CONDENSATION INSIDE SMOOTH HORIZONTAL TUBES
Part 2. Improvement of Heat Exchange Prediction

by

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Original scientific paper
https://doi.org/10.2298/TSCI140815045R

In this study, a theoretical model of film condensation inside horizontal tubes with more precise definition of friction coefficient on interphase is presented to calculate the heat transfer coefficient under two-phase annular and intermediate flow conditions. This more precise definition contains experimental substantiation of correction for calculation of pressure losses by friction and correction that takes into account surface suction on the interphase. Comparison of this model with the experimental data of various authors demonstrates that the results are in satisfactory and close agreement.

Key words: *heat transfer, film condensation, smooth horizontal tube, inside*

Introduction

Study of heat exchange under condensation inside horizontal tubes is going on for more than 65 years. One of the first works on the item has been published in 1947 [1]. Since then various authors have proposed more then 60 methods and correlations [2-7]. At present, precise estimation of heat transfer is possible only in absence of vapor velocity impact on the process. In this case, heat transfer is calculated by Nusselt's correlation [8] for condensation outside horizontal tube.

For theoretical solution of the problem of heat exchange in annular and intermediate modes of the phases it is necessary to know pressure loss by friction, $(\Delta P/\Delta x)_f$, friction factor, C_f, and void fraction, ε. Existing methods of these parameters prediction give high (above 50%) discrepancy between them [3, 4, 7, 9, 10].

The majority of the empirical correlations for heat exchange calculation in annular and intermediate modes are:

$$\mathrm{Nu} = c\,\mathrm{Re}^n \mathrm{Pr}_l^m\, \Phi\left(x\right) \tag{1}$$

where constant c, the exponents in Re, Pr_l, and complex $\Phi(x)$ are significantly different. For example, exponent n could vary from 0.33 [11] to 0.997 [12]. Reynolds number and $\Phi(x)$ included in the most known empirical correlations not always correctly describe film condensation inside horizontal tube.

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The correlations by Shah [13, 14] and Thome *et al.* [15] have the best convergence with the experiments of various authors. In these correlations, all included complexes are selected by intuition, without any theoretical or experimental substantiation [7].

In the present work, the method of heat exchange prediction under condensation inside horizontal tubes based on the film condensation theory developed by Dukler [16], Bae *et al.* [17, 18], and Travis *et al.* [19] is analyzed. Substantiated are the main complexes that define condensation process and the correlation for prediction of friction factor on the interphase.

Substantiation of calculating correlations

For annular phase flow mode the calculations, which results are represented in dimensionless terms:

$$\mathrm{Nu} = f(\beta, \mathrm{Re}_l, \mathrm{Pr}_l) \tag{2}$$

where $\mathrm{Nu} = [\alpha(\nu_l^2/g)^{1/3}]/\lambda_l$, $\beta = [\tau_f^3/(\rho_l g^2 \mu_l^2)]^{1/3}$, and $\mathrm{Re}_l = 4\mathrm{Re}_f$, are given in [16-19].

These correlations are plotted in [18] for Pr_l number from 1 to 5. In figs. 1 and 2 such diagrams are plotted for $\mathrm{Pr}_l = 1$ and $\mathrm{Pr}_l = 3$, respectively. The analysis of correlation makes possible to note the following features of heat exchange that could be laid down into improved calculation method. So, if to address to the left part of the diagrams in figs. 1 and 2, then it is seen that in the region of Re_l low values heat transfer decreases with Re_l increasing. Here degree of β and Re_l impact corresponds to Nusselt's theory of laminar film condensation [8]:

$$\mathrm{Nu} = 0.5\left(C_f \mathrm{Fr}_l\right)^{0.5} \mathrm{Re}_f^{-0.5} \tag{3}$$

As Re_l increases the heat transfer (Nu_f) does not decrease any more and a region of weak dependence of Nu_f upon Re_l sets in. At high Re_l numbers (Re_l > 3000-10000, depending upon Pr_l) Nu_f growing with Re_l is observed. In the region close to laminar flow of condensate film (Re_l < 100-200) effect of Pr_l is negligible and in some regimes (Re_l < 100 and β > 50-100) is absent in general. It is true for laminar flow. At turbulent condensate flow in accordance with the theory the heat transfer intensifies as Pr_l grows.

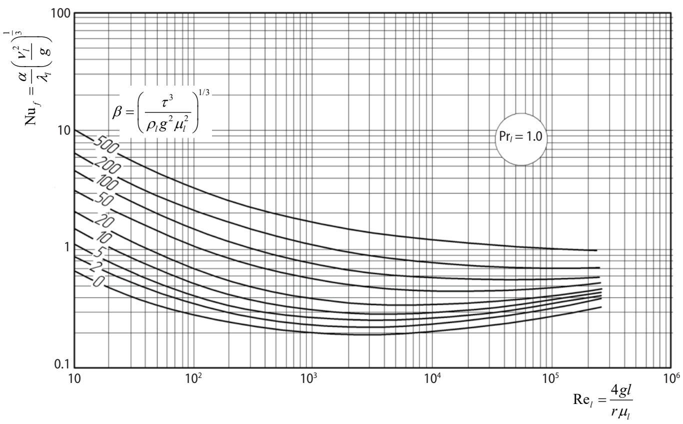

Figure 1. Dimensionless local heat transfer coefficients (Pr_l = 1)

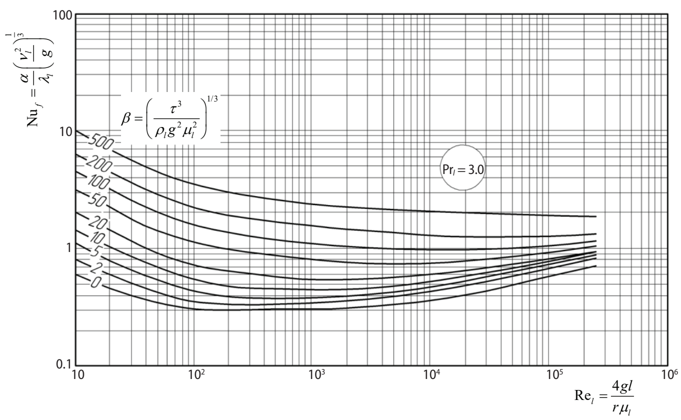

Figure 2. Dimensionless local heat transfer coefficients (Pr_l = 3)

Analysis of correlation (2) also reveals that the effect of the forces of interphase friction (parameter β) depends upon Re_f and Pr_l. The higher Re_f (the thicker condensate film and the greater gravity), the weaker impact of β under the same Pr_l. In this case impact of β under the same Re_f and Pr_l becomes stronger as β grows, since ratio of friction to gravity grows.

These theoretical calculations strictly (correctly) reflect process of condensation of flowing vapor inside vertical tubes and channels and could be used for annular phase flow in horizontal tubes.

Bae *et al.* [17, 18] and Traviss *et al.* [19] compare their experimental data on condensation of R12 and R22 inside horizontal tube of d = 8 and 12.5 mm and length up to 6 m in a wide range of G (from 160 to 1500 kg/m^{2}s) and q (from 7.8 to 85 kW/m^2) with theoretical calculations (in fact, with diagrams like figs. 1 and 2). In [17, 18] a good agreement of the experiments with the calculations is shown. In [19] deviation of the experimental data for high G (above 400 kg/m^{2}s) at x > 0.5-0.6 to the values exceeding the results of calculation was noted.

It is necessary to mention that accuracy of the calculations by this theory significantly differs from the design method for friction pressure drop or friction coefficient. None of the existing works has substantiated correlation for their calculation. Besides, as mentioned in [7], in all previous works φ-wise local heat transfer coefficient was not measured that makes impossible to gain correct enough idea on the effect of friction on the interphase and of gravity.

Experimental set-up

In fig. 3 the diagram of experimental set-up is shown. The experimental set-up included steam generator – 1, steam super heater – 2, the 1st presection – 3, the 1st experimental section – 4, the 2nd presection – 5, the 2nd experimental section – 6, terminal condenser – 7, rotameter – 8 for measurement of condensate flow rate, rotameters – 9, 10, and 11 for measurement of cooling water flow rate in condenser – 7, presections – 3 and 5 and experimental sections – 4 and 6, respectively, rotameter – 12 for measurement of cooling water flow rate supplied into the experimental sections. Presections – 3 and 5 gave the possibility to create corresponding modes of phase flow in the experimental sections – 4 and 6, respectively. All sections were located on one longitudinal axis. Inside diameter of the tubes in all sections was 17 mm, the length of

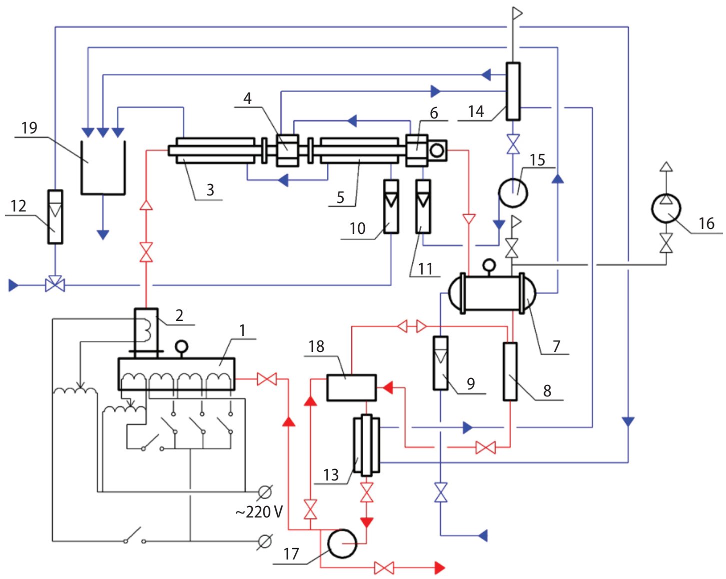

Figure 3. Diagram of experimental set-up;
1 – electric steam generator, 2 – separator super heater, 3 – 1st presection, 4 – 1st experimental section, 5 – 2nd presection, 6 – 2nd experimental section, 7 – terminal condenser, 8 – volumetric condensate gauge, 9, 10, 11, 12 – rotameters, 13 – condensate overcooler, 14 – mixer, 15 – circulation pump, 16 – vacuum pump, 17 – condensate pump, 18 – receiver, 19 – drain collector

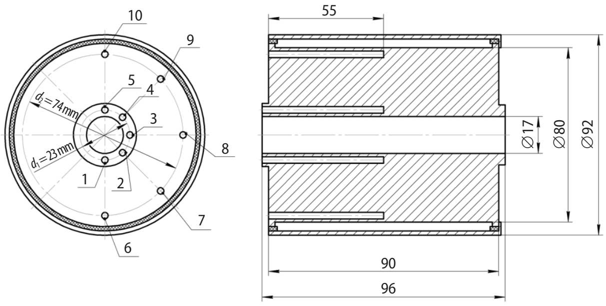

Figure 4. Layout of brass working section;
1-5 – channels for locating thermocouples at d_1 = 23 mm, 6-10 – channels for locating thermocouples at d_2 = 74 mm

both presections was 0.8 m and of experimental sections was 110 mm. All tubes were made of brass. The experimental sections (fig. 4) had 80 mm outside diameter and inside their walls at the diameters d_1 = 23 and d_2 = 74 mm in the center of each of the sections five chromel-copel thermocouples were laid down at the points with φ = 0°, 45°, 90°, 135°, and 180°. The experimental values of local heat flux q_φ and heat transfer coefficient α_φ were determined by using the temperatures measured in these points and the following correlations:

$$q_l = \frac{2\lambda_b \pi \left(t_i - t_j\right)}{\ln\left(d_2/d_1\right)}, \quad t_w = t_i + \frac{q_l}{\pi}\frac{1}{2\lambda_b}\ln\frac{d_1}{d}, \quad q_\varphi = \frac{q_l}{\pi d}, \quad \alpha_\varphi = \frac{q_\varphi}{t_s - t_w}$$

where q_l [Wm^{-2}] is a linear density of heat flux, λ_b [Wm^{-1}K^{-1}] – the thermal conductivity coefficient of brass experimental sections, t_w [°C] – the temperature of working section inside wall, and i, j are the numbers of the thermocouples at diameters d_1 and d_2, respectively, fig. 4.

Saturation temperature, t_s, was measured by the thermocouple located at the inlet into the 1st presection and evaluated due to the thermocouple installed directly after the 2nd experimental section. The maximal obtained relative uncertainty of the heat transfer coefficient was equal to 5.35%. Unbalance of the heat spent for heating water, which cooled the all sections and condenser, and determined by quantity of condensate created in this case did not exceed 2%.

The results of experimental studies

To prove correctness of the applied method of local α_φ measurement the modes corresponding to the maximal accuracy of theoretical local α_φ were investigated. There were two modes of condensation inside horizontal tube. The first one was without effect of vapor and condensate stream velocities, when $J_g \ll 1.0$ and $X_{tt} < 1$. The second mode was under prevailing impact of vapor velocity and annular flow of the phases, when $\mathrm{Re}_f < 100$ and laminar film of condensate took place ($J_g \gg 2.0$ and $X_{tt} < 1$).

In fig. 5 the changes in local heat transfer coefficients versus φ at w_v = 6 m/s, x = 0.5 and 1.0, and two values of heat flux $\overline{q}_\varphi$ =139 and 122 kW/m^2 averaged by φ are plotted. At these parameters J_g = 0.36, X_{tt} = 0 (at x = 1.0) and X_{tt} = 0.04 (at x = 0.5). Analysis of flow pattern maps in [20] shows that due to the map of Taitel and Dukler [21] one can find the maximal values of J_g corresponding to strictly stratified flow pattern. This value is J_g = 1.1 for X_{tt} = 1.0 and J_g = 0.7 for X_{tt} = 0.04. Dotted lines in fig. 5 are the local coefficients of heat transfer by Nusselt formula [8]:

$$\alpha_\varphi = \lambda_l \left[\frac{A\int\limits_0^\varphi \left(\sin\varphi\right)^{1/3} \mathrm{d}\varphi}{\left(\sin\varphi\right)^{4/3}} \right]^{-0.25} \tag{4}$$

where $A = 2\nu_l\lambda_l d\Delta T/(\rho_l g r)$.

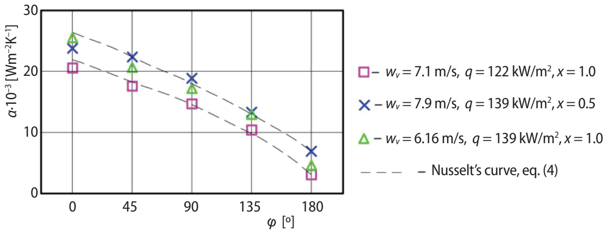

Figure 5. Comparison of experimental data with Nusselt formula

As seen from fig. 5, the experimental α_φ perfectly agree with those calculated by eq. (4).

Another verification of the accuracy of α_φ measurement and of the theory of flowing vapor film condensation has been performed for the second condensation mode. There are doubts about accuracy of the methods of velocity profile calculation, estimation of turbulent Pr_t, and influence of condensate entrainment. As well, there is still a problem to estimate friction coefficient, C_f, included in theoretical correlation (3) for prediction of heat transfer on the interphase. The values of C_f and $(\Delta P/\Delta z)_f$ depend upon two-phase flow parameters, the main of which are vapor content x and density ρ_v, as well as liquid density ρ_l. As follows from the theory of annular phase flow in book [22] and numerous prediction correlations from [3, 9], friction coefficients in one-phase and two-phase flows and their $(\Delta P/\Delta z)_f$ are equal, when Locart-Martinelli parameter X_{tt} is close to zero.

For water steam with pressure below 10^5 Pa, when $\rho_v/\rho_l \ll 1$ at $x > 0.9$ according to any available correlations for Φ_v^2 required to calculate $(\Delta P/\Delta z)_f$ and C_f, $\Phi_v^2 < 1.1$-1.2 is obtained. In tab. 1 the experimental results on condensation of water steam are represented, while in tab. 2 the calculated complexes C_{fo}, Fr_l, β_o, Re_f, Nu_o, Φ_v^2, β_v, and Nu_v are given. Here C_{fo} is a coefficient for one-phase flow:

$$C_{fo} = \frac{0.079}{Re_v^{0.25}} \quad \text{at} \quad Re_v < 10^5 \tag{5}$$

$$C_{fo} = \frac{0.046}{Re_v^{0.2}} \quad \text{at} \quad Re_v > 10^5 \tag{6}$$

The Φ_v^2 is a multiplier that expresses impact of two-phase flow upon $(\Delta P/\Delta z)_f$, C_f, and Nu_v. Nu_o, and Nu_v values were predicted by using MathCad package with spline interpolation of the diagrams from [18], figs. 1 and 2, by experimental values of Re_f, Pr_l, β_o, and β_v. Calculation of Φ_v^2 has been performed due to five formulas from Wang *et al.*, [23], Soliman *et al.*, [24], Koyama *et al.*, [25], Hasan *et al.,* [26], and Autee *et al.,* [27]. The all formulas give very close results (they differ by below 20%) at $x \geq 0.8$. At $x < 0.8$ the formulas from [23] and [24] do not take into account the effect of G and x and give overrated values of β_v. In all calculations we used formula by Hasan *et al.* [26] that includes effect of superficial forces (Bo criterion) on β_v:

Table 1. Experimental data at water steam condensation

Parameters	1	2	3	4	5	6	7	8	9	10	11
t_s, [°C]	105	106	104	101	103	101	103	101	103	105	102
G, [kgm^{-2}s^{-1}]	36	27.3	30.7	16	20	16	24.6	13.4	21	35	195
x	0.975	0.96	0.76	0.74	0.9	0.48	0.68	0.82	0.66	0.96	0.73
$\bar{q}_\varphi \cdot 10^{-3}$ [Wm^{-2}]	180	190	200	180	240	300	300	80	75	360	150
$Re_f \cdot 10^{-3}$	86	109	442	250	120	500	470	144	470	83	162
$\bar{\alpha}_\varphi \cdot 10^{-3}$ [Wm^{-2}K^{-1}]	42	33	31	25	31	30	26	24	22.3	54	25.5
Nu_{exp}	1.26	0.99	0.93	0.75	0.93	0.9	0.82	0.72	0.67	1.6	0.765

Table 2. Comparison of experimental and predicted data

Work Fluid	t_s	G/x	$(\bar{q}\cdot 10^{-3})/$ $(\bar{\alpha}\cdot 10^{-3})$	$\overline{\mathrm{Nu}}_{\mathrm{exp}}$	$(\mathrm{Fr}_l\cdot 10^{-3})/$ $(C_{fo}\cdot 10^{-3})$	$\mathrm{Re_f}$	Nu_k	$(\beta_0)/$ (Nu_{do})	$(\Phi_v^2)/$ (Nu_v)	$(\Phi_q)/$ (Nu_{vq})
Present R718	100	31.2 0.98	180 42	**1.26**	9.2 5.2	*85*	1.05	24 0.8	1.12 0.82	1.6 **1.12**
Present R718	106	23.4 0.98	180 33	**0.99**	5.4 5.6	100	0.82	15.5 0.62	1.2 0.68	1.84 **0.9**
Present R22	40	284 0.99	36 3.9	**0.59**	9.9 3.25	*566*	–	16 0.45	1.14 0.48	1.3 **0.58**
[37] R22	40	600 0.84	40 6.36	**0.96**	29.5 3.35	5530	–	49 0.61	2.7 0.98	1.19 **1.1**
[37] R236ea	40	600 0.768	28.3 8.44	**1.98**	41.2 3.3	3560	–	68 1.0	2.7 1.8	1.22 **2.05**
[37] R134a	40	750 0.8	51 7.9	**1.34**	50 3.15	7430	–	78 0.9	2.1 1.22	1.22 **1.36**
[38] Propane	40	300 0.88	7.7 5.54	**0.94**	32 3.48	4200	–	55 0.7	1.9 0.92	1.0 **0.92**
[39] R22	40	300 0.83	49.6 3.8	**0.57**	6.56 3.86	4250	–	13 0.4	2.07 0.5	1.44 **0.58**
[17] R12	34	430 0.92	33.4 4.45	**0.85**	20.5 3.0	2670	–	32 0.57	2.09 0.7	1.34 **0.86**

$$\Phi_v^2 = 1 + CX_{tt}^n + X_{tt}^2 \tag{7}$$

where

$$C = 21\left[1-\mathrm{e}^{(1-0.28\mathrm{Bo}^{0.5})}\right](1-0.9\mathrm{e}^{-0.2\mathrm{Fr}_l^{1.5}}), \quad n = 1-0.7\mathrm{e}^{-0.08\mathrm{Fr}_l}, \quad \mathrm{Fr}_l = \frac{Gx}{\left[\mathrm{g}dG(\rho_l-\rho_v)\right]^{0.5}}$$

In fig. 6 the experimental data on local heat transfer coefficients, α_φ, at water steam condensation and high steam content ($x \geq 0.9$), when $J_g > 1.2$ and $X_{tt} < 0.1$, *i. e.*, when annular flow pattern should take place, are shown. In this case, $\mathrm{Re}_f < 100$ that corresponds to laminar flow of condensate film. The experimental values of α_φ are obtained in the center of the exper-

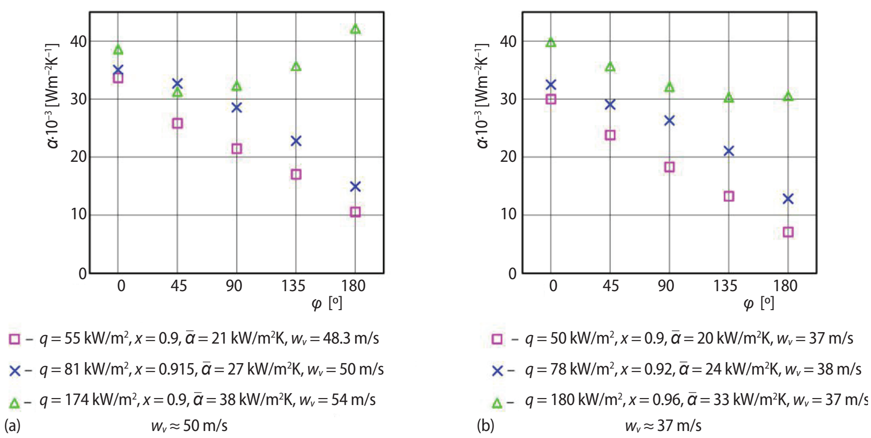

Figure 6. Effect of heat flux on heat transfer under water steam condensation

imental sections (at 40-mm length). Three features of α_φ change are clearly seen. The first of them represents decrease in α_φ, as φ increases that means asymmetric mode. The second feature is an increase in α_φ as steam velocity increases. The third one is the most interesting feature of increasing α_φ under equal (close values) steam velocities, when heat flux is growing.

Effect of heat flux on hydraulic resistance and heat transfer in two-phase flows under phase transformations is theoretically substantiated in [28, 29] and is explained by mass suction in boundary layer under condensation. In [28] it is shown that, if suction parameter $j = q/(rG) > 10^{-4}$, hydraulic resistance C_f on interphase increases as compared to one-phase flow resistance C_{fo} and is described by equation:

$$\frac{C_f}{C_{fo}} = 1 + 17.5\,\mathrm{Re}_v^{0.25} j \tag{8}$$

In [29] correlation for prediction of C_f yields:

$$\frac{C_f}{C_{fo}} = \frac{\left(1 - 0.25b\right)^2}{\left(1 + 0.25b\right)^2} \tag{9}$$

where $b = -2q/(GrC_{fo})$. Here, penetration parameter is limited, when suction effects on C_f/C_{fo} at $b < -4$.

Mickley *et al.* in [30] much earlier than the authors of works [28, 29] proposed the following correlation to take into account the effect of j upon C_f/C_{fo}:

$$\frac{C_f}{C_{fo}} = \frac{b}{\dfrac{1}{\mathrm{e}^b} - 1} \tag{10}$$

This correlation is cited in [31] and used for prediction of friction resistance in [32]. The calculations by eqs. (8)-(10) differ within ±10%. In further analysis of the experimental data we shall use correction Φ_q that takes into account entrainment on the interphase:

$$\Phi_q = \frac{C_f}{C_{fo}} \tag{11}$$

where relation C_f/C_{fo} is calculated by correlation with limitation of entrainment effect by boundary values of $b < -4$.

Increasing friction coefficient, C_f, and pressure losses $(\Delta P/\Delta z)_f$ lead to increase in parameter β. Respectively, local and average heat transfer coefficients are increased. The facts of such heat flux impact under the conditions of *convective* heat exchange are noted in experiments [17, 18], investigations [33] and for the first time were found on the basis of measured local α_φ performed by Rifert [34]. At the same time, in some works, *e. g.*, [35, 36], existence of the mode independent upon $\Delta T(q)$ is supposed.

In tab. 2 the experimental data by various authors on, α_φ, obtained for the case, when correction $\Phi^2_v < 1.2$, while correction $\Phi_q > 1.6$, are presented. Taking into account these two corrections by $\beta_{vq} = \beta_0\Phi^2_v\Phi_q$ makes possible to achieve perfect convergence of calculated and experimental data.

Table 2 contains the values of Nu_k predicted by correlation eq. (3) for laminar film condensation of flowing vapor, in which C_f is determined taking into account corrections Φ_v^2

and Φ_q. Good convergence of predicted and experimental data proves correctness of our experiments and the theory.

Validity of laminar film condensation laws for flowing vapor under annular flow of condensate film could be also checked by law of $\alpha = \lambda/\delta$, where δ is a thickness of condensate film. Film thickness at $Re_l < 1100$ could be determined by theoretical correlation from [19]:

$$\delta^+ = \frac{\delta}{\nu_l}\left(\frac{\tau_f}{\rho_l}\right)^{0.5} = 0,707\,\mathrm{Re}_l^{0.5} \tag{12}$$

On defining τ_f with taken into consideration the effect of Φ_v^2 and Φ_q the values of δ and α_φ were calculated. So defined heat-transfer coefficients perfectly agree with the experimental results that once again proves the accuracy of the experiments and validity of the methods of heat-exchange calculations.

In fig. 7 the experimental data on local $\alpha_\varphi = f(q)$ under condensation of R22 ($t_s = 40$°C, $G = 284$ kg/m^{2}s, $x = 0.99$) are plotted. Under such parameters $J_g > 2.4$ and $X_{tt} \approx 0$ and according to the model in [40] an annular flow of the phases should take place. However, in fig. 7 descend in local α_φ under increasing φ is seen. Heat transfer coefficients decrease as q increases thus following the theory of laminar film condensation. As well as under water steam condensation it is seen from calculation for R22 that the calculated and experimental α_φ completely converge, when the corrections Φ_v^2 for two-phase flow and Φ_q for impact of q are taken into account.

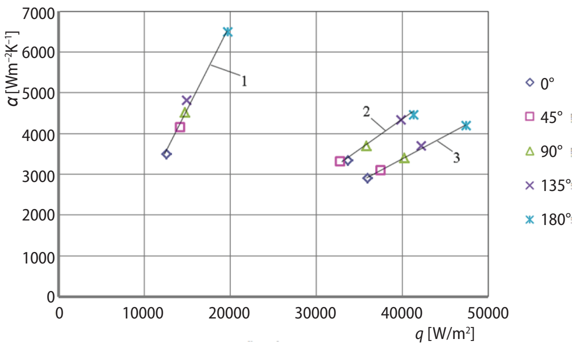

Figure 7. Effect of heat flux on heat transfer under condensation of R22;
1 – $\bar{q}_\varphi = 15 \cdot 10^3$ W/m^2, 2 – $\bar{q}_\varphi = 36 \cdot 10^3$ W/m^2, and 3 – $\bar{q}_\varphi = 42 \cdot 10^3$ W/m^2

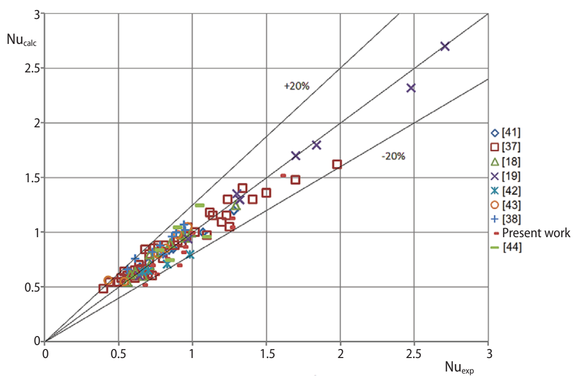

Figure 8. Comparison of experimental and predicted data

In tab. 2 the calculations of the experimental data from [37] for R22, R134, and R236, from [39] for R22, from [38] for propane, and from [17] for R12 are represented. For different G and q a good convergence of prediction with the experimental data is clearly seen. The effect of heat flux on heat transfer could be neglected, when $\Phi_q \ll \Phi_v^2$. If $\Phi_v^2 \ll \Phi_q$ then it is possible to neglect the impact of Φ_v^2 on heat transfer.

In fig. 8 the experimental data from tabs. 1 and 3 are compared for the modes of phase flow close to annular and asymmetric (intermediate) pattern with the prediction in accordance with the previously considered methode.

Such modes depending upon physical properties of condensing substance involve wide range of J_g and X_{tt} change and are also affected by tube diameter and length. For example, in [45], where local by φ thickness of liquid film in two-

Table 3. The experimental data of various authors

Work	[37]	[41]	[18]	[19]	[42]	[43]	[38]	[44]
Fluid	R22, R134a, R125, R236ea, R32, R410a	R404a	R22	R12, R22	R134a	R22	Propane, isobutene	R123
t_s, [°C]	40	40	40	25	25	60	40	69
G, [kgm^{-2}s^{-1}]	300, 400, 600, 750	400, 500, 600	430, 635	800, 1000	300	300, 600	300	300
x [–]	0.86-0.4	0.875-0.6	0.94-0.46	0.925-0.15	0.82-0.5	0.84-0.37	0.88-0.31	0.99-0.5

phase flow was measured, asymmetry increased under constant vapor velocity, J_g, with increasing ratio l/d. More detailed analysis of phase flow patterns, predicted parameters of condensate stream and heat-transfer calculation methods under the patterns close to stratified one will be performed in the further work. For comparison the experiments from the previously mentioned works, mainly for $x > 0.5$ are selected. As follows from fig. 8, prediction of Nu_{vq}, made in accordance with the proposed improved method, agrees with the experimental data within ±20%.

Conclusions

The improved model of film condensation inside horizontal tubes for prediction of heat transfer with application of the results of numerical solutions of Bae *et al*. is proposed. In this model more precise definition of friction coefficient on interphase as the main parameter crucial for condensation is given. This more precise definition contains experimental substantiation of Φ^2_v – prediction for calculation of pressure losses by friction and correction Φ_q that takes into account surface suction at the interphase.

The unique measurements of heat fluxes and heat-transfer coefficients local by circumference have been carried out during such condensation modes, when only Φ^2_v or only Φ_q or equally Φ^2_v and Φ_q affected.

Heat exchange predicted by the proposed method was compared with the experimental data of various authors for twelve substances in annular and intermediate modes. Good agreement of the experiments with calculations (divergence within 20%) proves correctness of the theoretical models both for laminar flow of condensate film (Nusselt's theory) and turbulent flow (models by Bae *et al.*).

Nomenclature

Bo – Bond number, [$= gd^2(\rho_l - \rho_v)/\sigma$], [–]
C_f – friction coefficient, [–]
d – inner diameter of tube, [m]
Fr_l – liquid Froude number, {$= [\rho_v(\rho_l - \rho_v)w_v^2]/[\rho_l^2(\nu_l g)^{2/3}]$}, [–]
G – mass velocity, [kgm^{-2}s^{-1}]
g – gravitational acceleration, [ms^{-2}]
J_g – dimensionless vapor velocity, {$= xG/[gd\rho_v(\rho_l - \rho_v)]^{0.5}$}, [–]
l – length of the tube, [m]
Nu – Nusselt number, [–]
Pr – Prandtl number, [–]
q – heat flux, [Wm^{-2}]
r – heat of vaporization, [Jkg^{-1}]
Re_f – film Reynolds number, [$= ql/(r\mu_l)$], [–]
Re_l – liquid Reynolds number, [$= G(1 - x)d/\mu_l$], [–]
Re_v – vapor Reynolds number, ($= Gxd/\mu_v$), [–]
ΔT – temperature difference ($= t_s - t_w$), [K]
t – temperature, [°C]
w – velocity, [ms^{-1}]
x – vapor quality, [–]
X_{tt} – Martinelli parameter, {$= (\mu_l/\mu_v)^{0.1}(\rho_v/\rho_l)^{0.5}[(1 - x)/x]^{0.9}$}, [–]

Greek symbols

α – heat transfer coefficient, [Wm^{-2}K^{-1}]
β – parameter related to friction stress in the interphase, [–]
λ – thermal conductivity, [Wm^{-1}K^{-1}]
μ – dynamic viscosity, [Pa·s]
ν – kinematic viscosity, [m^{2}s^{-1}]
ρ – density, [kgm^{-3}]

σ – surface tension, [Nm^{-1}]
τ – shear stress, [Pa]
φ – angular co-ordinate, [°]

Subscripts and superscripts

f – frictional factor
l – liquid
v – vapor/gas
exp – experimental
calc – calculated
$+$ – non-dimensional symbol

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Paper submitted: August 15, 2014
Paper revised: January 31, 2016
Paper accepted: March 2, 2016