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Heat transfer and flow modes of phases in laminar film vapour condensation inside a horizontal tube

V. G. RIFERT
Kiev Polytechnical Institute, Kiev, U.S.S.R.

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Abstract—By using the gradient method for heat transfer investigation, local heat transfer coefficients are determined for laminar film vapour condensation inside a horizontal tube. A study is made of the effect of vapour velocity, condensate inflow, and transverse mass flux on the intensity of the process. The occurrence of a pronounced asymmetry in the local heat transfer coefficients around the tube perimeter is noted, the decrease of which is promoted by an increase in vapour velocity and local heat flux. The flow modes of phases in the case of incomplete vapour condensation in a tube are identified; the need to take into account the condensate film thickness and the transverse mass flux effect when constructing the map of the modes is noted. A correlation is suggested to calculate local and mean-integral heat transfer coefficients for laminar film condensation.

1. INTRODUCTION

THE PROCESS of vapour condensation inside horizontal tubes is widely used in film evaporators of desalinating and refrigerating plants, pipelines of power engineering equipment and in heat pipes.

As has been shown earlier [1], there is a marked contradiction in heat transfer relations suggested by various authors and a great quantitative discrepancy between the reported measured mean coefficients of heat transfer $\bar{\alpha}_L$ from vapour to a wall.

The theoretical analysis of the process deals predominantly with two approaches to the determination of $\bar{\alpha}_L$. In one approach, for the condensation of a slowly moving vapour, an analogy is adopted between the processes taking place inside and outside of a horizontal tube, and it is recommended to calculate $\bar{\alpha}_L$ from Nusselt's relation [1–3]

$$\overline{Nu}_{\text{film}} = 0.95 Re_d^{-1/3}. \qquad (1)$$

The other approach has been developed for the prevailing effect of the forces of interphase friction and turbulent flow of both phases in the presence of strong liquid entrainment within the framework of Reynolds' hydrodynamic analogy. It is in this way that the following relation was obtained for the tube length-average heat transfer coefficient $\bar{\alpha}_L$ in ref. [4]

$$\overline{Nu}_d = C\,Re^{0.8}\,Pr^{0.43}\sqrt{(1+x_1(\rho_l/\rho_v))} + \sqrt{(1+x_2(\rho_l/\rho_v-1))} \qquad (2)$$

where the constant C has different values for copper, brass and steel tubes. Equation (2) is recommended in ref. [4] for the case when $Re \geqslant 5000$.

The analysis of experimental data [4, 5] shows that in a number of cases for $Re > 5000$ to the mean heat transfer $\bar{\alpha}_L$ with complete condensation ($x_2 = 0$) is proportional to the heat flux density in a wall which is approximately equal to 0.5 rather than 0.8 as follows from equation (2).

For the range of operational parameters within which the gravity and interphase friction forces are commensurable, there are no analytical solutions for the mean heat transfer that agree with the experiment whereas empirical correlations suggested by different authors give very dissimilar results on the mean heat transfer [1].

In this paper the case of incomplete condensation with $Re \leqslant 400$ is considered which is most characteristic for film evaporators of desalinating plants, for horizontal tubular condensers of refrigerating plants and of plants used in the chemical industry.

2. FLOW MODES OF PHASES AND HEAT TRANSFER IN INCOMPLETE VAPOUR CONDENSATION

The estimation of the flow mode of phases with condensation inside a horizontal tube is given in a few publications [3, 6, 7]. The authors of refs. [6, 7] construct the map of flow modes of phases in the coordinates $J_g - X$, where J_g and X are the dimensionless gas velocity and void fraction, respectively

$$J_g = \frac{G_{\text{mix}}x}{[dg\rho_v(\rho_l-\rho_v)]^{0.5}} \qquad (3)$$

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

When deriving equation (3), the friction coefficient at the phase interface was eliminated from the balance of friction and gravity forces as applied to a homogeneous flow. The limits for the flow modes of phases were found from equations (3) and (4) [4, 7] on the

NOMENCLATURE

a	thermal diffusivity [$m^2 s^{-1}$]
C_f	coefficient of vapour friction against a film
C_p	specific heat at constant pressure [$kJ\,kg^{-1} K^{-1}$]
d	inner diameter of a tube [m]
g	gravity acceleration [$m\,s^{-2}$]
L	length of a tube [m]
M	mass flow rate [$kg\,s^{-1}$]
P	pressure [$N\,m^{-2}$]
q_φ, $\bar{q}_\varphi$	local and φ-average heat flux densities [$W\,m^{-2}$]
R	radius of a tube [m]
r	heat of vapour generation [$kJ\,kg^{-1}$]
T_s	saturation temperature [K or °C]
W	medium velocity [$m\,s^{-1}$]
x	vapour quality of a mixture
z	coordinate along the tube axis [m].

Greek symbols

α_φ, $\bar{\alpha}_\varphi$	local and φ-average coefficients of heat transfer [$W\,m^{-2} K^{-1}$]
δ	condensate film thickness [m]
λ	thermal conductivity [$W\,m^{-1} K^{-1}$]
μ	dynamic viscosity [$N\,s\,m^{-2}$]
ν	kinematic viscosity [$m^2 s^{-1}$]
Π_r	perimeter of condensate rivulet [m]
ρ	medium density [$kg\,m^{-3}$].

Subscripts

an	annular
c	condensate
en	entrainment
film	liquid film
gr	gravitational
in	inner
L	parameter related to the full tube length (longitudinal coordinate)
l	liquid
out	outer
v	vapour
1, 2	values of the parameter at the inlet and exit from a tube, respectively.

Dimensionless groups

A	$2\nu_l\lambda_l\Delta Td/\rho_l gr$
Fr	$\rho_v(\rho_l-\rho_v)W_v^2/\rho_l^2(\nu_l g)^{2/3}$
Nu_d	$\alpha d/\lambda_l$
Nu_{film}	$(\alpha/\lambda)\,(\nu^2/g)^{1/3}$
Pr	ν/a
Re_d	$q\pi d/r\mu_l$
Re_{en}	$M_{en}/\pi d\mu$
Re_{film}	$qL/r\mu_l$
Re_r	$M_c/\Pi_r\mu$.

basis of subjective visual observations of vapour condensation and, as a result, the modes were identified which are typical of adiabatic two-phase flows: laminated and wavy, nucleate, annular.

In ref. [3] the balance of forces applied to a liquid film is considered. This approach gives a parameter which characterizes the relationship between the forces of friction and gravity

$$P_\tau = \frac{C_f\rho_v W_v^2}{2\rho_l g\delta} \tag{5}$$

which involves the friction coefficient C_f and the condensate film thickness.

Over the initial portion of the tube there is an annular flow of phases with predominantly laminar flow of condensate film under the action of the interphase friction forces. Then, the condensate film thickness can be determined from the relation [8]

$$\delta = \left(\frac{4q\nu_l z}{rC_f W_v^2\rho_v}\right)^{0.5}. \tag{6}$$

As is known [9–11], the friction coefficient C_f in a two-phase flow with condensation should be determined taking into account the mass suction from the vapour boundary layer. By analogy with ref. [11], it can be determined as

$$C_f = C_{f0}\frac{(1-0.25b)^2}{(1+0.25b)^{0.2}} \tag{7}$$

where $b = -2\tilde{J}/C_{f0}$, $\tilde{J} = q/r\rho_v W_v$, and C_{f0} is the friction coefficient for an impenetrable wall at the same numbers $Re_v = \bar{W}_v d/\nu_v$.

Taking into account equation (6) and some simple transformations P_τ will acquire the form

$$P_\tau = \frac{(C_f Fr)^{3/2}}{4Re_{film}^{1/2}}. \tag{8}$$

When the blow-through velocities are small, the gravity force becomes predominant at the end of the tube. Here, the condensate film falls down by gravity mainly in the azimuthal direction and then, as is known, $\delta \sim Re_d^{1/3}$. Therefore, for the tube cross-section, which neighbours the region with the predominant effect of gravity, this dimensionless number involves the same parameters as does equation (8). Numerical values of P_τ at the boundary were found experimentally and are given below.

When $d/2 \gg \delta$, the local heat transfer over the initial length of the tube with a laminar condensate film and annular flow of phases should be determined from Nusselt's relation [8]

$$\alpha_\varphi = 0.5\left[\frac{\lambda_l^2 C_f r\bar{W}_v^2\rho_v(\rho_l-\rho_v)}{\nu_l\rho_l qz}\right]^{0.5} \tag{9}$$

which in dimensionless form is

$$Nu_{\text{film}} = 0.5C_f^{0.5}\,Fr^{0.5}\,Re_{\text{film}}^{0.5}. \qquad (10)$$

In the case of gravitational runoff of the condensate film around the tube perimeter and relatively small amount of condensate in the rivulet, an analogy should be expected with condensation on the outer surface of a horizontal tube, when, according to ref. [8], the following relation holds for α_φ:

$$\alpha_\varphi = \lambda_l\left(\frac{A\int_0^\beta \sin^{1/3}\beta\,\mathrm{d}\beta}{\sin^{4/3}\beta}\right)^{1/4} \qquad (11)$$

and $\bar{\alpha}_\varphi$ can be determined from equation (1).

In rather long tubes, provided however that everywhere $Re_{\text{film}} \leqslant 400$, such a combination of operational parameters is possible at which there would be a stretch inside of the tube where, because of the commensurate effects of gravity and friction forces, the condensate flow will become two-dimensional and this will greatly complicate the derivation of an analytical relation for the local and mean heat transfer.

It will be assumed that, just as for the case of condensation on a vertical surface [13], the following functional relation will hold for Nu_{film} in the region with the commensurate effects of these forces

$$Nu_{\text{film}} = f(C_f, Re_{\text{film}}, Fr). \qquad (12)$$

As was noted earlier [12, 13], relation (12) for a vertical surface follows from predictions given in ref. [14].

3. EXPERIMENTAL PROCEDURE

It was noted earlier [1] that the main reason for the disagreement between experimental results of various authors is that in all the experiments the mean-integral values of $\bar{\alpha}_L$ were measured, whereas the tube can have zones with different modes of the flow of phases and of heat transfer. In ref. [12] a method for investigating condensation inside a horizontal tube is described which consists of the determination of the local (around the tube perimeter) heat fluxes q_φ and heat transfer coefficients α_φ from the temperature gradient in the wall of a thick-walled tube.

A segment of an 18 mm i.d., 83.5 mm o.d. and 83 mm long thick-walled tube made of L070-1 brass was aligned with the setting segment having a length of 1.3 m. The two segments were cooled independently by streams of water. Along the inner and outer contour in the middle portion of the test section and on equal radii at five points around the perimeter (with $\varphi = 0, \pi/4, \pi/2, 3\pi/4$ and π), copper–constantan thermocouples with 0.15 mm diameter electrodes were fitted into 0.5 mm diameter holes drilled strictly coaxially on a jig-boring machine. The distance between the thermocouples was $\Delta R = 23.5$ mm and the distance from the inner and outer walls of the tube was $\Delta R = 5$ and 2.5 mm, respectively. Moreover, on these very radii controlling thermocouples were imbedded in another tube section located 5 mm from the measuring section.

Before carrying out the runs, the measuring section was degreased with alcohol to exclude the possibility of dropwise vapour condensation.

Before each test series the inflow of air into the system was checked under vacuum. In all the tests it did not exceed 0.05% of the overall vapour flow rate.

When the tests were conducted at atmospheric pressure, measurements were preceded by the forcing of vapour through the tube for 5–7 min.

This technique allowed determination of the local heat fluxes q_φ and heat transfer α_φ around the tube perimeter by setting any values of $\tilde{J}$, Re_{film} and Re_d, Fr over the test section. This was a big advantage over the measurements of the mean-integral coefficients of heat transfer $\bar{\alpha}_L$ when a change in any of the parameters q, $\bar{W}_v$ or L entailed a change of all the rest numbers Fr, Re_{film} and $\tilde{J}$.

The experiments were conducted so that no less than 3–5 measurements could be made at the same invariable (over the test section) vapour velocity $\bar{W}_v$, φ-average heat flux density $\bar{q}_\varphi$, condensate inflow from the setting section M_c and saturation temperature T_s. The sought-after quantity was taken to be the mean of the measured values. The spread of the measured local values of α_φ about mean values did not exceed 5%.

Note also that the visual observations in all the experiments showed that there was no flooding of the entire cross-section with condensate since the vapour blow-through was present all the time. The maximum vapour velocity at the exit from the test section at $T_s = 373$ K was 0.5 m s^{-1}.

The measured wall temperatures at the points served as boundary conditions for solving the heat conduction equation. The algorithm of the solution and the associated problems of the accurate determination of q_φ and α_φ were developed according to recommendations given in ref. [15]. The greater part of the experiments were carried out at local temperature differences in the wall and between the wall and the vapour above 9 and 2 K, respectively. In all the experiments the temperature gradient in the wall in the axial direction was an order of magnitude smaller than the temperature gradient in the radial direction. An estimate made of the errors shows that α_φ is determined within 3.71%.

The φ-average heat transfer coefficient $\bar{\alpha}_\varphi$ was determined from

$$\bar{\alpha}_\varphi = \frac{\bar{q}_\varphi}{\Delta\bar{T}_\varphi}$$

where $\Delta\bar{T}_\varphi$ is the φ-average temperature difference. The experiments were carried out with steam at $T = 44$–$100°$C, $\bar{q}_\varphi = 70$–500 kW m^{-2}, $\bar{W}_v = 0.9$–60 m s^{-1}, with the amount of condensate inflowing on-

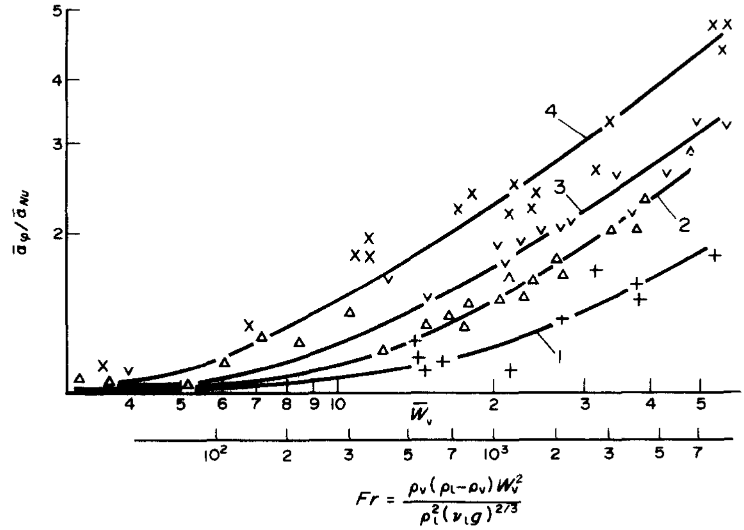

FIG. 1. Effect of vapour velocity on condensation rate inside a tube at $T_s = 100°C$: 1, $\bar{q}_\varphi = 70$; 2, 130; 3, 240; 4, 340.

to the measuring section varying within the range $M_c = 0.3–6.7 \times 10^{-3}$ kg s^{-1}. In this case the dimensionless groups varied in the following ranges: $4.5 \leqslant Fr \leqslant 8400$; $4 \leqslant Re_{film} \leqslant 400$; $10^{-3} \leqslant \tilde{J} \leqslant 5 \times 10^{-2}$; $6 \leqslant Re_d \leqslant 33$.

4. RESULTS OF INVESTIGATION

Since the flow modes of phases and the laws governing condensation depend not only on $\bar{W}_v$, but also on the transverse mass flux $\tilde{J}$, then first the effect of the local heat flux density $\bar{q}_\varphi$ on the rate of condensation will be estimated.

The character of the effect of $\bar{q}_\varphi$ on the condensation rate can be best seen from the plot of Fig. 1 presenting $\bar{\alpha}/\bar{\alpha}_{Nu}$ vs the local mean vapour velocity or vs Fr at $Re_{film} = 40–100$, with $\bar{\alpha}_{Nu}$ being the heat transfer coefficient calculated from equation (1). The scatter of the data for identical $\bar{q}_\varphi$ is associated with the effect of Re_{film} on the condensation rate. The higher $\bar{q}_\varphi$ or $\tilde{J} = \bar{q}_\varphi / r\rho_v \bar{W}_v$, the smaller is $\bar{W}_v$ or Fr at which the interphase friction starts to influence the heat transfer rate.

The same character of the effect of $\bar{W}_v$ is observed at low pressures.

It is seen from Fig. 1 that numerically $\bar{\alpha}_\varphi \simeq \bar{\alpha}_{Nu}$ only at low values of $\bar{q}_\varphi$ and $\bar{W}_v$, i.e. there is an analogy with condensation on the outer surface of a horizontal tube. In this case the measured local values of the heat transfer coefficients coincide with those predicted by equation (11) (Fig. 2) when the mean values $\Delta\bar{T}_\varphi$ are substituted into it.

Figure 2 also shows the measured lines for ΔT_φ and for the corresponding values of q_φ.

In the region of small values of $\bar{W}_v$ ($\bar{W}_v < 1–5$ m s^{-1} for atmospheric pressure, $\bar{W}_v < 10–16$ m s^{-1} for $P = 0.01$ MPa) the local values of $\bar{\alpha}_\varphi$ decrease with an increasing $\bar{q}_\varphi$ or $\Delta\bar{T}_\varphi$ according to equation

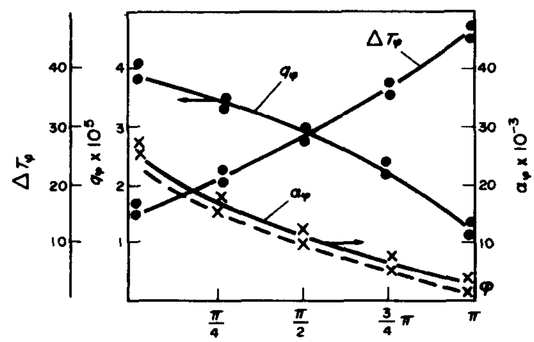

FIG. 2. Distribution of local thermal characteristics at $\bar{W}_v = 3.3$ m s^{-1}; $M_c = 0.59 \times 10^{-3}$ kg s^{-1}; $T_s = 100°C$.

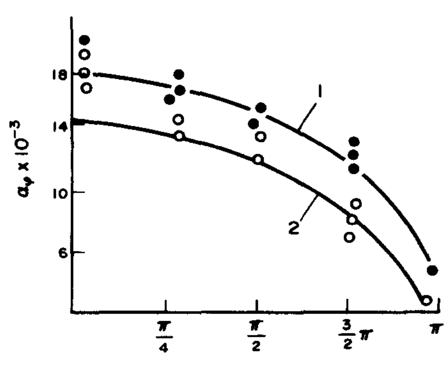

FIG. 3. Local heat transfer in condensation for $W_v < 5$ m s^{-1}: 1, $\bar{q}_\varphi = 10^5$ W m^{-2}; 2, 2.5×10^5. Line, calculation by equation (11).

(11) (Fig. 3) and the values of $\bar{\alpha}_\varphi$ decrease according to equation (1).

As shown earlier [12], an increase in the fraction of a tube occupied by a condensate rivulet because of its inflow onto the measuring section, which follows after

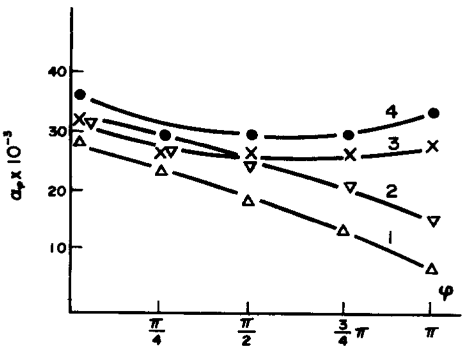

FIG. 4. Effect of the φ-average heat flux density $\bar{q}_\varphi$ over the test section on heat transfer laws at $\bar{W}_v \simeq 34$ m s^{-1}: 1, $\bar{q}_\varphi = 50$ kW m^{-2}, $Re_{film} = 39$; 2, 80 and 30; 3, 150 and 12; 4, 240 and 13.

the setting section, influences $\bar{\alpha}_\varphi$ only in the bottom portion of the tube. First, up to $Re_r = M_c/\Pi_r\mu_l = 1500$ an increase of M_c causes a decrease of α_φ, while at $Re_r > 1500$ an increase of α_φ is observed. Therefore, equation (1) can be used to calculate $\bar{\alpha}_\varphi$ in the case of the gravitational fall of condensate film up to $Re_r = 1500$.

The data on the effect of the mean vapour velocity $\bar{W}_v$ on α_φ at $\bar{q}_\varphi = 170$ kW m^{-2} and $M_c = 0.2 \times 10^{-3}$ kg s^{-1} for $T_s = 100$°C are given in Fig. 4. It follows from this figure that only at $\bar{W}_v \geqslant 40$ m s^{-1} virtually an annular flow mode of phases develops at the given values of $\bar{q}_\varphi$ and $M_c(Re_{film})$. When $\bar{W}_v < 10$ m s^{-1}, there is no effect of velocity on heat transfer as follows also from Fig. 1.

The quantity $\bar{q}_\varphi(\tilde{J})$ determines the character of the distribution of α_φ over φ. While for a slowly moving vapour α_φ decreases with an increasing $\bar{q}_\varphi$ at all the points φ, then under the conditions of vapour velocity effect on the condensate film flow the values of α_φ level off over φ on an increase in $\bar{q}_\varphi$ (Fig. 5).

An increase of M_c in the mode of moving vapour condensation leads, as shown in ref. [12], to a decrease of α_φ at all the points of the tube perimeter.

Using the data on the distribution of α_φ over φ for different M_c, $\bar{W}_v$ and $\bar{q}_\varphi$, the map of the flow modes of phases was constructed (Fig. 6) for incomplete laminar condensation inside of a horizontal tube in

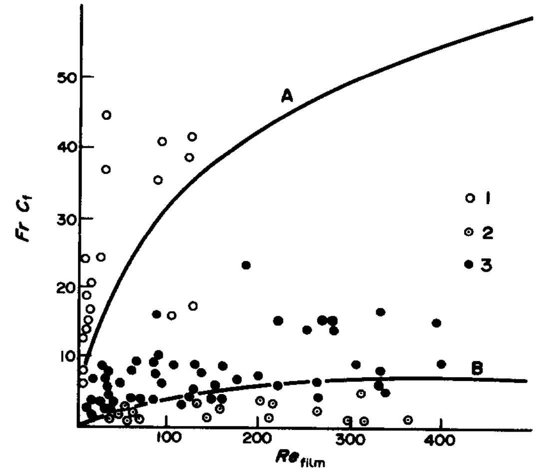

FIG. 6. The map of flow modes during condensation inside a horizontal tube: 1–3, data on $\bar{\alpha}_\varphi$ (1, annular mode; 2, gravitational mode; 3, asymmetric mode).

the coordinates $C_f\,Fr$ and Re_{film} which result from a theoretical determination of the parameter P_τ, equation (8).

To the region with the annular flow of phases, located above curve A, there correspond the experimental data on α_φ which are nearly identical at different points around the tube perimeter. In the region with gravitational fall of condensate film, which is located below curve B, the distribution of the quantity α_φ obeys the law given by equation (11). In the intermediate zone between lines A and B the distribution of α_φ over φ is asymmetric and differs from that predicted by equation (11).

The upper curve A is approximated by the relation

$$(C_f Fr)_c = 7.36 Re_{film}^{1/3} \tag{13}$$

while the lower curve B by the relation

$$(C_f\,Fr)_{lim} = 0.86 Re_{film}^{1/3}. \tag{14}$$

The estimation of the flow modes of phases by the parameters (3) and (4) does not agree with the data on $\alpha_\varphi = f(\varphi)$.

The investigations have also revealed the presence of a substantial entrainment of condensate with vapour. Figure 7 presents the data on the entrainment

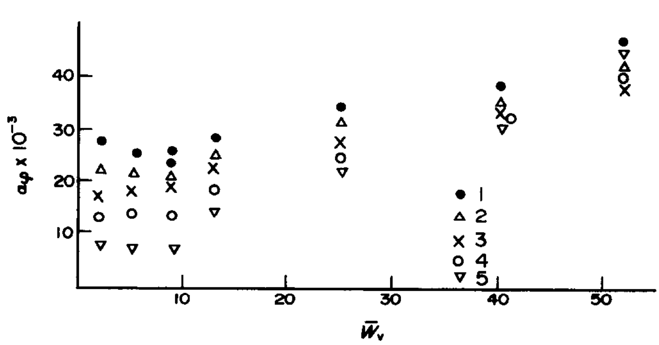

FIG. 5. Effect of the test section-average vapour velocity on heat transfer rate at $\bar{q}_\varphi = 140$–150 kW m^{-2} and $Re_{film} = 10$–15: 1, $\varphi = 0$; 2, $\pi/4$; 3, $\pi/2$; 4, $3\pi/4$; 5, π.

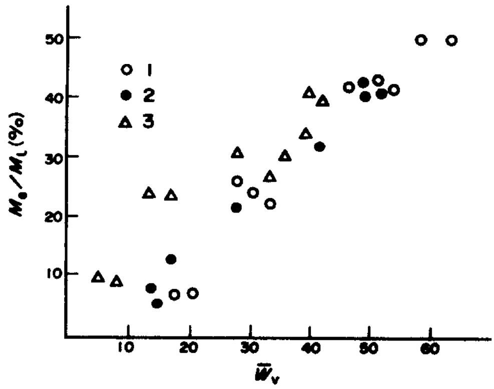

FIG. 7. Effect of vapour velocity on entrainment at $T_s = 100°C$: 1, $Re_{film} = 65$; 2, 100; 3, 400.

measured by mixture tapoff downstream of the measuring section after the branch pipe through which the condensate was discharged into a batch meter. As is seen from Fig. 7, at high vapour velocities the entrainment fraction exceeds 50% despite small Re_{film} numbers. For high vapour pressures and long tubes the fraction of entrainment may exceed 80% [16], however no investigations of entrainment in condensation was made.

By taking into account the fraction of entrainment when calculating the real number $Re^*_{film} = Re_{film} - Re_{en}$ in equation (10), the mean experimental numbers Nu_{film} will rather satisfactorily agree with those predicted (Fig. 8).

In the region with a comparable effect of gravity and interphase friction forces, i.e. when $0.86Re_{film}^{1/3} < C_f Fr < 7.36Re_{film}^{1/3}$, all the experimental data on the local values of $\bar{\alpha}_\varphi$ are correlated [12] by the relation

$$\overline{Nu}_{film_\varphi} = 1.5C_f^{0.5}\,Fr^{0.38}\,Re_{film}^{-0.26}. \quad (15)$$

The tube length-mean heat transfer coefficients $\bar{\alpha}_L$ or Nu_{film_L} will depend on the trend of variation of $\bar{q}_\varphi$ over the tube length and on the degree (complete or incomplete) of vapour condensation in the tube. When $x_2 = 0$ and $\bar{q}_\varphi = \text{const.}$, $\overline{Nu}_{film_L} = \overline{Nu}_{film_\varphi}$.

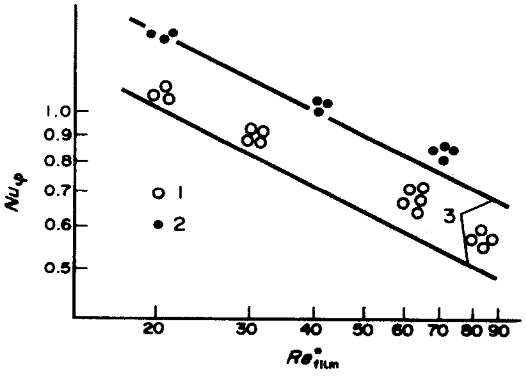

FIG. 8. Condensation in annular flow mode of phases: 1, $\bar{W}_v = 32$ m s^{-1}; 2, 50.

According to equation (15), $\bar{\alpha}_\varphi \sim \bar{q}_\varphi^{0.5}$ in agreement with the degree of the effect of q_L on $\bar{\alpha}_L$ observed in a number of experiments [5, 17, 18]. A great effect of $\bar{q}_L$ on $\bar{\alpha}_L$ is observed only when the turbulent flow of condensate film spreads over the entire perimeter and over the greater portion of the tube length and also when vapour condensation occurs in the flow of a moving turbulent homogeneous mixture—the case which takes place in the tube cross-section flooded with condensate or when there is an intensive entrainment of liquid by vapour. In the present experiments such condensation regimes were not observed.

5. CONCLUSION

Local heat transfer coefficients were measured for vapour condensation inside of a horizontal tube by using the method of a thick wall. In the region of $Re_{film} < 400$ and $Fr < 8400$ the laws persist that govern laminar film condensation. The map of the flow modes of phases is constructed and the relation is obtained for experimental data correlation which takes into account the effect of the transverse mass flux.

REFERENCES

1. V. G. Rifert, Vapour condensation inside horizontal tubes, *J. Engng Phys.* **44**(6), 1017–1029 (1983).
2. J. A. Myers and H. F. Rosson, Condensing coefficient inside a horizontal tube near atmospheric pressure, *Chem. Engng Prog. Symp. Series* **57**, 150–155 (1961).
3. V. G. Rifert and V. Yu. Zadiraka, Condensation of steam inside a smooth and a profiled horizontal tube, *Teploenergetika* No. 8, 77–80 (1978).
4. E. P. Ananiev, L. D. Boyko and G. N. Kruzhilin, Heat transfer in the presence of steam condensation in a horizontal tube. In *Int. Developments in Heat Transfer*, pt. 2, pp. 290–298. AMSE, New York (1961).
5. L. D. Boyko, Investigation of heat transfer with vapour condensation inside a tube. In *Heat Transfer in the Elements of Power Plants*, pp. 197–212. Izd. Nauka, Moscow (1966).
6. G. Breber, J. W. Palen and J. Taborek, Prediction of horizontal tube side condensations of pure components using flow regime criteria, *J. Heat Transfer* **102**(3), 93–100 (1980).
7. T. N. Tandon, H. K. Varma and C. P. Gupta, A new flow regime map for condensation inside horizontal tubes, *Trans. ASME, J. Heat Transfer* **104**(4), 763–768 (1982).
8. W. Nusselt, Die oberflächen Kondensation des Wasserdampfes, *VDJ* **60**(27), 541–546 (1916).
9. A. I. Leontiev, Engineering methods for calculation of friction and heat transfer on a permeable surface, *Teploenergetika* No. 9, 19–24 (1972).
10. A. A. Mikhalevich, *Mathematical Simulation of Condensation Heat and Mass Transfer*. Izd. Nauka i Tekhnika, Minsk (1982).
11. A. R. Dorokhov, Effect of condensing vapour velocity on heat transfer. In *The Problems of Hydrodynamics and Heat Transfer*, pp. 54–58. Izd. SO AN SSSR, Novosibirsk (1972).
12. V. G. Rifert, A. I. Sardak and A. N. Tobilevich, Flow modes of phases and heat transfer during vapour condensation inside horizontal tubes, *Izv. Akad. Nauk. SSSR, Energ. Transp.* No. 4, 101–109 (1985).

13. K. K. Shalbayev, V. P. Isachenko and F. Salomzoda, Heat transfer with film condensation on the outer surface of tubes in a descending vapour flow, *Teploenergetika* No. 12, 63–66 (1982).
14. A. E. Duckler, Fluid mechanics and heat transfer in vertical falling film systems, *Chem. Engng Prog. Symp. Series* **56**(30), 1–10 (1960).
15. V. K. Shchukin, A. A. Khalatov and V. A. Filin, Gradient method for investigation of heat transfer in channels of variable cross-section, *Izv. VUZov SSSR, Aviats. Tekh.* No. 4, 121–128 (1969).
16. G. F. Hewitt and N. S. Hall-Taylor, *Annular Two-phase Flow*. Chem. Engng Div., A.E.R.E., Harwell, Pergamon Press, Oxford (1974).
17. S. S. Kutateladze, Heat transfer in film vapour condensation inside horizontal tubes. In *Problems of Heat Transfer and Hydraulics of Two-phase Media*, pp. 138–156. Izd. GEI, Moscow (1961).
18. V. V. Konsetov, Experimental investigation of heat transfer in steam condensation inside horizontal and slightly inclined tubes, *Teploenergetika* No. 12, 67–71 (1960).

MODES DE TRANSFERT THERMIQUE ET D'ECOULEMENT DES PHASES DANS LA CONDENSATION DE VAPEUR EN FILM LAMINAIRE DANS UN TUBE HORIZONTAL

Résumé—En utilisant la méthode du gradient on détermine le coefficient de transfert thermique local pour la condensation de vapeur en film laminaire dans un tube horizontal. On étudie l'effet de la vitesse de la vapeur, de l'entrée de condensat et du flux de masse transversal sur l'intensité du mécanisme. L'existence d'une dissymétrie prononcée dans les coefficients locaux de transfert de chaleur autour du tube est notée, ainsi que sa diminution avec un accroissement de la vitesse de vapeur et du flux thermique local. On identifie les modes d'ecoulement des phases dans le cas de la condensation incomplète de vapeur ; on note la nécessité de prendre en compte l'épaisseur du film de condensat et l'effet du flux de masse transversal quand on construit la carte des modes. On suggère une formule pour calculer les coefficients de transfert thermique locaux et globaux pour la condensation en film laminaire.

WÄRMEÜBERGANG UND STRÖMUNGSFORMEN BEI DER LAMINAREN FILMKONDENSATION IN EINEM WAAGERECHTEN ROHR

Zusammenfassung—Die örtlichen Wärmeübergangs-Koeffizienten bei der laminaren Filmkondensation in einem waagerechten Rohr wurden mit Hilfe der Gradienten-Methode bestimmt. Dabei wurden die Enflüsse von Dampfgeschwindigkeit, Kondensatströmung und Kondensat-Massenstromdichte untersucht. Eine ausgeprägte Asymmetrie der örtlichen Wärmeübergangs-Koeffizienten am Rohrumfang wird festgestellt, was jedoch bei steigender Dampfgeschwindigkeit und Wärmestromdichte geringer wird. Die unterschiedlichen Zweiphasen-Strömungsformen bei unvollständiger Kondensation werden registriert. Es wird angemerkt, daß Kondensat-Filmdicke und -Massenstromdichte bei der Erstellung von Strömungsbilderkarten berücksichtigt werden müssen. Abschließend wird eine Korrelation zur Berechnung örtlicher und mittlerer Wärmeübergang-Koeffizienten bei der laminaren Filmkondensation vorgeschlagen.

ТЕПЛООБМЕН И РЕЖИМЫ ТЕЧЕНИЯ ФАЗ ПРИ ЛАМИНАРНОЙ ПЛЕНОЧНОЙ КОНДЕНСАЦИИ ПАРА ВНУТРИ ГОРИЗОНТАЛЬНОЙ ТРУБЫ

Аннотация—Используя градиентный метод исследования теплообмена, определены локальные коэффициенты теплоотдачи при ламинарной пленочной конденсации пара внутри горизонтальной трубы. Изучено влияние на интенсивность процесса скорости пара, натекания конденсата, поперечного потока массы. Отмечено наличие существенной асимметрии локальных коэффициентов теплоотдачи по периметру трубы, уменьшению которой способствует увеличение скорости пара и местного теплового потока. Идентифицированы режимы течения фаз при неполной конденсации пара в трубе; при построении карты режимов отмечена актуальность учета толщины пленки конденсата и влияния поперечного потока массы. Дано корреляционное уравнение для расчета локальных и среднеинтегральных коэффициентов теплоотдачи при ламинарной пленочной конденсации.