CONDENSATION INSIDE SMOOTH HORIZONTAL TUBES Part 1. Survey of the Methods of Heat-Exchange Prediction

by

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Review paper
DOI: 10.2298/TSCI140522036R

Survey of the works on condensation inside smooth horizontal tubes published from 1955 to 2013 has been performed. Theoretical and experimental investigations, as well as more than twenty five methods and correlations for heat transfer prediction are considered. It is shown that accuracy of this prediction depends on the accuracy of volumetric vapor content and pressure drop at the interphase. The necessity of new studies concerning both local heat transfer coefficients and film condensation along tube perimeter and length under annular, stratified, and intermediate regimes of phase flow was substantiated. These characteristics being defined will allow determining more precisely the boundaries of the flow regimes and the methods of heat transfer prediction.

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

Introduction

Interest in study of heat exchange in condensing inside horizontal tubes is permanently actual in connection with the demand to qualify the design methods applicable for various heat exchangers (condensers of cooling equipment and air conditioning systems, horizontal tubular evaporators of water-desalinating thermal units, heaters of power systems, heat pipes).

Theoretical study of condensation inside smooth horizontal tube is mainly conducted for only two flow regimes: (1) annular, when flow of condensate primarily depends upon vapor velocity and (2) gravitational, when due to gravity condensate flows top-down along tube perimeter. In this case, heat transfer calculation in the section from the tube upper generatrix down to the radius that limits condensate stream in the tube lower part is performed by the Nusselt formula [1]. There are several solutions to predict parameters of the stream and heat transfer in it. To calculate heat transfer in an annular film depending upon vapor velocity there is theoretical Nusselt solution for laminar flow of condensate film [1], as well as there are several solutions for turbulent film flow [2-6].

In overwhelming majority of the proposed design correlations the structure of formulas has numerous versions of the complexes that include mass velocity G and vapor content x. The experimental investigations of vapor condensation inside horizontal tubes are mainly aimed to determine the influence of G and x upon mean heat-transfer coefficient (instead of local one

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as many authors state) over tube perimeter and at a definite length (minimally 0.3 m, but more often from 0.5 m to 1.0 m). Local heat fluxes q and heat-transfer coefficient were measured only in the works of Rifert [7].

Presentation of the experimental data as the functions of heat transfer *vs.* local G and x does not make it possible to obtain clearer comprehension of film condensation behavior, regimes of phase flow, impact of heat flux $q(\Delta T)$, vapor velocity, and condensate loss upon heat exchange. Probably, the reason is that often the experimental data on heat exchange substantially, differ from the calculated by various formulas.

Cavallini *et al.* [8], Garcia-Valladares [9], Shah [10], Kandlikar *et al.* [11], Dalkilic and Wongwises [12], Wei *et al.* [13], and Santa [14] have represented the surveys of the researches conducted by various authors. In some surveys design correlations are submitted, besides in several works different models are compared. More detailed analysis of the researches performed until 2014 including primary data on heat exchange is made as well.

Theoretical study

According to the well-known physical model of two-phase flow with condensation inside horizontal tubes there are an annular regime, where friction of vapor exceeds gravity (at a high mass velocity and loss a disperse regime could take place), intermediate and gravitational regimes of phases' flow downstream the movement of vapor and condensate. Under full condensation of vapor at the end of the tube $x \to 0$. Then the tube could be completely flooded with condensate and overcooling of condensate occurs in this zone.

Under prevailing impact of vapor velocity heat exchange occurring in laminar film condensation in tube could be estimated by theoretical correlation of Nusselt [1] obtained by him for condensation of vapor flowing on vertical surface. Design relationship looks:

$$\alpha = \sqrt[3]{\frac{\lambda_1^3 r C_f \rho_v u_v^2}{6 v_1 z \Delta T}} \tag{1}$$

Assigned heat flux q = const. and transformation of eq. (1) into dimensionless form yields:

$$\mathrm{Nu_f} = 0.5(C_f \mathrm{Fr})^{0.5}\ \mathrm{Re_f^{-0.5}} \tag{2}$$

In vertical tube, laminar flow of condensate film is broken at $\mathrm{Re_f} > 100$-500. Exact values of critical number $\mathrm{Re_f}$ for flowing vapor are not known, particularly for condensation inside horizontal tube.

When the flow of condensate film is turbulent, to estimate heat exchange the equations of motion and energy:

$$\tau = \mu_1 \left(1 + \frac{\varepsilon_\tau}{v_1}\right)\frac{\mathrm{d}u}{\mathrm{d}y} \tag{3}$$

$$q = -\rho_1 c_{p1} \lambda_1 \left(1 + \frac{\varepsilon_q}{\lambda_1}\right)\frac{\mathrm{d}T}{\mathrm{d}y} \tag{4}$$

where ε_τ is the coefficient of the turbulence of momentum transfer and ε_q – the coefficient of turbulence of heat transfer, are jointly solved.

More often, for turbulent flow the universal relationships reflecting film flow distribution that were submitted by Karman [15] are used. They are:

– in laminar layer, if $y^+ < 5, u^+ = y^+$ (5)

– in buffer layer, if $5 \le y^+ < 30, u^+ = -3.05 + 5 \ln y^+$ (6)

– in turbulent layer, if $30 \le y^+, u^+ = 5.05 + 2.5y^+$ (7)

Dimensionless velocity distribution during turbulent flow is expressed:

$$u^+ = \frac{u}{u^*} \tag{8}$$

where $u^* = (\tau_\mathrm{f}/\rho_1)^{0.5}$.

Dimensionless wall distance is expressed:

$$y^+ = \frac{yu^*}{\nu_1} \tag{9}$$

To solve eq. (3) it is necessary to know friction stress, τ determining frictional pressure drop:

$$\left(\frac{\Delta P}{\Delta z}\right)_\mathrm{f} = \frac{-4\tau_\mathrm{f}}{d} \tag{10}$$

Estimation of frictional pressure drop is one of the complicated procedures as to accuracy and correctness.

By using values of τ_f, it is possible jointly to solve eqs. (3), (5)-(7) for estimation of ε_τ, in any zone of film thickness. The known solutions [16, 17] serve for definition of parameter E – the ratio of vortex (turbulent) viscosity ε_τ to vortex heat conductivity ε_q. Usually $E = 1$.

The values of ε_τ and ε_q for each zone define local heat-transfer coefficient as:

$$\frac{1}{\alpha} = \int_0^{\delta+} \frac{\nu_1}{\rho_1 c_{p1}(a_1 + E\varepsilon_\tau)u^*} \mathrm{d}y^+ \tag{11}$$

On making some assumption substantiated in Treviss *et al* [4] submit dimensionless relationship for local Nuselt number:

$$\frac{\mathrm{Nu}F_2}{\mathrm{Pr_l}\ \mathrm{Re_l^{0.9}}} = F(X_\mathrm{tt}) \tag{12}$$

$$F(X_\mathrm{tt}) = 0.15\left(\frac{1}{X_\mathrm{tt}} + \frac{2.85}{X_\mathrm{tt}^{0.476}}\right) \tag{13}$$

To find F_2 for each layer the corresponding relationships are obtained:

$$F_2 = 0.707\ \mathrm{Pr_1}\ \mathrm{Re_1}^{0.5}, \quad \text{if } y^+ < 5 \tag{14}$$

$$F_2 = 5\ \mathrm{Pr_l} + 5\ \ln[1 + \mathrm{Pr_l}(0.09636\ \mathrm{Re_l^{0.585}} - 1)], \quad \text{if } 5 \le y^+ < 30 \tag{15}$$

$$F_2 = 5\ \mathrm{Pr_l} + 5\ \ln(1 + 5\ \mathrm{Pr_l}) + 2.5\ \ln(0.00313\ \mathrm{Re_l^{0.812}}), \quad \text{if } 30 \le y^+ \tag{16}$$

Dukler [18] solved the problem of heat exchange for condensation of vapor flowing inside vertical tube and therefore gravity is taken into account in his solution. The results of calculation are graphically represented as $\mathrm{Nu_f} = \mathrm{f}(\mathrm{Re_l}, \beta)$ for $\mathrm{Pr_l}$ within the range from 1 to 5. Bae *et al.* [3] plotted the diagrams that completely agree with the Dukler's diagrams. In these diagrams the dimensionless heat-transfer coefficient:

$$\mathrm{Nu_f} = \frac{\alpha}{\lambda_l}\sqrt[3]{\frac{\nu_l^2}{\mathrm{g}}} \tag{17}$$

used for liquid film flow is plotted *vs.* the values of $\mathrm{Re_l}$. As a parameter related to friction stress at the interphase, correlation:

$$\beta = \sqrt[3]{\frac{\tau_f^3}{\rho_l g^2 \mu_l^2}} \tag{18}$$

is reflected in the diagram.

Solutions concerning heat exchange, when condensation occurs in an annular flow of the phases, are represented in the works of Kosky and Staub [5] and Agra and Teke [19] similarly to [3, 4], [18]. Cavallini *et al.* [20] used the method of Kosky and Staub [5] with friction correction of pressure drop.

In the case of high mass flow rates, when the most part of tube from inlet is in the regime of intensive condensate loss by vapor and flow is almost dispersed, Ananiev *et al.* [21] and Akers *et al.* [22] proposed semi-empirical model of homogenous phases flow with numerous assumptions and the procedure for estimation of local and mean heat-transfer coefficients. On making several disputable simplifications the design correlation for prediction of local $\mathrm{Nu} = \alpha d/\lambda_l$ was proposed in [21]:

$$\mathrm{Nu} = c\ \mathrm{Re_l^{0.8}}\ \mathrm{Pr_l^{0.43}}\sqrt{1 + x\left(\frac{\rho_l}{\rho_v} - 1\right)} \tag{19}$$

where coefficient c is 0.024 for stainless steel and 0.032 for copper and brass.

The main problems and differences in theoretical study of condensation in an annular regime, as well as in deriving empirical correlations for estimation of heat transfer, consist in the methods of definition of frictional pressure drop in two-phase flow $(\Delta P/\Delta z)_f$. The value of this quantity is required to solve the system of eqs. (3)-(7) or to find Nusselt number by the diagrams from [3]. The value of $(\Delta P/\Delta z)_f$ is also needed to calculate total pressure drop $\Delta P/\Delta z$, which for horizontal tube includes accelerating pressure drop $(\Delta P/\Delta z)_m$:

$$\frac{\Delta P}{\Delta z} = \left(\frac{\Delta P}{\Delta z}\right)_f + \left(\frac{\Delta P}{\Delta z}\right)_m \tag{20}$$

The component of accelerating pressure drop is calculated for annular flow by the formula from [20]:

$$\left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{m}} = G^2 \frac{d}{\Delta z}\left[\frac{x^2}{\rho_{\mathrm{v}}\varepsilon} + \frac{(1-x)^2}{\rho_{\mathrm{l}}(1-\varepsilon)}\right] \tag{21}$$

For calculation of ε, 35 formulas were analyzed only in the study by Dalkilic and Wongwises [12]. At high vapor contents (annular flow), when $x > 0.5$, these formulas differ in the value of ε in the second or third character after zero. It is easy to show that if $\varepsilon > 0.9$, then such a small difference could change $(\Delta P/\Delta z)_{\mathrm{m}}$ by 20 to 200%.

In most works the Lockard-Martinelli method [23] developed for homogenous two-phase flow without condensation is used for estimation of $(\Delta P/\Delta z)_{\mathrm{f}}$:

$$\left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{f}} = \Phi_{\mathrm{v}}^2 \left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{v}} = \Phi_{\mathrm{l}}^2 \left(\frac{\Delta P}{\Delta z}\right)_{\mathrm{l}} \tag{22}$$

where $(\Delta P/\Delta z)_{\mathrm{v}}$ and $(\Delta P/\Delta z)_{\mathrm{l}}$ are pressure losses, respectively, either only vapor or only liquid flows in the tube; Φ_{v}^2, Φ_{l}^2 are the coefficients depending upon dimensionless parameter X_{tt}.

Autee *et al.* [23], Dalkilic *et al.* [24], and Kandlikar *et al.* [11] represented more than twelve empirical correlations for $(\Delta P/\Delta z)_{\mathrm{f}}$. As follows from these works, the available design correlations are significantly unmatched.

Dalkilic [25] determines frictional pressure loss by using other methods. He applies six different correlations for calculation of friction coefficient C_{f} that enters into well-known relationship for τ_{f} and, respectively, for $(\Delta P/\Delta z)_{\mathrm{f}}$:

$$\tau_{\mathrm{f}} = \frac{C_{\mathrm{f}}\rho_{\mathrm{v}}u_{\mathrm{v}}^2}{2} \tag{23}$$

The author compares calculated values of C_{f} with Eckels and Pate's experimental data, where the total pressure drop under condensation of refrigerants R400a, R502, and R507c inside smooth horizontal tube at G = 120-600 kg/m^{2}s and x = 0.23-0.6 was measured. Experimental friction coefficient was calculated by formula:

$$C_{\mathrm{f}} = \frac{2d\rho_{\mathrm{l}}\Delta P_{\mathrm{f}}}{G_{\mathrm{eq}}^2 4z} \tag{24}$$

Equivalent mass flow rate G_{eq} is the function of the relationship cited in [26] and [22]:

$$G_{\mathrm{eq}} = G\left[(1-x) + x\sqrt{\frac{\rho_{\mathrm{l}}}{\rho_{\mathrm{v}}}}\right] \tag{25}$$

Frictional pressure drop ΔP_{f} was calculated by subtraction of the accelerating pressure drop from the measured total pressure drop ΔP computed by eq. (21), in which volumetric vapor content ε is determined by Rigots's formula:

$$\varepsilon = \frac{1}{1 + \dfrac{2(1-x)\sqrt{\dfrac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}}}}{x}} \tag{26}$$

Thus, there are the errors resulted again from correctness of high (more than 0.8) values of ε calculated by different correlations. Comparison of the predicted and experimental values of $(\Delta P/\Delta z)_\mathrm{f}$ enabled Dalkilic to come to the conclusion that the best convergence had occurred, if Carey's formula:

$$C_\mathrm{f} = 0.079\left[\frac{Gx(d-\delta)}{\mu_\mathrm{v}\left(1-\dfrac{4\delta}{d}\right)}\right]^{-0.25} \tag{27}$$

which, in fact, was the known correlation for one-phase flow with correction for volumetric vapor content, was used for calculation of C_f.

Experimental researches

Beginning from one of the first works [27, 28] and until now in all except [7] experimental works on condensation inside horizontal tubes, the perimeter-averaged heat-transfer coefficients at minimal tube length of about 0.3 m, but more often up to 1.5 m, were determined. In the experiments, wall temperature in 3 to 5 places along tube perimeter was measured, while heat flux was averaged for the whole test section as a function of heat transferred to coolant.

Hitherto, the authors tried to conduct the experiments in a way that Δx in the test section was minimal and usually did not exceed 0.1-0.2. Besides, most researchers apply the methods of heat-exchange study, representation of the experimental data, their generalization and prediction based on definition of local heat-transfer dependency upon *local* vapor content x and mass velocity G. In the experiments wall temperature was measured in 2 to 4 points along tube perimeter (top, middle, and bottom) per one cross-section in several parts of the test section. As well only length-averaged heat flux was measured. That is why angular co-ordinateaveraged and lengthaveraged heat-transfer coefficient was always determined.

In the first work of Crosser [27] the experimental data on full condensation of R12 and methanol are represented as the functions of $\alpha = f(G, \Delta T)$ type. The design relationship proposed in [28] has 2-D complexe (not taking into account $\mathrm{Pr_l}$): one in the form of Gd/μ_l (in fact, $\mathrm{Re_l}$), and another connected with ΔT by dimensionless complex M:

$$\mathrm{Nu}\ \mathrm{Pr}_\mathrm{l}^{1/3} = 0.388 M^{1/6}\sqrt[4]{\frac{Gd}{\mu_\mathrm{l}}}\sqrt[8]{\frac{\rho_\mathrm{l}}{\rho_\mathrm{v}}} \tag{28}$$

where $M = [(\rho_\mathrm{l}-\rho_\mathrm{v})\rho_l r d^3]/8\lambda_\mathrm{l}\mu_\mathrm{l}\Delta T$.

Akers *et al.* [22] proceeded to derivation of the relationships for calculation of local and average heat transfer in Nu = f ($\mathrm{Re_1}$, $\mathrm{Pr_1}$) form with various additions including vapor content x. In this way the authors obtained the following relationship for calculation of average heat transfer:

$$\overline{\mathrm{Nu}} = c\ \mathrm{Re}_\mathrm{eq}^n\ \sqrt{\mathrm{Pr}} \tag{29}$$

$$\mathrm{Re_{eq}} = \frac{G_\mathrm{eq} d}{\mu_\mathrm{l}} \tag{30}$$

where $c = 0.0265$, $n = 0.8$ at $\mathrm{Re_{eq}} > 5\cdot10^4$ and $c = 0.503$, $n = 1/3$ at $\mathrm{Re_{eq}} < 5\cdot10^4$. The G_eq was calculated by eq. (25) at arithmetical mean tube along value of x.

Kaushik and Azer [29] showed that eq. (29) demonstrated good agreement with the experimental data by Said [30] on condensation of R113 and by Venkatesh [31] on condensation of R11.

The $\mathrm{Re_{eq}}$ is also used in work [29], in which condensation of R113 and R11 inside the test sections consisting of sequentially jointed sections of 0.6 m length is investigated. Three test sections with inner diameters of 12.5, 15.8, and 17.8 mm were used in the experiments. The mean value of α for l = 0.6 m and G = 70-220 kg/m^{2}s was determined. The authors proposed the following relationship for prediction of α:

$$\overline{\mathrm{Nu}} = 2.078\,\mathrm{Re_{eq}^{0.507}}\,\mathrm{Pr_l^{0.33}}\left(\frac{\Delta x d}{z}\right)^{0.198} p_r^{-0.14} \tag{31}$$

It should be noted that $p_r = p/p_{cr}$ (ratio of R113 experimental pressure to critical one) was used for prediction of heat transfer under condensation inside tubes yet by Borishanskiy *et al.* [32].

Kaushik and Azer [29] noticed satisfactory convergence (±30%) of prediction by eq. (31) with our own experiments, the experiments from [30, 31, 33] and a part of the data from [34]. However, in some experiments from [33] in the region of α high values (annular regime) they exceeded the predicted values by more than 30%.

Cavallini and Zecchin [35] used $\mathrm{Re_{eq}}$ in another form:

$$\mathrm{Re_{eq}} = \mathrm{Re_v}\sqrt{\frac{\rho_\mathrm{l}}{\rho_\mathrm{v}}}\frac{\mu_\mathrm{v}}{\mu_\mathrm{l}} + \mathrm{Re_l} \tag{32}$$

where $\mathrm{Re_v} = Gdx/\mu_\mathrm{l}$ and $\mathrm{Re_l} = Gd(1-x)/\mu_\mathrm{l}$.

In the similar unusual form (in the denominator, liquid viscosity is written instead of μ_v) Re_v is used by Tandon *et al.* [36] and Shao and Granryd [37].

Correlation of Cavallini and Zecchin [35] for α is:

$$\mathrm{Nu} = 0.05\,\mathrm{Re_{eq}^{0.8}}\,\mathrm{Pr_l^{0.35}} \tag{33}$$

This correlation is often used by many authors for comparison of the experimental data.

Fujii [38, 39], Koyama *et al.* [40], and Yu *et al* [41] compared the experimental data obtained by the authors under condensation of R22, R134a, and R123 inside horizontal tube with the results predicted by the relationships of Azer *et al.* [42], Cavallini and Zecchin [35], Shah [43] and Kaushik and Azer [29], as well as with the method developed by the authors. To determine mean heat-transfer coefficients under condensation inside tubes in case of annular, stratified, and intermediate flow regimes they propose the following relationship for the first time cited in [44]:

$$\overline{\mathrm{Nu}} = (\mathrm{Nu_f^{\mathit{n}}} + \mathrm{Nu}_b^n)^{1/n} \tag{34}$$

where $\mathrm{Nu_f}$ is a *convective* heat transfer occurring in the zone of dominating effect of vapor velocity, Nu_b – a heat transfer with dominating effect of gravity. It should be noted that the Fujii formulas for $\mathrm{Nu_f}$ and Nu_b [38, 39] differ from the Koyama *et al.* formulas [40, 41].

Fujii [39] the comparison of the experimental and predicted data on condensation of R22 is demonstrated. The design relationships by Azer *et al.* [42], Shah [43], Cavallini and Zecchin [35], and Kaushik and Azer [29] give divergence with the experimental data above 100%.

In fig.1 the experimental results obtained by Yu *et al.* [41] for R134a and R22 are plotted. The heat transfer coefficient for R22 weakly changes with increasing x and G that does not agree with the data of many other researchers presented in this work. The data for R134a reveal yet one interesting feature: α differs at close G, 300 and 340 kg/m^{2}s, equal x, but different q. In particular, at q = 26-40 kW/m^2 it is by 40-60% higher than at q = 14-22 kW/m^2.

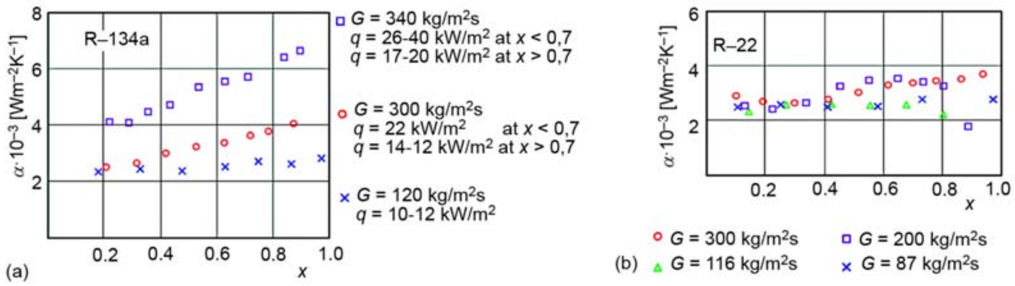

Figure 1. Experimental heat transfer coefficients [41] for condensation of R-134a (a) and R-22 (b) *vs.* vapor quality

Shah [43] submitted simple method of local and mean heat transfer prediction, in which the correlations for one-phase convection in liquid turbulent flow in tubes are corrected by function Ψ including vapor content x and reduced pressure p_r. For local heat transfer the design formula looks like:

$$\alpha = \alpha_c \Psi \tag{35}$$

where $\alpha_c = 0.023\,\mathrm{Re}_\mathrm{l}^{0.8}\mathrm{Pr}_\mathrm{l}^{0.43}(\lambda_\mathrm{l}/d)$ and Ψ is written:

$$\Psi = (1-x)^{0.8} + 3.8x^{0.76}\,\frac{(1-x)^{0.04}}{p_r^{0.38}} \tag{36}$$

For mean heat transfer at x_in = 1 and x_out = 0:

$$\alpha = \alpha_c\left(0.55 + \frac{2.09}{p_r^{0.38}}\right) \tag{37}$$

Shah [10] in specified correlation (35) inserting the complex $(\mu_\mathrm{l}/14\mu_\mathrm{v})^n$:

$$\alpha_I = \alpha_c\left(\frac{\mu_\mathrm{l}}{14\mu_\mathrm{v}}\right)^n\left[(1-x)^{0.8} + 3.8x^{0.76}\,\frac{(1-x)^{0.04}}{p_r^{0.38}}\right] \tag{38}$$

where $n = 0.0058 + 0.557\,p_r$.

The author proposed to apply this correlation for heat transfer in regime I that occurs, if:

$$J_g \geq 0.98(Z + 0.263)^{-0.62} \tag{39}$$

where $Z = [(1/x) - 1]^{0.8} \mathrm{Pr}_l^{0.4}$.

Besides, Shah proposed very simplified determination of the boundary between regimes I and II (fig. 2) and the design method for α in regime II and $\mathrm{Re}_{vo} > 35000$:

$$\alpha = \alpha_I + \alpha_{\mathrm{Nu}} \tag{40}$$

where α_{Nu} was calculated from the Nusselt formula for vertical tube:

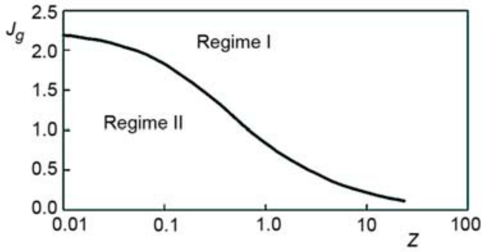

Figure 2. Regimes in horizontal tubes according to Shah [10]

$$\alpha_{\mathrm{Nu}} = 1.32 \, \mathrm{Re}_l^{-1/3} \left[\rho_l(\rho_l - \rho_v) \frac{g\lambda_l^3}{\mu_l^2} \right]^{1/3} \tag{41}$$

Shah [43], compared his correlation (35) with the experimental data of ten authors on condensation of R11, R22, R113, H_2O, methanol, and benzene within the range of *G* from 22 to 1600 kg/m^{2}s, *x* from 0 to 1.0, and reduced pressure from 0.013 to 0.44. Since the results have been represented as dimensionless complex $\Psi = h_n/h_l$ *vs.* $Z = [(1/x) - 1]^{0.8} \mathrm{Pr}^{0.4}$, then it is impossible to make detailed analysis, *i. e.*, to determine the value of the parameters affecting heat transfer that corresponds to the maximal divergence.

Shah [10], compared the calculations by new formulas (38) and (40) with the experimental data of twenty six authors including the works till 2008. In this work, there are tables with the information on the authors of the experiments, experimental data and the rate of divergence with the calculations. Several diagrams of $\alpha = f(x)$ type with these comparisons are represented as well. In general, Shah considers that new correlations (38) and (40) perfectly generalize the data he used for comparison.

Let us consider two other correlations that include p_r. The first one is proposed by Tang *et al.* [45] for calculation of mean heat-transfer coefficient:

$$\mathrm{Nu} = 0.023 \, \mathrm{Re}^{0.8} \mathrm{Pr}_l^{0.4} \left\{ 1 + 4.863 \left[-\ln(p_r) \frac{x}{1-x} \right]^{0.836} \right\} \tag{42}$$

The second one is proposed by Bives and Yokozeki [46]:

$$\mathrm{Nu} = \mathrm{Nu}_{\mathrm{Shah}} \left(0.78738 + \frac{6187.89}{G^2} \right) \tag{43}$$

where $\mathrm{Nu}_{\mathrm{Shah}} = \alpha_c d/\lambda_l$, and α_c is determined as in correlation (35).

On the basis of the experiments on condensation of R12 and R22 inside the tubes of d = 10 mm at G = 175-650 kg/m^{2}s and $0.1 < x < 0.99$ Tandon *et al.* [36] proposed two formulas close to the Akers *et al.* [22] correlations and Shah [43]:

– for annular and intermediate flow regimes ($\mathrm{Re}_v \geq 30\,000$):

$$\mathrm{Nu} = 0.084 \, \mathrm{Re}_v^{0.67} \mathrm{Pr}_l^{1/3} \left(\frac{1}{\mathrm{Ja}_l} \right)^{1/6} \tag{44}$$

– for gravity flow regime ($\mathrm{Re_v}$ < 30 000):

$$\mathrm{Nu} = 23.1\,\mathrm{Re_v^{1/8}}\;\mathrm{Pr_l^{1/3}}\left(\frac{1}{\mathrm{Ja_l}}\right)^{1/6} \tag{45}$$

According to these correlations for all of the regimes the effect of both gravity and friction at the interphase expressed by $\mathrm{Re_v}$ and $\mathrm{Ja_l}$ numbers takes place. On the other hand, this effect is not detected in most correlations derived by other authors for the region of annular and intermediate flow of the phases. Thome [47], Dobson and Chato [48], and Lim *et al.* [49] also use $\mathrm{Re_v}$ in its various forms in the correlations for heat transfer in a stratified regime.

Often, many authors compare their experimental data with Dobson and Chato [48] following relationship:

$$\mathrm{Nu} = 0.023\,\mathrm{Re_l^{0.8}}\;\mathrm{Pr_l^{0.4}}\left(1+\frac{2.22}{X_{\mathrm{tt}}^{0.89}}\right) \tag{46}$$

For dominating gravity regime Dobson and Chato [48] propose the relationship containing $\mathrm{Re_v}$:

$$\mathrm{Nu} = \frac{0.23\,\mathrm{Re_{vo}^{0.12}}}{1+1.1X_{\mathrm{tt}}^{0.58}}\left(\frac{\mathrm{Ga\;Pr_l}}{\mathrm{Ja_l}}\right)^{0.25} + \left[\frac{\arccos\left(2\varepsilon-1\right)}{\pi}\right]\mathrm{Nu_{forced}} \tag{47}$$

where $\mathrm{Nu_{forced}} = 0.0195\,\mathrm{Re_l^{0.2}}\;\mathrm{Pr_l^{0.4}}(1.376 + c_1/X_{tt}^{c_2})^{0.5}$.

Constants c_1 and c_2 are of different values depending upon $\mathrm{Fr_l}$.

Garcia-Valladares [9] compares the calculations by eqs. (35), (42)-(47) and others mentioned above with the experiments of different authors used various liquids. This comparison demonstrates probability of a great divergence of the cited formulas and the calculations. At that, because of absence of the primary data (first of all, for q) it is difficult to determine the conditions, when the experimental data deviate from the design correlations.

Breber *et al.* [50] advise the relationship for α:

$$\alpha = \alpha_c(\Phi_l^2)^m \tag{48}$$

where $\Phi_l^2 = 1 + c/X_{\mathrm{tt}} + 1/X_{\mathrm{tt}}^2$ and the constants in these formulas are: c_1 = 0.022, c = 20.0, m = 0.5, a = 0.9, and b = 0.5.

Park *et al.* [51] researched condensation of R22, propylene, dimethyl ether, and isobutene in the tubes of d = 8.8 mm and l = 0.53 m at q = 7.3-7.7 kW/m^2 and t_s = 40 °C. The experimental data are compared with the design correlations developed by Akers *et al.* [22], Cavallini and Zecchin [35], Shah [43], Dobson and Chato [48], Soliman *et al.* [52], and Traviss *et al.* [4] with the Jung *et al.* formula [53]:

$$\alpha = 22.4\alpha_c\left(1+\frac{2}{X_{\mathrm{tt}}}\right)^{0.81}\left(\frac{q}{rG}\right)^{0.33} \tag{49}$$

In this formula, parameter $q/(rG)$ of liquid suction effect in vapor-liquid interphase appears. For the first time its impact on variation of local heat transfer has been studied in [7] and will be in detail analyzed in part 2 of this work. As follows from [51], all relationships, except (49), give deviation from the experimental heat-transfer coefficients up to 27-50%.

Nualboonrueng *et al.* [54] obtained very high rates of G effect on α in the experiments with condensation of R134a in the tube of l = 2.5 m and d = 8.1 mm, if t_{s} = 30 °C and 40 °C. When G changes from 400 to 800 kg/m^{2}s and x > 0.5, annular flow regime of the phases takes place and the rate of G effect reaches 1.0. The experimental data of Nualboonrueng [54] agreed with the calculations by the Cavallini and Zecchin correlation [35] within ±30%. To predict α the author proposes updated Cavallini and Zecchin [35] relationship that differs by the rate of $\mathrm{Re_{eq}}$ and $\mathrm{Pr_l}$ effect and generalizes the experimental data with uncertainty ±20%:

$$\mathrm{Nu} = 0.003\,\mathrm{Re_{eq}^{0.997}}\;\mathrm{Pr_l^{0.932}} \tag{50}$$

Most methods of heat transfer prediction and researches concerning condensation of various refrigerants were developed by Cavallini with co-authors and by Thome. The same authors in detail analyzed the flow pattern maps made by different authors and specified boundaries of the regimes. Equation (33) was one of the first proposed by Cavallini and Zecchin [35] for calculation of heat transfer and until now it is often used by different authors for comparison with the experimental data.

Cavallini *et al.* [20] compared their experimental data on condensation of R134a, R125, R32, R410A, R236, and R22 with Kosky and Staub [5] theoretical solution for annular regime and with Nusselt formula (1) for stratified flow. In the experiments condensation took place inside the tube of d = 8 mm and l = 1.0 m. Wall temperature was measured in four places [top, bottom, and two places in the middle (φ = 90°) of the tube] at the distance of 0.1 m downstream from tube inlet and 0.1 m upstream to tube outlet. In the experiments vapor temperature was the same (t_s = 40 °C), G = 65-750 kg/m^{2}s, and heat flux rate varied from 6 to 62 kW/m^2 depending upon G, if vapor content Δx was within the range of 0.2-0.4. In all experiments total $\Delta P/\Delta z$ was measured too. Then accelerating pressure drop $(\Delta P/\Delta z)_\mathrm{m}$ was estimated by using eq. (21). At that, volumetric vapor content ε in $(\Delta P/\Delta z)_\mathrm{m}$ was calculated following the model of Rouhani [55]. Frictional pressure drop $(\Delta P/\Delta z)_\mathrm{f}$ used in the Kosky and Staub [5] design model was determined on the basis of eq. (20). In fig. 3 the obtained experimental data were compared with the calculations. Despite the authors note in their work that an average divergence does not exceed 21%, as follows from fig. 3, the data on R32 and R134 are diverged by 40-50%.

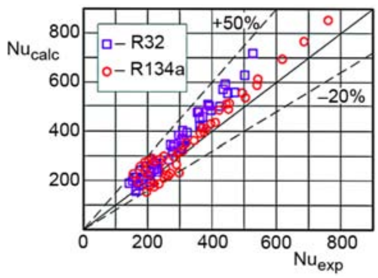

Figure 3. Calculated *vs*. experimental Nusselt number [20]

Cavallini *et al.* [8] proposes the Kosky and Staub [5] model for calculation of heat transfer in an annular regime (J_g > 2.5), but in this case they determine the frictional pressure drop by eq. (22). In contrast to the Dobson and Chato method [48], besides the stratified region (J_g < 2.5 and X_tt > 1.6) an intermediate (annular-stratified) flow region, where J_g < 2.5 and X_tt > 1.6, is also separated. In order to estimate heat transfer in these regions intricate correlations are written down.

In [8] the model described above is compared with the calculations by Dobson and Chato [48] and Shah [43]. For R22 and R134a and for all values of G three methods give close (±15%) convergence with the experiments. For R410a, if G = 750 kg/m^{2}s, the Shah [43] and Dobson and Chato [49] methods result in α exceeding the experimental data by 25-35%. Comparison of all experimental data for R32 with the calculations is plotted in fig. 4 and it demonstrates that some data lie by more than 50% lower than those calculated.

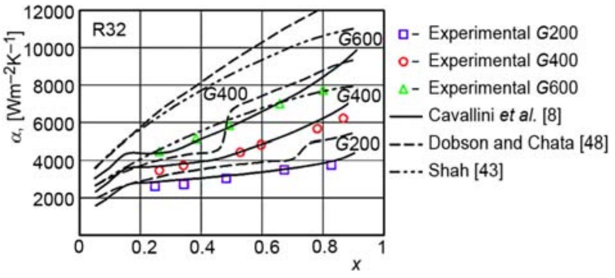

Figure 4. Comparison between experimental and predicted heat transfer coefficient for R32 [8]

Thome *et al.* [56] propose another *new* model of condensation heat transfer that is based, as the author notes, on a simplified structure of phase flow regimes and takes into account impact of the interphase waves on heat transfer.

The authors propose a simplified structure of liquid distribution. In annular, intermediate and mist regimes the condensate flows over the whole tube's perimeter (flooding angle $\theta = 0$) and film flow is turbulent. In stratified or wavy-stratified patterns, condensate film flows down from the tube's top to the stream under gravity and heat transfer in this section of the tube is determined by Nusselt formula (1).

The following correlation is proposed to calculate heat transfer in annular, intermediate, and mist regimes:

$$\alpha_c = c\ \mathrm{Re}_\mathrm{l}^n\ \mathrm{Pr}_\mathrm{l}^m\ \frac{\lambda_\mathrm{l}}{\delta}\ f_i \tag{51}$$

where $\mathrm{Re}_\mathrm{l} = 4G(1-x)\delta]/[(1-\varepsilon)\mu_\mathrm{l}$.

Thickness of condensate film δ is adopted in this model as a characteristic dimension. It follows from an evident relationship for annular regime:

$$\delta = d\,\frac{1-\varepsilon}{4} \tag{52}$$

To estimate ε the authors proposed the following relationship without any objective substantiation:

$$\varepsilon = \frac{\varepsilon_h - \varepsilon_{ra}}{\ln\dfrac{\varepsilon_h}{\varepsilon_{ra}}} \tag{53}$$

where ε_h and ε_{ra} are calculated by the Zivi [57] formulas for homogenous flow and by the Rouhani [55] formula, respectively.

El Hajal *et al.* [58] compare the data calculated by eq. (51) for f_i = 1.0, c = 0.0039, n = 0.734, and m = 0.5. But in [56] heat transfer is estimated for c = 0.003, n = 0.74, and coefficient f_i is expressed:

$$f_i = 1 + \left(\frac{u_\mathrm{v}}{u_\mathrm{l}}\right)^{0.5} \left[(\rho_\mathrm{l} - \rho_\mathrm{v})\,\frac{\mathrm{g}\delta^2}{\sigma}\right]^{0.25} \tag{54}$$

where $u_\mathrm{v} = Gx/\rho_\mathrm{v}\varepsilon$ and $u_\mathrm{l} = [G(1-x)]/[\rho_\mathrm{l}(1-\varepsilon)]$ are the mean velocities of the phases.

Substantiation of function f_i with included complex $(u_\mathrm{v}/u_\mathrm{l})^{0.5}$ and $[(\rho_\mathrm{l} - \rho_\mathrm{v})\mathrm{g}\delta^2/\sigma]^{0.25}$ is as follows. Calculation of the data of the different authors by the proposed method shows good agreement, however it is not better than by another method of the same authors [20], where the Kosky and Staub [5] theoretical model is used.

Cavallini *et al.* [59] proposed yet one simplified model of heat transfer calculation. Referring to their work [20] the authors consider that in the region of convective heat exchange occurring in annular pattern of phase flow heat transfer does not depend upon temperature drop. This viewpoint is not valid, at any rate, for laminar flow of condensate film at tube inlet. In gravity regime the authors accept the dependence of heat transfer upon Δt in accordance with Nusselt formula (1). By using criteria J_g and X_{tt} for definition of the pattern boundaries the authors submit the following flow map (fig. 5) and calculation method.

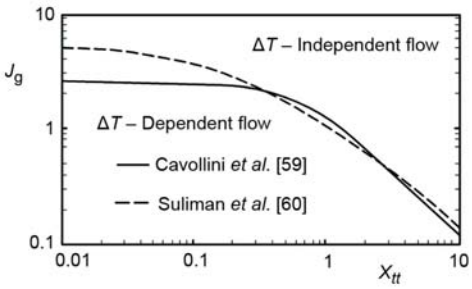

Figure 5. Cavallini flow regime map [59] improved by Suliman *et al.* [60]

In the regime independent upon ΔT ($J_g > J_g^T$):

$$J_g^T = \left[\left(\frac{7.5}{4.3X_{\mathrm{tt}}^{1.111} + 1} \right)^{-3} + C_T^{-3} \right]^{-1/3} \tag{55}$$

where $C_T = 1.6$ for hydrocarbons and $C_T = 2.6$ for refrigerants.

$$\alpha_A = \alpha_{LO} \left[1 + 1.128x^{0.8170} \left(\frac{\rho_\mathrm{l}}{\rho_\mathrm{v}} \right)^{0.3685} \left(\frac{\mu_\mathrm{l}}{\mu_\mathrm{v}} \right)^{0.2363} \left(1 - \frac{\mu_\mathrm{v}}{\mu_\mathrm{l}} \right)^{2.144} \mathrm{Pr}_\mathrm{l}^{-0.1} \right] \tag{56}$$

In the regime depending upon Δt ($J_g \le J_g^T$):

$$\alpha_D = \left[\alpha_A \left(\frac{J_g^T}{J_g} \right)^{0.8} - \alpha_{\mathrm{STRAT}} \right] \left(\frac{J_g}{J_g^T} \right) + \alpha_{\mathrm{STRAT}} \tag{57}$$

$$\alpha_{\mathrm{STRAT}} = 0.725 \left[1 + 0.741 \left(\frac{1-x}{x} \right)^{0.3321} \right]^{-1} \left[\frac{\lambda_\mathrm{l}^3 \rho_\mathrm{l} (\rho_\mathrm{l} - \rho_\mathrm{g}) rg}{\mu_\mathrm{l} d \, \Delta T} \right]^{0.25} + (1 - x^{0.087}) \alpha_{\mathrm{LO}} \tag{58}$$

$$\alpha_{\mathrm{LO}} = 0.023 \, \mathrm{Re}_{\mathrm{lo}}^{0.8} \, \mathrm{Pr}_\mathrm{l}^{0.4} \, \frac{\lambda_\mathrm{l}}{d} \tag{59}$$

The authors consider that the proposed model perfectly generalizes most experimental data represented in [59], as is often made by many authors, in terms of diagram $\alpha_{\mathrm{calc}} = f(\alpha_{\mathrm{exp}})$. However, it is clear from the same work that *new* model leads to wide deviation from the experiments (fig. 6).

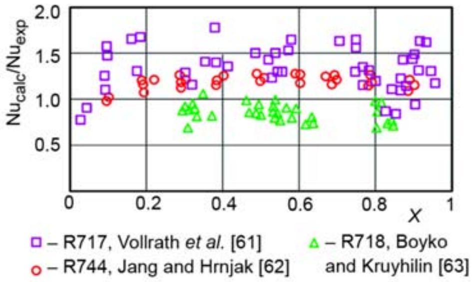

Figure 6. Calculated and experimental Nusselt number *vs.* vapor quality [59]

Soliman *et al.* [60] performed the experiments on condensation of R134 at t_s = 40 °C inside the tubes of l = 2.5 m and d = 8.38 mm in changing vapor content from 0.03 to 0.76. To provide the minimal change in Δx along the tube (within 0.07-0.1) the experiments were performed at low values of q (from 4 to 8 kW/m^2). In fig. 7, the experimental data were obtained at G = 100 and 300 kg/m^{2}s. The experimental data from [20] and [48] for G = 75, 100, and 300 kg/m^{2}s were plotted as well. It should be noted that α from [60] is by 40-60% lower than the data of other authors. Comparison of the experimental values of α with calculations by Thome *et al.* model [56] and Cavallini *et al.* [59] showed that estimation by these models results in higher values of α. In particular, the Thome *et al.* model increases the values up to 80% and the Cavallini *et al.* model increases them by 25-30%.

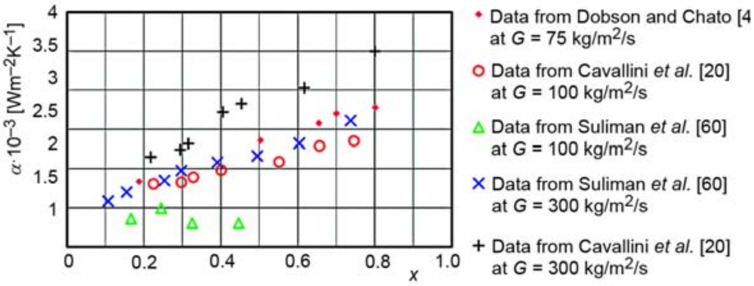

Figure 7. Comparison between Suliman *et al.* [60] and other authors' experimental data at the same parameters

The authors [60] consider that the Cavallini *et al.* flow map, shown in fig. 5, should be defined more precisely, exactly the area of gravity flow that depends upon Δt and where heat transfer, on opinion of the authors, is lower than in annular regime, should be increased. Application of clarified boundaries made it possible to obtain better agreement of the authors' experiments with calculations. However, it is still unclear why the values of α in [60] are lower than in [56].

Comparison between different models for calculation of α is given in [64]. Here, the calculations by the models proposed by Thome *et al.* [56], Cavallini *et al.* [8], Dobson and Chato [48] and by the authors of the model using the eq. (51) of Thome *et al.* and eq. (46) of Dobson-Chato for annular regime with definition of applicability range of these formulas in accordance with a new probabilistic flow map were compared. To create the map the authors implemented a new criterion and called it a *time fraction* without analysis of the accuracy of its substantiation. Comparison between all the models and the experiments on seven refrigerants was analyzed in [64]. As seen, with the exception of the data on mixture R32/R125 60/40%, without doubt, more precise agreement with the experiments has simple the Dobson and Chato model [48] and the theoretical model submitted by Cavallini *et al.* [8].

Van Rooyen *et al.* [65], define the boundary of existence of shear-force and stratified flow patterns a time fraction parameter is used as well. The results of heat transfer estimation during condensation of R22 and R134 in the tube of l = 1.5 m and d = 2.53 mm demonstrated significant difference (above 100%) from Jassim *et al.* [64] data represented in the considered work.

Comparison between theoretical (semi-empirical) solutions and experimental data

Bae *et al.* [2, 3] and Traviss *et al.* [4] carried out numerous experiments on condensation of R12 and R22 at similar experimental set-ups. The only difference was that the tubes in [2, 3] were of 12.5 mm diameter and 5.55 m length, while in [4] they were of 8.0 mm diameter and 5.03 m length. It is one of interesting results that q local within the test section effects on α at unchangeable t_s and G, as seen in tab. 1, if to tabulate the data from [2].

Table 1. Experimental data from [2]

Parameters	Run	Mean vapor quality, x	q, [kWm^{-2}]	α, [Wm^{-2}K^{-1}]
R12; t_s = 40 °C; G = 420-440 kg/m^{2}s	22	0.9	35.5	4201
	23	0.89	42.2	5631
	24	0.87	50.7	4911
	22	0.54	27.2	3139
	23	0.45	32.9	4167
	24	0.59	49.7	5086

The second interesting result is the difference between the data from [4] and [2, 3] at x > 0.4-0.6 for R12 and R22. In fig. 8, the heat-transfer coefficients are plotted *vs.* average velocity u_v at constant t_s. It is seen that the heat-transfer coefficients taken out from [4] at constant u_v exceed α from [3] and [20] by 20-50%. We should note that data for R22 from [3] at constant t_s and G are in a good agreement with the experiments [20] within the whole range of x changing.

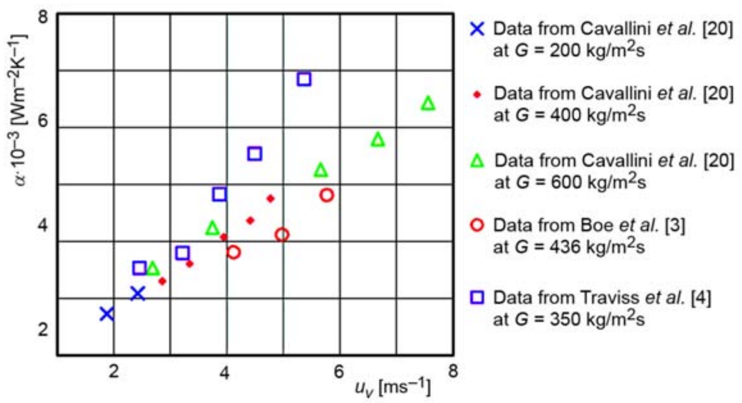

Figure 8. Comparison between Traviss *et al.* [4] and other authors' experimental data at the same parameters

Comparison between the experiments and the calculations based on similar models described above demonstrated good convergence of experimental α obtained in [2, 3] and divergence of the experimental data obtained in [4] with the predicted results at x > 0.5-0.6.

Comparison between all the experiments performed by Traviss *et al.* [4] on R12 and R22 and the calculations is represented by dimensionless complexes:

$$\frac{\mathrm{Nu}\ F_2}{\mathrm{Pr_l}\ \mathrm{Re_l^{0.9}}} = f[F(X_{tt})]$$

If $f[F(X_{tt})] > 2$ that corresponds to G > 300-400 kg/m^{2}s and x > 0.5-0.6, the experimental data exceed the predicted results by more than 30%. Therefore, to calculate heat transfer the authors propose two correlations:

$$\frac{\mathrm{Nu}\ F_2}{\mathrm{Pr_l}\ \mathrm{Re_l^{0.9}}} = F(X_{tt}) \qquad \text{if} \qquad F(X_{tt}) < 2 \tag{60}$$

$$\frac{\text{Nu}\, F_2}{\text{Pr}_\text{l}\ \text{Re}_\text{l}^{0.9}} = [F(X_\text{tt})]^{1.15} \qquad \text{if} \qquad F(X_\text{tt}) > 2 \tag{61}$$

Kwon *at al.* [66] compare their calculation model and experimental data on condensation of R22 inside the tube of d = 9.5 mm and l = 0.4 m at t_s = 40 °C for two values of G, 300 and 400 kg/m^{2}s. Despite not enough clear details of the loss effect on heat transfer, it is seen in [66] that the models by Kwon *at al.* and Traviss *et al.* [4], as well as the correlation by Cavallini and Zecchin [35] give close results and at x = 0.06-0.8 they agree with the experiments within ±15%. Besides, for assessment of the Traviss *et al.* model the authors apply correlation (60) only and it evidently cannot provide convergence at x > 0.6.

Calculations of Agra and Teke [19] are well converged (±20%) both with the Hulburt and Newell experiment [67] on condensation of R134a inside the tube of d = 3.14 mm and l = 0.99 m at G = 300 and 650 kg/m^{2}s and with the predictions by Traviss *et al.* [4] and Shah [43]. Comparison with the design correlation by Cavallini and Zecchin [35] reveals divergence of the experiments exceeding 40%. Their work does not explain in what way (by using what correlation) to predict frictional drop (dP/dz) on the interphase to solve the problem.

The performed above comparison of semi-empirical methods of heat transfer design is valid except for the Thome *et al.* [56] model only for an annular phase flow pattern and probably for an intermediate regime, when there is the effect of vapor velocity on condensate film flow. Therefore, coincidence of the predicted and experimental results at all x and small G corresponding with a stratified pattern seems strange. In additions, there is rather limited number of the experiments selected for comparison with calculations in the works considered above. As noted earlier, to estimate τ_f various methods (relationships) were used. It is shown in [25] that these methods give the divergence exceeding 100%. So, there is also a question as to the convergence of the experimental and predicted data obtained by different authors.

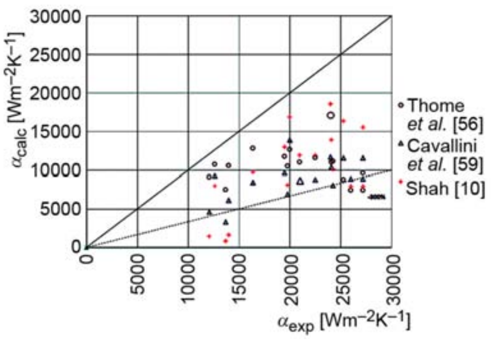

Figure 9. Calculated heat transfer *vs.* experimental data from [68]

In thermal desalinating plants and some water-steam heaters with steam condensation inside horizontal tubes steam pressure and heat flux are of such a value that along almost the whole tube $\text{Re}_\text{s} < 1000$, *i. e.*, there is a laminar flow of condensate film. Then, if p < 1 bar (vacuum desalters), steam velocity reaches 60-80 m/s at tube inlet and its impact on condensate film velocity and heat transfer is strong [68]. As seen in fig. 9, the experimental data for a laminar flow of condensate film substantially differ from the values predicted by Cavallini *et al.* [59], Shah [10], and Thome *et al.* [56].

The experiments at high (from 0.68 to 21.8 MPa) pressures of water steam, when $p_r = 0.03\text{-}0.988$, were performed in the 60$^\text{th}$ by Ananiev [21] Boyko and Kruzhylin [63]. Boyko [69] conducted the experiments on heat transfer rate under full and partial condensation both of pure saturated steam and of water-steam mixture's steam. For the experiments, three steel tubes of 2.5 m length and 17, 13, and 10 mm diameter, as well as the tube of l = 11.9 m and d = 13 mm were used. Pressure of condensing steam was 1.23, 2.45, 5.88, and 8.8 MPa, steam content was from 1.0 to 0.26 at tube inlet and from 0.0 to 0.69 at its outlet, specific heat flux varied from 162 to 1579 kW/m^2, steam mass velocity changed from 154 to

2239 kg/m^{2}s. The results are tabulated. This work demonstrates that all related experimental data of the author quite well agree with the semi-empirical model by Ananiev, Boyko and Kruzhylin (19).

Vera-Garcia *et al.* [70] compared five correlations from [8, 35, 43, 48, and 56] with the experimental data on condensation of R290. As seen from this work, when G = 50 kg/m^{2}s, the Thome *et al.* [56] method gives α that on average are by 25-60% higher than those predicted by other correlations and, when G = 125 and 200 kg/m^{2}s at x > 0.2, this method gives the heat-transfer coefficients by 20-45% lower than the methods by Shah [43], Dobson and Chato [48], Cavallini and Zecchin [35].

Wei *et al.* [13] compared the Aprea *et al.* [71] experimental data on condensation of R22 and R407c in the tube of d = 20 mm and l = 6.6 m at G = 45.5-120 kg/m^{2}s, as well as the data by Cavallini *et al.* [20] and Park *et al.* [51] with the prediction by the Akers *et al.* [22], Cavallini *et al.* [59], and Chato [72] formulas for gravity flow of the phases and by the Dobson-Chato [48] for annular flow, Shah [43], Jung *et al.* [53] and Singh *et al.* [73]. The results of comparison are plotted in a table form, where average deviation and mean absolute deviations are reflected. As follows from this comparison, neither of the correlations agrees with all experimental data. It should be also noted that such comparison does not provide an information on what particular data (G, q, t_s, d) correspond to the maximal divergence of predicted and experimental data.

Conclusions

The review involves the results of theoretical and experimental studies of condensation inside horizontal tubes carried out in the period from 1955 to 2013. The theoretical solutions have been submitted for two phase flow patterns: annular flow, when condensate film is flowing under friction forces, and gravitational (stratified) flow, when gravity dominates among the forces affecting condensate flow. Prediction of heat transfer in a stratified flow on the upper part of tube is recommended in accordance with the Nusselt formula for condensation outside horizontal tube. Heat transfer by vapor condensing in a stream (bottom part of tube) is predicted by different correlations not proved by the experiments, but related to convective heat transfer in tube.

Theoretical solutions for annular turbulent flow of condensate affected by vapor velocity require information on frictional pressure drop $(\Delta P/\Delta z)_f$ and volumetric vapor content ε. To predict $(\Delta P/\Delta z)_f$ and ε a lot of the correlations give high discrepancy between each other and respectively result in wide deviations of the parameters demanded for heat transfer calculation. In the experimental studies the heat-transfer coefficients averaged along the perimeter of pilot section were measured. It does not give the concept of heat transfer behavior along tube perimeter and impedes assessment of actual impact of phases flow on heat transfer. The proposed empirical correlations (over 60) for heat transfer prediction reveal significant discrepancies both in heat-transfer coefficients and in the rate of main parameters ($\mathrm{Re_l}$, $\mathrm{Re_v}$, $\mathrm{Pr_l}$) that effect on condensation. For the liquids like freons and hydrocarbons in the region of $\mathrm{Re_l} > 1.6 \cdot 10^3$, the methods proposed by Shah [10] and Thome *et al.* [56] demonstrate the best agreement with the experiments performed by many authors.

Nomenclature

a – thermal diffusivity, [m^{2}s^{-1}]
C_f – friction coefficient
c_p – liquid specific heat, [Jkg^{-1}K^{-1}]
d – inner diameter of tube, [m]
Fr – Froude number
$\{= [\rho_v(\rho_l - \rho_v)w_v^2]/[\rho_l^2(v_l g)^{2/3}]\}$

G – mass velocity, [kgm^{-2}s^{-1}]
Ga – Galileo number $[= \rho_l(\rho_l - \rho_v)gd^3/\mu_l^2]$
g – gravitational acceleration, [ms^{-2}]
Ja_l – liquid Jakob number $[= \rho_l(\rho_l - \rho_v)gd^3/\mu_l^2]$
J_g – dimensionless vapor velocity $\{= xG/[gd\rho_v(\rho_l - \rho_v)]^{0.5}\}$
l – length of test tube, [m]
Nu – Nuselt number
Pr – Prandtl number
p_r – reduced pressure
q – heat flux, [Wm^{-2}]
r – heat of vaporization, [Jkg^{-1}]
Re_f – film Reynolds number $[= qz/(r\mu_l)]$
Re_l – liquid Reynolds number $[= G(1-x)d/\mu_l]$
Re_{lo} – Reynolds number assuming total mass flowing as a vapor $(= Gd/\mu_l)$
Re_v – vapor Reynolds number $(= Gxd/\mu_v)$
Re_{vo} – Reynolds number assuming total mass flowing as a vapor $(= Gd/\mu_v)$
u – axial velocity, [m^{-2}s^{-1}]
X_{tt} – Martinelli parameter $\{= (\mu_l/\mu_v)^{0.1}(\rho_v/\rho_l)^{0.5}[(1-x)/x]^{0.9}\}$
x – vapor quality
y – radial distance from the wall, [m]
z – axial co-ordinate, [m]

Greek symbols

α – heat transfer coefficient, [Wm^{-2}K^{-1}]
δ – thickness of the condensate film, [m]
ΔP – pressure drop, [Pa]
ΔT – temperature difference $(= t_s - t_w)$, [K]
ε – void fraction
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]

Subscripts and superscripts

eq – equivalent
f – frictional term
l – liquid
m – momentum
s – saturated
v – vapor/gas
w – wall
\+ – non-dimensional symbol

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Paper submitted: May 22, 2014
Paper revised: March 12, 2015
Paper accepted: March 16, 2015