacement thickness;
v_0,	velocity of uniform suction;
γ^*,	$= \rho' g - \dfrac{dP}{dx}$;
Re,	$= \dfrac{4qx}{r\mu}$;
Re^*_x,	$= \dfrac{x\sqrt{\left(\dfrac{qU''_x}{r\rho'}\right)}}{v}$;
Nu_x,	$= \dfrac{\alpha_x \cdot x}{\lambda}$;

Pr, $= \frac{\nu}{a}$;

v^*, $= \sqrt{\frac{\tau_w}{\rho'}}$

y^+, $= \frac{yv^*}{\nu}$;

ω^+, $= \frac{u_x}{v^*}$;

δ^+, $= \frac{\delta v^*}{\nu}$.

AN ANALYSIS of a film condensation process of moving vapour inside a cylinder for a case of axisymmetric and separated flow of vapour and condensate has been an object of a number of theoretical studies [1–3]. In these works tangential stresses at the phase interface are determined from the well-known relations of friction between a single-phase turbulent flow and a dry surface of an impenetrable cylindrical channel. In [4] the process is analyzed on the basis of the analogy between hydraulic friction resistance and heat transfer. In this work the case of axisymmetric friction is also considered, and vapour–liquid mixture was assumed to move in a core of a two-phase flow in a tube, liquid droplet exchange occurring continuously between this mixture and the turbulent condensate film. The analysis of laminar film condensation in a vapour flow on a plate based on consideration of a boundary layer vapour flow using momentum equation [5, 6] has revealed essential dependence of interphase friction on phase conversion rate. Existence of this dependence follows also from the results of numerical solutions of two-phase laminar boundary layer equations for various cases of condensation on a plate obtained in [7–9]. The results of the above works indicate the necessity of the effect of the phase conversion on condensation hydrodynamics of moving vapour to be taken into consideration in analysing a condensation process inside a cylinder which is not a boundary layer problem in a general case. This necessity is quite evident that the essential effect of phase conversion on hydrodynamics of the process considered was originally found experimentally in the case of condensation in a tube in the well-known measurements of the velocity field in a flow of condensating vapour carried out by Jakob and collaborators [10].

The results of the theoretical investigation of a high-rate vapour film condensation inside a vertical cylinder are given below. A two-region model of a flow inside a cylinder (a thin liquid film on a wall and a flow of pure saturated vapour in the remaining part of the effective cross section) has been adopted for the analysis. In the analysis of the vapour flow the thickness of a condensate film compared to the cylinder radius and the interface velocity compared to the velocities in the flow core are neglected. A constant of the vapour density along the channel length is also assumed. All solutions are obtained for the case of the constant specific heat flux along the cylinder (q = const). The model adopted and coordinate system are shown in Fig. 1.

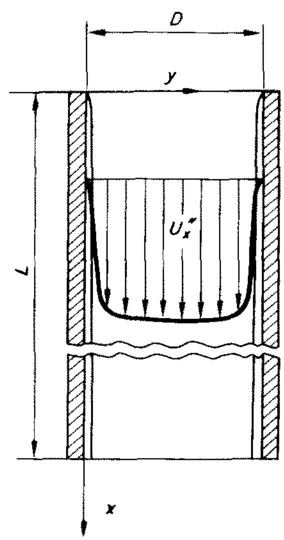

FIG. 1.

1. RELATIONSHIPS FOR INTERPHASE FRICTION AND CALCULATION OF A PRESSURE GRADIENT

Methods of the boundary layer theory are generally not applicable for the analysis of a flow inside a cylinder. However, the solution, given by Schiller [11] has demonstrated that the ideas underlying the integral Kármán relation can successfully be used in the analysis of a flow in the initial cylinder length. Assuming that the impulse, pressure drop and friction forces are balanced, Schiller presented the velocity distribution as a straight line for the flow velocity in the core and as sections of two parabolic lines for the velocity at the wall. The comparison of the obtained solution with the measured velocity fields in a tube by Nikuradze has shown good agreements for the first third of the starting length [11]. This result allows a conclusion, that calculation of the flow inside a tube using balance momentum relations is valid for the case when a boundary layer developing at the wall still occupies a small part of the total channel cross-section. In the case of vapour flow condensation the condensation process, equivalent practically to the suction of a vapour boundary layer, constitutes a sharp limitation to the development of the latter along the channel length. Moreover, suction of a boundary layer is known to prevent also from transition to a turbulent flow. If we take into consideration the said peculiarities of a condensation flow, it may be assumed that with adequate rate of the condensation process, the boundary layer of a vapour flow would occupy a small portion of the effective cross-section and velocity distribution along the whole tube length would remain the same as that at the entrance section of the duct. In this case the flow in a cylinder may be described with good approximation using balance momentum relations.

The momentum equation for incompressible boundary layer with suction is written in the following form

$$U^2 \frac{d\partial}{dx} + (2\vartheta + \delta^*) U \frac{dU}{dx} - V_0 U = \frac{\tau_s}{\rho''}. \quad (1)$$

Since in case of condensation $V_0 = -q/r\rho''$, the case under investigation corresponds to the boundary layer flow with uniform suction. If we take into consideration that with asymptotic velocity distribution δ, ϑ and δ^* are independent of the flow velocity outside the boundary layer and the longitudinal coordinate x [11], it may be assumed that after the starting section of a certain length the boundary layer is stabilized. In this case equation (I) becomes of the form

$$U \frac{dU}{dx}(2\vartheta + \delta^*) - V_0 U = \frac{\tau_s}{\rho''}. \quad (2)$$

With further assumption that the velocity in the core u changes along the cylinder length as the average velocity U_x'', we obtain for dU/dx:

$$\frac{dU}{dx} = \frac{d}{dx}\left(U_0 - \frac{4qx}{r\rho'}\right) = -\frac{4q}{r\rho'' D} = \frac{4V_0}{D}. \quad (3)$$

Substitution of equation (3) into (2) gives

$$\frac{\tau_s}{\rho''} = -V_0 U\left[1 - \frac{4(2\vartheta + \delta^*)}{D}\right]. \quad (4)$$

If we take into consideration that the velocity in the core U is somewhat greater than the average velocity $\bar{U}_x''$ and $1 - 4(2\vartheta + \delta^*)/D$ is somewhat less than unity, we may write

$$U\left[1 - \frac{4(2\vartheta + \delta^*)}{D}\right] \approx \bar{U}_x''. \quad (5)$$

Correspondingly equation (4) takes the form:

$$\tau_s = -\rho'' V_0 \bar{U}_x'' = \frac{q}{r}\bar{U}_x''. \quad (6)$$

It should be noted that with very high condensation rates i.e. when $U \to U_x''$ and $1 - 4(2\vartheta + \delta^*)/D$ is practically equal to unity equation (6) becomes in the limit a rigorous solution of momentum equation.

Let us determine the domain of validity of the friction interphase law.

As it was indicated above, the momentum equation with good accuracy describes the flow of the first third of the starting section of a circular tube. Taking into consideration that at the end of this part of the starting section, in a laminar flow in the impenetrable cylinder the boundary layer thickness achieves 30 per cent of the tube radius, we may write the following condition for validity of the present analysis

$$\frac{\delta}{D} < 0{\cdot}15. \tag{7}$$

In a flow with uniform suction asymptotic thickness of the boundary layer is defined by the following relation:

$$\delta = 4{\cdot}6\,\frac{v''}{-V_0}.$$

Accordingly, condition (7) attains the form

$$\frac{4{\cdot}6v''}{-V_0 D} < 0{\cdot}15. \tag{8}$$

Having rewritten equation (8) in terms of parameters of the condensation process, we obtain the following final form of the condition of validity of the present approximate solution

$$\frac{v'' r\rho''}{qD} < 0{\cdot}033. \tag{9}$$

It should be noted, that in the case of steam condensation in a tube, 20mm i.d., the condition (9) is satisfied under pressure 1atm with heat fluxes greater than 40 000 W/m^2, under pressure 0·035 atm with $q > 35000$ W/m^2, and under pressure 200 atm with $q > 25000$ W/m^2. The above examples show that the adopted law of interphase friction includes the overwhelming majority of the cases of interest for engineering. It should be also noted that the maximum error of approximate equality (5) which holds when $\delta/D = 0{\cdot}15$, is 7 per cent. Taking into account that as follows from the analysis below, the heat transfer coefficient is proportional to a square root of the vapour flow velocity, it may be concluded that for processes satisfying inequality (9), the error in α_x, determined by approximate formula (5) is negligibly small.

The asymptotic velocity distribution which serves as the basis for derivation of equation (6) is obtained, achieved after the so-called starting section, whose length is found from the following condition:

$$\left(\frac{-V_0}{U}\right)\sqrt{\left(\frac{UL_0}{v''}\right)} = 2. \tag{10}$$

Taking into account that under conditions of complete condensation inside a tube the core velocity in the section $x = L_0$ is determined from the relation

$$U = \frac{4q(L - L_0)}{Dr\rho''}$$

we obtain

$$\frac{L}{L - L_0} = 16\,\frac{r\rho'' v''}{qD}. \tag{11}$$

Accordingly, the maximum fraction of the starting section in the total length of the condensation channel for cases, satisfying condition (9) is

$$\left(\frac{L_0}{L}\right)_{\max} = 0{\cdot}35. \tag{12}$$

Thus, the asymptotic velocity profile in the processes under consideration becomes developed in the worst case approximately to the end of the first third of the tube.

For practical calculations the above length of the starting section may considerably be reduced without essential errors in α_x. Particularly, if the interphase friction law (6) is extended to that tube section where the tangential stress is exact 20% higher than that calculated by equation (6), then the error in α_x at this point is 10 per cent (since the relation $\alpha_x \sim \sqrt{(\tau_s)}$ holds). From the data of [11] the length of the starting section corresponding to such approximation, is determined by the equality

$$\left(\frac{-V_0}{U}\right)\sqrt{\left(\frac{UL_0}{v''}\right)} = 0{\cdot}8 \tag{13}$$

which leads to the following relation for L_0:

$$\frac{L}{L - L_0} = 2{\cdot}56\,\frac{r\rho'' v''}{qD}. \tag{14}$$

The maximum length of the starting section in this approximation with $v''r\rho''/qD = 0.033$ is only 8 per cent of the total length of the condensation tube. With higher heat fluxes the length of this section will in general be inessential.

Having obtained the interphase friction law, it is possible to determine the pressure gradient along the cylinder length. Pressure change in a cylinder in the case under condensation is controlled by two factors: on the one hand, the friction at the cylinder surface results in the pressure drop, on the other hand, the decrease of vapour flow rate along the duct length due to condensation results in deceleration of the vapour flow and, correspondingly, increase of the static vapour pressure.

Change of the kinetic energy of the vapour consumption per second between sections x and $x + \mathrm{d}x$ is

$$-\frac{\pi D^2\rho''}{8}\bar{U}_x''^3 + \frac{\pi D^2\rho''}{8}\left(\bar{U}_x'' - \frac{4q\mathrm{d}x}{r\rho''D}\right)^3 = -\frac{3\bar{U}_x''^2\cdot q\cdot \mathrm{d}x\cdot \pi D}{2r} \quad (15)$$

if the terms of higher order of smallness are neglected.

The above decrease of the kinetic energy of a vapour flow must be compensated by increase in the potential pressure energy of the vapour consumption per second. The part of the total pressure gradient corresponding to this increase of the potential energy will be determined in the form

$$\left(\frac{\mathrm{d}P}{\mathrm{d}x}\right)_{\mathrm{pot}} = \frac{3\pi D\bar{U}_x''^2\cdot q\cdot \mathrm{d}x}{2r\cdot \frac{1}{4}\pi D^2\cdot \bar{U}_x''\cdot \mathrm{d}x} = \frac{6\bar{U}_x''q}{rD}. \quad (16)$$

Further, taking into account that τ_w and τ_s may be assumed as a fair approximation, we finally arrive to the expression for the total pressure gradient in a cylinder.

$$\frac{\mathrm{d}P}{\mathrm{d}x} = -\frac{4\tau_s}{D} + \left(\frac{\mathrm{d}P}{\mathrm{d}x}\right)_{\mathrm{pot}} = \frac{2\bar{U}_x''q}{rD}. \quad (17)$$

As follows from equation (17), pressure along the duct length should always increase at sufficiently high rates of condensation. It should be noted that such a behaviour of pressure in a condensation vapour flow was found in experiments with thermal pipes [12].

Integrating equation (17) from x_1 to x_2, we obtain the expression for pressure drop between two cylinder cross-sections:

$$p_2 - p_1 = \int_{x_1}^{x_2} \frac{2q}{rD}\left(U_0 - \frac{4qx}{r\rho''D}\right)\mathrm{d}x = \frac{2q}{r}\frac{x_2 - x_1}{D}\left[U_0 - \frac{2q}{r\rho''}\left(\frac{x_2 - x_1}{D}\right)\right]. \quad (18)$$

Expression (18) is valid for cases when the pressure drop is smaller than the absolute pressure P (in view of the assumption $\rho'' = \mathrm{const}$). To use this expression at high pressure drops, it is necessary to carry out successive calculation over several individual sections (at each section appropriate constant pressure ρ'' is assumed) and to sum the results obtained. For conclusion it should be noted, that expressions (6) and (17) as follows from the succession of their obtaining, are valid irrespective of the cylinder orientation in the gravity field and irrespective of what kind of thermal resistance dominates in the condensation process (thermal resistance of a condensate film, resistance of the phase transition or contact resistance between the film and cylinder surface).

2. HEAT TRANSFER IN LAMINAR FLOW OF A CONDENSATE FILM

The process of vapour condensation in a descending flow inside a vertical cylinder is considered. The resistance of a laminar condensate film (condensation of nonmetallic liquids) is a dominant resistance. A plane, steady liquid motion is considered. On the basis of work [5, 6] in the analysis of the problem we neglect the inertia and superheat of condensate particles are neglected. Interphase friction is determined by equation (6). The effect of gravity forces and the pressure gradient are accounted for. The pressure gradient is determined by equation (17).

With the assumptions adopted the equations which describe the process, will be of the form

$$\mu \frac{d^2 U_x'}{dy^2} + \gamma^* = 0; \ \frac{d}{dx}(U_x' \cdot \delta_x \cdot \rho') = \frac{q}{r}; \quad (19)$$

$$\alpha_x = \frac{\lambda}{\delta_x}.$$

We shall find the solution with the following boundary conditions

$$U_x' = 0 \quad q = \text{const. with } y = 0$$

$$\frac{dU_x'}{dy} = \frac{q}{r\mu}\overline{U}_x''. \quad q = \text{const. with } y = \delta_x \quad (20)$$

$$\delta_x = 0 \quad \text{with} \quad x = 0.$$

The solution of system (19) with the boundary conditions (20) gives the following cubic equation for local heat transfer coefficient

$$\alpha_x^3 - \frac{\overline{U}_x''\lambda^2\rho'}{2x\mu}\alpha_x - \frac{\lambda^3\gamma^*\rho' r}{3qx\mu} = 0. \quad (21)$$

The effect of wave flow of a condensate film should be taken into account in deriving the final relation for heat transfer coefficient. Since the effect of moving vapour favours the waveformation in the film, so in case of condensation in moving vapour the intensifying effect of waves on heat transfer must be maximum. According to experimental work [13] in condensation of stagnation vapour on a vertical tube the maximum correction for wave behaviour of the flow is 1·6. Comparison of the solution (21) with the results presented in [4] shows that in the case of flowing vapour in the expression for the heat transfer coefficient a correction factor of 1·7 should be introduced. With this correction factor for wave behaviour of the flow introduced, the solution of the equation (21) will be of the form

$$\alpha_x = 0{\cdot}935 \times \left[\left[\sqrt[3]{\left\{\frac{\lambda^3\rho'}{\mu x}\left[\frac{r\gamma^*}{q} - \sqrt{\left(\frac{r\gamma^*}{q}\right)^2 - \frac{2\overline{U}_x''\rho'}{6\mu x}}\right]\right\}} + \sqrt[3]{\left\{\frac{\lambda^3\rho'}{\mu x}\left[\frac{r\gamma^*}{q} + \sqrt{\left(\left(\frac{r\gamma^*}{q}\right)^2 - \frac{\overline{U}_x''\rho'}{6\mu x}\right)}\right]\right\}}\right]\right] \quad (22)$$

with

$$\left(\frac{\lambda^3\gamma^*\rho' r}{6qx\mu}\right)^2 > \left(\frac{\lambda^2\overline{U}_x''\rho'}{6\mu x}\right)^3$$

$$\alpha_x = 1{\cdot}32\sqrt{\left(\frac{\overline{U}_x''\lambda^2\rho'}{x\mu}\right)} \times \cos\frac{\arccos\frac{\gamma^* r}{q}\sqrt{\left(\frac{6\mu x}{\overline{U}_x''\rho'}\right)}}{3} \quad (23)$$

with

$$\left(\frac{\lambda^3\gamma^*\rho' r}{6qx\mu}\right)^2 < \left(\frac{\lambda^2\overline{U}_x''\rho'}{6\mu x}\right)^3.$$

In the case with $\gamma^* = 0$ we shall have

$$\alpha_x = 1{\cdot}2\sqrt{\left(\frac{\lambda^2\rho'}{x\mu}\right)}\sqrt{\left(U_0 - \frac{4qx}{r\rho'' D}\right)}. \quad (24)$$

3. HEAT TRANSFER IN A TURBULENT FLOW OF CONDENSATE FILM

The model of a process considered above is analysed with regard for turbulent momentum and energy transfer. The analogy between heat transfer and the hydraulic resistance of friction according to Karman is taken as the basis for analysis.

Equations of heat flux and tengential stress in a turbulent condensate film are written in the following form

$$q = \lambda\left(1 + Pr\frac{\varepsilon_q}{\nu}\right)\frac{dt}{dy} \quad \tau_0 = \mu\left(1 + \frac{\varepsilon_\tau}{\nu}\right)\frac{dU_x'}{dy}. \quad (25)$$

For nonmetallic liquids we assume the turbulent transfer coefficient to be equal to transfer and the tangential stress coefficient

$$\varepsilon_q = \varepsilon_\tau = \varepsilon.$$

For solving the heat problem the value ε/ν is to be determined from the solution of the corresponding hydrodynamic problem.

The three-layer model of the flow in a condensate film is assumed:

The laminar region

$$\omega^+ = y^+ \quad (0 \leqslant y^+ \leqslant 5).$$

The transient region

$$\omega^+ = -3{\cdot}05 + 5\ln y^+ \quad (5 \leqslant y^+ \leqslant 30).$$

The turbulent region

$$\omega^+ = 5{\cdot}5 + 2{\cdot}5\ln y^+ \ (30 \leqslant y^+ \leqslant \delta_x^+).$$

In the laminar sublayer and transient region friction is assumed to be τ_w and to follow the law

$$\tau_y = \gamma^* y + \tau_s. \tag{26}$$

Having determined ε/v, thermal resistance of individual layers by the second equation (25), we arrive at

$$R_1 = \frac{5v}{v^*\lambda} \tag{27}$$

$$R_2 = \frac{5v}{v^*\lambda\cdot Pr}\ln(1 + 5Pr) \tag{28}$$

$$R_3 = \frac{2{\cdot}5v}{v^*\lambda\cdot Pr}\ln\frac{(\delta_x^+/30)\,\tau_w - \tau_0}{\tau_s} \tag{29}$$

where

$$\tau_0 = \gamma^*\delta_x, \quad \tau_w = \gamma^*\delta_x + \tau_s.$$

In this case the value τ_s is determined by equation (6).

Summing up the thermic resistance of individual layers, we obtain for the local heat transfer coefficient

$$\alpha_x = \frac{v^*\lambda}{5v\left[1 + \frac{1}{Pr}\ln(1+5Pr) + \frac{1}{2Pr}\ln\frac{(\delta_x{}^+/30)\,\tau_w - \tau_0}{\tau_s}\right.}. \tag{30}$$

For calculation of α_x the values v^* and δ_x^+ should also be determined. Introducing the values τ_w and δ_x into the expression for v^*, we obtain

$$v^{*3} - \frac{\tau_s}{\rho'}v^* - gv\delta_x^+ = 0. \tag{31}$$

By the plotting the relation $\overline{U}'_x = f(\delta_x^+)$ we find the interpolation formula, which yields the satisfactory approximation within the parameter range of interest.

$$\delta_x^+ = -5450 + \sqrt{\left(29{\cdot}85\cdot 10^6 + 625\frac{qx}{vr\rho'}\right)}. \tag{32}$$

Thus, relations (30)–(32) give the final solution to the problem.

In the case of the turbulent heat transfer considered the width of the developed turbulent flow region in the condensate film (the third of regions found in the film) is essentially smaller than that in the case of a turbulent single-phase flow, where this region occupies the main portion of the duct cross-section. In view of the just said, in case of film condensation the thermal resistance of the region discussed is rather small and in the analysis of the process, it may be neglected.

$$\alpha_x = \frac{v^*\lambda}{5v[1 + (1/Pr)\ln(1 + 5Pr)]} \tag{33}$$

Further, at high vapour velocities and large heat fluxes the effect of gravity forces becomes also important. For this case the solution of a problem is considerably simplified and reduced to the following expression for the local heat transfer coefficient

$$\alpha_x = 0{\cdot}2\frac{\lambda\cdot Pr\sqrt{(qU_x''/r\rho'')}}{v[Pr + \ln(1 + 5Pr)]}. \tag{34}$$

4. DISCUSSION OF RESULTS

The results of the present analysis may be compared with the experimental data of the work [14]. Regimes realized in above work, includes both laminar and turbulent flows in a condensate film. In case of laminar film condensation, the comparison of the results of [14] with the solution (21) allows the correction factor accounting for the effect of the wave flow of a laminar film, to be determined.

For the comparison, the physical parameters of condensate, which control the transfer processes inside the liquid were taken at the average temperature of the film, but the parameters, controlling the interphase friction were taken at the temperature of vapour saturation (in the experiments discussed, the temperature of steam saturation was 123–125°C).

In Fig. 2 the present results are compared in dimensionless coordinates with the experimental data of [14]. High vapour flow velocities and specific heat fluxes realized in work [14], allow the effect of the gravitational field to be neglected in the comparison and to carry out the calculation by the simple equation (34).

As is seen from the comparison the experimental data on the local heat transfer are in good agreement with calculations by the present approximate theory of the process.*

In Fig. 3 the same experimental data are compared with the Carpenter–Colburn's equation, which for the local heat transfer may be presented in the following form

$$\frac{\alpha_x}{M_x}\left(\frac{\mu\rho''}{C_p\rho'\lambda}\right)^{\frac{1}{2}} = 0{\cdot}023\,\sqrt{(\xi_x)}. \qquad (35)$$

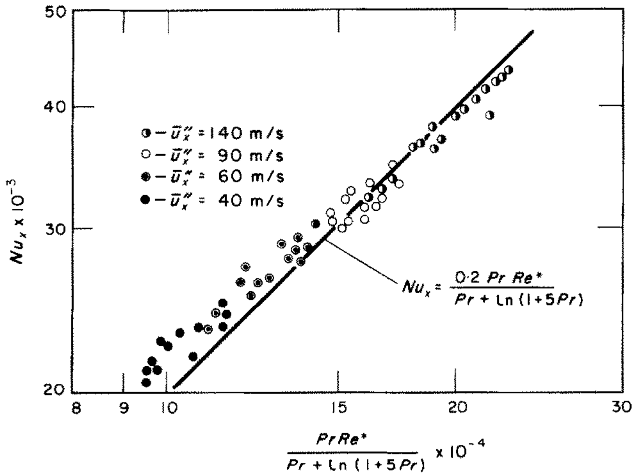

FIG. 2. Comparison of experimental data [14] with equation (34).

Light points show experimental data treated exactly following Carpenter–Colburn's theory (using Blasius' law for calculation of ξ_x). As seen from Fig. 3, there is the divergence between the theory and experiment at such treatment. At a certain velocity equation (35) gives a constant heat transfer coefficient independent of the heat flux. According to the experimental data in the case of a turbulent flow in a film at constant vapour velocity there exists a relation between the heat transfer coefficient and the specific heat flux (increase of the heat transfer rate increases with heat flux). The conclusion readily follows that the above relation is the result of increase of interphase tangential stresses with increasing heat flux. It is also clear that this fact cannot be accounted for by the ordinary relationships of turbulent friction. Therefore it is of interest to treat the experimental data in the same coordinates as in equation (35) provided

* At rather low Reynolds numbers of the film the value δ^+ calculated by equation (32) appears to be less than 30. In such a case for a more rigorous calculation by equation (34) the factor 5 at the Prandtl number in the denominator of equation (34) should be replaced by the factor ($\delta^+ - 5/5$). It should be noted that the maximum value of this correction factor for minimum Reynolds number for turbulence film to appear is only 8 per cent. In the present comparison this correction is not included.

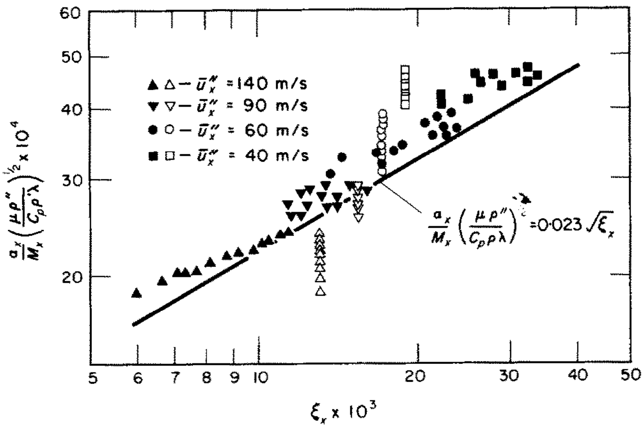

FIG. 3. Comparison of experimental data [14] with Carpenter–Colburn's equation (35).

that ξ_x is taken from the analysis of the present work. In this case the expression for the friction resistance coefficient is rewritten in the form:

$$\xi_x = \frac{8\tau_s}{\rho''\bar{U}_x''^2} = \frac{8q}{r\rho''\bar{U}_x''}. \qquad (36)$$

The experimental data correlated by equation (36) are plotted in Fig. 3 by blackened points. As seen from the comparison, in this case the experimental points fall with good accuracy along the single curve correlated by the equation

$$\alpha_x = 0.065\left(\frac{qC_p\lambda\bar{U}_x''}{vr}\right)^{\frac{1}{2}}.$$

The results of comparison show that the defect of Karpenter and Colburn's theory is the estimation of interphase friction forces from the relationships of the ordinary "dry" turbulent friction. Equation (37) seems to be practically equivalent to equation (34). On the other hand, the simplicity of these relations makes them convenient in practical calculations.

The results of the work allow to make a conclusion, that the account for the effect of the phase of the transformation process on interphase friction relations is the necessary condition for the analysis of film condensation process of moving vapour in a vertical tube.

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THEORIE DU PROCESSUS DE CONDENSATION A GRAND FLUX D'UN ECOULEMENT DE VAPEUR DANS UN CYLINDRE VERTICAL

Résumé—On a considéré sur la base de la relation intégrale de Karman un processus de condensation en film de vapeur en mouvement dans un cylindre vertical. On a obtenu des relations pour la détermination du frottement interphase et du gradient de pression dans un processus de condensation à l'intérieur d'un tube. Le processus du transfert thermique est analysé pour les cas d'un film de condensat laminaire et turbulent par considération à la fois de l'effet dynamique d'un écoulement de vapeur et de l'effet de forces de gravité et de gradient de pression. Les résultats de l'analyse sont comparés aux résultats expérimentaux disponibles.

KONDENSATIONSTHEORIE FÜR EINE DAMPFSTRÖMUNG IM SENKRECHTEN ROHR BEI HOHEM UMSATZ

Zusammenfassung—Aufgrund der Karman'schen Integralbeziehung wird der Vorgang der Filmkondensation bei Dampfströmung im senkrechten Rohr betrachtet. Es ergeben sich Beziehungen zur Bestimmung der Reibung zwischen den Phasen und des Druckgradienten bei Kondensation in einem Rohr. Der Wärmeübergang wurde für die beiden Fälle des laminaren und des turbulenten Kondensatfilmes untersucht mit Berücksichtigung des dynamischen Einflusses des Dampfstromes, der Schwerkraft und des Druckgradienten. Die Ergebnisse der Analyse wurden mit den verfügbaren Versuchsdaten verglichen.

ТЕОРИЯ ИНТЕНСИВНОГО ПРОЦЕССА КОНДЕНСАЦИИ ПАРОВОГО ПОТОКА ВНУТРИ ВЕРТИКАЛЬНОГО ЦИЛИНДРА

Аннотация—На основе интегрального соотношения Кармана рассмотрен процесс пленочной конденсации движущегося пара внутри вертикального цилиндра. Получены зависимости для определения межфазного трения и градиента давления в процессе конденсации в трубе. Анализ процесса теплоотдачи проведен для случаев ламинарной и турбулентной пленки конденсата с учетом как динамического воздействия парового потока, так и влияния сил тяжести и градиента давления. Результаты анализа сопоставлены с имеющимися экспериментальными данными.