International Journal of Mechanical, Industrial and Aerospace Sciences
ISSN: 2517-9950
Vol:11, No:8, 2017

An Improved Heat Transfer Prediction Model for Film Condensation inside a Tube with Interphacial Shear Effect

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Abstract**—**The analysis of heat transfer design methods in condensing inside plain tubes under existing influence of shear stress is presented in this paper. The existing discrepancy in more than 30-50% between rating heat transfer coefficients and experimental data has been noted. The analysis of existing theoretical and semi-empirical methods of heat transfer prediction is given. The influence of a precise definition concerning boundaries of phase flow (it is especially important in condensing inside horizontal tubes), shear stress (friction coefficient) and heat flux on design of heat transfer is shown. The substantiation of boundary conditions of the values of parameters, influencing accuracy of rated relationships, is given. More correct relationships for heat transfer prediction, which showed good convergence with experiments made by different authors, are substantiated in this work.

Keywords**—**Film condensation, heat transfer, plain tube, shear stress.

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 ($= \dfrac{\rho_v(\rho_l - \rho_v)w_v^2}{\rho_l^2(v_l g)^{2/3}}$)
G	– mass velocity, [kgm^{-2}s^{-1}]
g	– gravitational acceleration, [ms^{-2}]
l	– length of the tube, [m]
l_G	– capillary constant, ($= \left[\sigma/\left(g(\rho_l - \rho_v)\right)\right]^{0.5}$)
Nu	– Nusselt number
Pr	– Prandtl number
q	– heat flux, [W·m^{-2}]
r	– heat of vaporization, [J·kg^{-1}]
Re_f	– film Reynolds number ($= ql/(r\mu_l)$)
Re_l	– liquid Reynolds number ($= G(1-x)d/\mu_l$)
Re_{lo}	– only liquid Reynolds number ($= Gd/\mu_l$)
Re_v	– vapor Reynolds number ($= Gxd/\mu_v$)
t	– temperature, [°C]
w	– velocity, [ms^{-1}]
x	– mass vapor quality
X_{tt}	– Martinelli parameter ($= (\mu_l/\mu_v)^{0.1}(\rho_v/\rho_l)^{0.5}\left[(1-x)/x\right]^{0.9}$)

Greek Symbols

α	– heat transfer coefficient, [Wm^{-2}K^{-1}]
δ	– thickness of the condensate film, [m]
ΔP	– pressure drop, [Pa]
ε	– void fraction
λ	– thermal conductivity, [Wm^{-1}K^{-1}]
μ	– dynamic viscosity, [Pas]
ν	– kinematic viscosity, [m^{2}s^{-1}]
ρ	– density, [kgm^{-3}]
σ	– surface tension, [Nm^{-1}]
τ_w	– shear stress, [Pa]
τ_g	– gravity force, [Pa]
φ	– angular coordinate, [°]
Φ_v^2	– parameter that takes into account influence of two-phase flow on shear stress
Φ_q	– parameter that takes into account surface suction at the interphase

Sub and Superscripts

eq	– equivalent
l	– liquid
v	– vapor
exp	– experimental
$calc$	– calculated
+	– dimensionless symbol

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I. INTRODUCTION

CONDENSATION inside tubes occurs in evaporative systems of thermal desalinating plants, air conditioning systems, safety systems of reactors, heaters of power plants and condensers of cooling equipment. It is very important to have an exact knowledge of condensation heat transfer coefficients when their value is close to heat transfer from the side of cooling.

One hundred years ago Nusselt [1] described the basis of heat transfer design during laminar film condensation. Nusselt gave simple relationships for heat transfer coefficient prediction in condensing on vertical flat surface both when shear stress is absent and when shear stress exists.

Dakler [2], Bae at al. [3], [4], Traviss at al. [5], Kosky and Staub [6], Nouri-Borujerdi [7], Kwon et al. [8] calculated heat transfer coefficients at a turbulent flowing of the condensate film under existing influence of shear stress τ_w. Boyko and

Kruzhilin [9] developed a semi-empirical model for design of heat transfer in condensing of a turbulent mixture of steam and liquid in tubes, and Thome at al. [10] suggested a model for design of heat transfer in condensing inside a horizontal tube when different regimes of phase flow take place (annular, stratified and intermediate).

Numerous works (theses, articles, conference reports) showed the results of experimental investigations in vertical and horizontal tubes with vapor condensing of various liquids; a lot of semi-empirical relationships for design of local heat transfer; a comparison of experimental data to theoretical solutions and to semi-empirical correlations. The analysis of these works showed such features of the investigative results:

1. A difference in the degree of convergence between the various design correlations and experiments. This feature is shown in [11]-[17].
2. There are no remarks concerning boundaries for use of the proposed relationships for heat transfer design in most of the works with the exception of Cavallini et al. [18] and Thome et al. [10], where the authors described their design methods and the conditions when these methods can be used in the smallest details with correction of flow regimes.
3. There is no substantiated selection of relationships for a definition of friction coefficient C_f and accordingly shear stress τ_w in most of the works.
4. The influence of the suction parameter on C_f and τ_w, which is related to the cross flow of mass at the interface was not taken into account.

A comparison of the experimental heat transfer coefficients to existing and improved methods of heat transfer prediction in pure single-component vapor condensing inside the plain tube has been made in this work.

II. Analysis of Theoretical Methods for Heat Transfer Prediction

A. Heat Transfer in Laminar Flowing of Condensate Film under Existing Influence of Shear Stress

In 1916, Nusselt [1] first drew the theoretical solution for heat transfer prediction in condensing pure vapor on the vertical plain surface. This relationship in the coordinates Nu, Fr_l, Re_l takes the form:

$$Nu = 0.5\left(C_f Fr_l / \mathrm{Re}_l\right)^{0.5}, \quad (1)$$

where

$$Nu = \frac{\alpha}{\lambda_l}\left(\frac{v_l^2}{g}\right)^{2/3}.$$

A precise definition of heat transfer coefficient by (1) depends on a precise definition of the friction coefficient C_f. The importance of correct estimation of the influence of Φ_v^2 as well as the suction parameter q/Gxr on C_f has been shown in [19], [20]. In these works, the validity of (1) in the annular flowing of condensate film, when $\tau_w \gg \tau_g$, was proved.

Equation (1) can be used in condensing inside tubes and channels with a definite geometrical characteristic. As was noted in [21], the size of the channels has an influence on two phase flow, when $d_{eq} > 5l_G$. This gives a permit to use (1) in any channels and tubes in condensing vapor of such liquid as steam, all types of Freon's, hydrocarbons, carbon dioxide for heat transfer prediction.

B. Heat Transfer in Turbulent Flowing of Condensate Film

Mathematical methods of heat transfer prediction in turbulent flowing of condensate film are grounded on the solution of motion and energy equations for various turbulent models. Rifert et al. in [19], [20] used the graphs $Nu=f(\beta, \mathrm{Re}_l, \mathrm{Pr}_l)$ from [2]-[5]. These graphs are shown in Figs. 1-3. As well as for a laminar flowing of condensate film, for turbulent flowing of condensate film it is important to determine the friction coefficient C_f correctly. This friction coefficient C_f is contained in parameter β:

$$\beta = 0.5 C_f Fr_l. \quad (2)$$

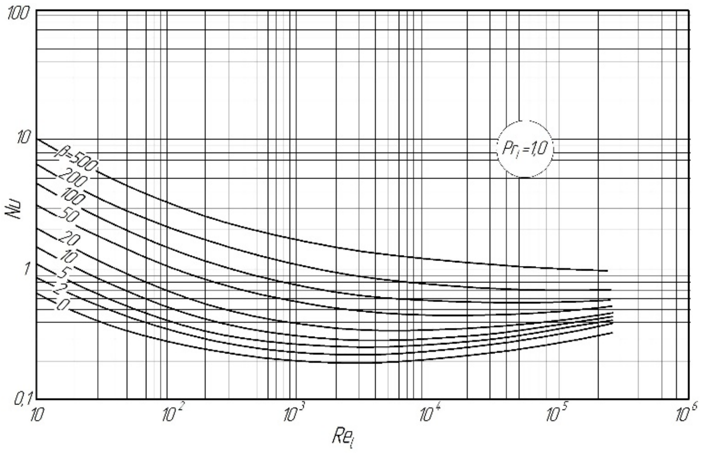

Fig. 1 Dimensionless local heat transfer coefficients (Pr_l=1)

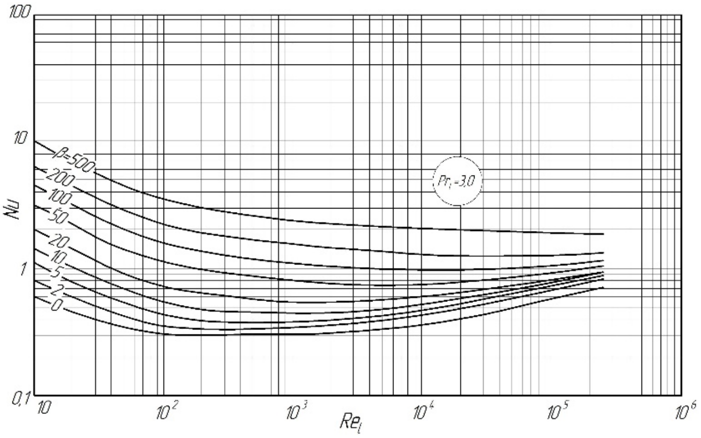

Fig. 2 Dimensionless local heat transfer coefficients (Pr_l=3)

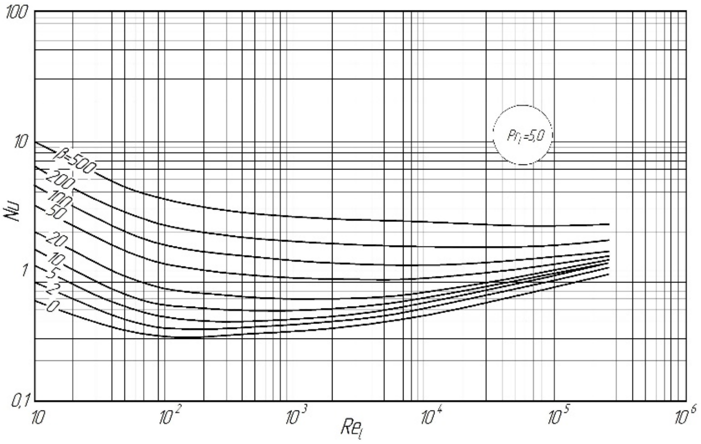

Fig. 3 Dimensionless local heat transfer coefficients (Pr_l=5)

III. Analysis of Semi-Empirical Methods for Heat Transfer Prediction

Nowadays two semi-empirical relationships are often used to compare them with experimental data on heat transfer.

One of them belongs to Boyko and Kruzhilin [9]. The authors deal with a homogeneous model of phase flow in condensing in tubes. The authors are of the opinion that active entrainment of a condensate takes place in most of a tube at a high mass velocity. It is considered the flow of vapor and condensate mix to be turbulent. The authors restricted using their model when $\mathrm{Re}_{lo}>5\cdot10^3$. Accepting the Reynolds analogy, the authors think that heat transfer in mix condensing is completely analogous to convective heat transfer in turbulent liquid flow in a tube i.e. $Nu=\alpha d/\lambda_l=0.023\,\mathrm{Re}_{lo}^{0.8}\,\mathrm{Pr}_l^{0.43}\,d/\lambda_l$. Two phase flow were taken into account by introduction of the following complex:

$$Nu=c\,\mathrm{Re}_{lo}^{0.8}\,\mathrm{Pr}_l^{0.43}\left[1+x(\rho_l/\rho_v-1)\right]^{0.5}. \tag{3}$$

The average heat transfer coefficient is determined by (4) when x varies from x_1 (tube inlet) to x_2 (tube outlet).

$$Nu=c\,\mathrm{Re}_{lo}^{0.8}\,\mathrm{Pr}_l^{0.43}\,\frac{1}{2}\left[\sqrt{1+x_1\left(\frac{\rho_l}{\rho_v}-1\right)}+\sqrt{1+x_2\left(\frac{\rho_l}{\rho_v}-1\right)}\right]^{0.5}. \tag{4}$$

In [22], the experiments on steam condensation in horizontal stainless steel tubes of d=10 mm, 13 mm and 17 mm and l=2.5 m and 12 m was performed. The average temperature of the tube wall was measured by the Marcbant method [23] using the tube surface as a resistance thermometer. The experiments were made at p=1.23 MPa, 2.45 MPa, 8.8 MPa and at the following mass vapor qualities: one cycle – x_1=1, x_2=0; second cycle – 1>x_1>0, x_2=0; third cycle – x_1=1, 1>x_2>0; fourth cycle – 1>x_1>0, 1>x_2>0. All experiments were performed with a change in q from $0.162\cdot10^6$ to $1.57\cdot10^6$ W/m^2, G from 93 to 2000 kg/(m^2·s). All 540 experiments have convergence with (3) at c=0.024 within the limits of ±20%.

Two values of constant c in (4) (for stainless tubes c=0.024 and for copper and brass tubes c=0.032) were shown in [24], [25] and in many other articles. However, there is no correct proof of this fact.

The second known semi-empirical relationship for heat transfer prediction in condensing inside horizontal tube were drawn in [10] by Thome et al.

$$Nu = 0.0039 f_i \,\mathrm{Re}_\delta^{0.7}\,\mathrm{Pr}_l^{0.5}, \quad (5)$$

where

$$Nu = \frac{\alpha\delta}{\lambda_l}, \quad \mathrm{Re}_\delta = \frac{4G(1-x)\delta}{(1-\varepsilon)\mu_l},$$

$$f_i = 1 + (w_v/w_l)^{0.5}\left[(\rho_l - \rho_v) g\delta^2/\sigma\right]^{0.25}.$$

The authors did not prove the introduction of f_i by any experimental data. Equation (5) gives good convergence with some experimental data in annular and intermediate flow.

Equation (5) for annular flow regime can be represented in this form $Nu=f(\beta, \mathrm{Re}_l, \mathrm{Pr}_l)$. The thickness of a condensate film in the annular flow regime is determined by:

$$\delta = d(1-\varepsilon)/4. \quad (6)$$

The thickness of a condensate film is related to the dimensionless thickness by:

$$\delta = \delta^+ \nu_l (\tau_w/\rho_l)^{0.5}. \quad (7)$$

If $\mathrm{Re}_l>1125$ (turbulent flow regime), then dimensionless thickness is calculated from [5] by:

$$\delta^+ = 0.095\,\mathrm{Re}_l^{0.812}. \quad (8)$$

τ_w is determined by known relationship,

$$\tau_w = 0.5 C_f \rho_v w_v^2. \quad (9)$$

Substituting (6)-(9) in (5) the following relationship can be obtained:

$$Nu = c_{new} f_i (0.5 C_f Fr_l)^{0.5}\,\mathrm{Re}_l^{-0.07}\,\mathrm{Pr}_l^{0.5}, \quad (10)$$

where c_{new} is a new constant.

The calculations of function f_i for experimental data of steam from [22], R22, R32, R134a, R236ea, R410a from [26], hydrocarbons from [27] and carbon dioxide from [28] were shown, that f_i changes from 1,5 to 2,0. Taking mean value of f_i=1.75, following relationship can be obtained:

$$Nu = 1.75 c_{new} (0.5 C_f Fr_l)^{0.5}\,\mathrm{Re}_l^{-0.07}\,\mathrm{Pr}_l^{0.5}, \quad (11)$$

associates with the results in [2]-[4].

Equation (11) was used in the experimental data processing on vapor condensation inside a vertical tube [29] and inside a horizontal tube [30]. The demerit of (5) and (11) lies in the constant power of Fr_l, Re_l and Pr_l, regardless of the numerical values of those numbers. This diverges both with the theory (Figs. 1-3) and with the experiments.

IV. Comparison of Theoretical Methods for Heat Transfer Prediction

Rifert et al. [19], [20], [31] conducted experiments for the condensation of steam and Freon R22 inside a horizontal tube of d=17 mm, l=110 mm and measured φ-wise local heat transfer coefficients α_φ by the "thickness wall" method. The obtained data is in the region of the low values of Re_l ($\mathrm{Re}_l<10^3$ i.e. in two regimes: laminar and transitional to turbulent flow of a condensate film). The investigative procedure lets estimate the influence of two-phase flow (parameter $\Phi_v^2 = f(X_{tt})$) as well as a vapor mass suction on interphase (parameter $\Phi_q = q/rGx$) on the friction coefficient C_f and correspondingly on the shear stress τ_w. It was shown that the heat transfer coefficients, which are average over φ, but they are local over l, are calculated by (1) accurately when the friction coefficient C_f is determined by:

$$C_f = C_{fo}\Phi_v^2\Phi_q, \quad (12)$$

where $C_{fo} = \dfrac{0.079}{\mathrm{Re}_v^{0.25}}$ at $\mathrm{Re}_v < 10^5$ or $C_{fo} = \dfrac{0.046}{\mathrm{Re}_v^{0.2}}$ at $\mathrm{Re}_v \geq 10^5$.

Parameter Φ_v^2 is determined by Miyara's equation from [32]:

$$\Phi_v^2 = 1 + C X_{tt}^n + X_{tt}^2, \quad (13)$$

where

$$C = 21\left\{1-\exp\left(1-0.28 Bo^{0.5}\right)\right\}\left\{1-0.9\exp\left(-0.02 Fr_l^{1.5}\right)\right\},$$

$$n = 1 - 0.7\exp(-0.08 Fr_l), \quad Fr_l = Gx/\left[gdG(\rho_l-\rho_v)\right]^{0.5}.$$

Coefficient Φ_q, which takes the influence of a mass suction into account, is calculated from [33] by:

$$\Phi_q = 1 + 17.5\,\mathrm{Re}_v^{0.25}\, j, \quad (14)$$

where the suction parameter $j = q/(rGx)$.

Bae et al. [3], [4] compared their experimental data for condensation of R12 and R22 inside a horizontal tube with the theoretical method of heat transfer prediction for a condensate turbulent film (1) by using the equation of Soliman et al. [34] to determine τ_w. The good agreement between α_{exp} and α_{calc} is observed for the experiments with R22. However, a discrepancy of more than 30% between α_{exp} and α_{calc} was obtained in the experiments with R12.

Traviss at al. [5] drew the calculated data in the graph form $Nu=f(\beta, \mathrm{Re}_l, \mathrm{Pr}_l)$, and in the approximation:

$$\frac{Nu F_2}{\mathrm{Pr}_l\,\mathrm{Re}_l^{0.9}} = F(X_{tt}), \quad (15)$$

where $F(X_{tt}) = 0.15\left(1/X_{tt} + 2.85/X_{tt}^{0.476}\right)$. But, the difference in dependence of α from x for most experiments at x>0.4÷0.6

was obtained in [5]. That is why, Traviss at al. offered another equation for heat transfer prediction when $F(X_{tt}) > 2$:

$$\frac{NuF_2}{\Pr_l \mathrm{Re}_l^{0,9}} = \left[F\left(X_{tt}\right)\right]^{1,15}. \tag{16}$$

Kwon et al. [35], Agra and Teke [36] and others authors used (16) for comparison with their experiments.

Cavallini et al. [26] made a comparison of their experimental date for condensation of R22, R32, R134a, R236ea and R410a inside a horizontal tube of d=8 mm and l=1.0 m with Kosky and Staub's theoretical model [6] by using Friedel's formula [37] to calculate τ_w. As was noted in [26], most experiments for annular and intermediate phase flow are in good agreement with the calculations.

Park et al. [27] drew in tables the results of a comparison of the experiments for condensation of R22, propylene, propane, DME and isobutene inside a horizontal tube of d=8.8 mm and l=0.53 m with the Treviss at al. method (16). There is the discrepancy of 46.9% between the experimental and calculated data for propylene.

Sapali and Patil [38] investigated the condensation of R134 and R404a inside a horizontal tube of d=8.56 mm and l=4.5 m at G=90÷800 kg/(m^2·s). The experimental heat transfer coefficients α_{exp} were lower by 75÷85% than the calculated heat transfer coefficients α_{calc} at x>0.4÷0.6. It should be noted that the values of α_{exp} for R134 in [38] are lower by 30÷40% than in [26]. Also, in [38] the discrepancy in 60÷90% between the experimental data and calculations by (16) were obtained.

Ghim and Lee [39] made a comparison of heat transfer coefficients in condensing R245fa, NOVEC649 and HFE7000 inside a horizontal tube of d=7.75mm and l=0.33m at G=150÷700 kg/(m^2·s) with calculations by (16). The experimental data are lower by 25÷40% than the calculated data for all x and G.

Macdonald [16] made a comparison of his experimental data on condensation of propane with calculations by (16) and with Jaster and Kosky's method [40]. The discrepancy in more than 30÷50% was obtained.

In [41], the experimental data on condensation of FC-72 were compared with 12 relationships, including numerous solutions of Kosky and Staub [6]. The local values of heat transfer coefficients α_{exp} at G=144÷402 kg/(m^2·s) are higher than the calculated values α_{calc} along the full length of the tube.

V. Comparison of Semi-Empirical Methods for Heat Transfer Prediction

Boyko and Kruzhilin's model (3), as shown below, is one of most successful relationships for a generalization of experimental data on vapor condensation inside horizontal and vertical tubes in a wide range of G, x and refrigerant physical properties. The experimental data and average heat transfer coefficients from [22] have a good (less than 15%) convergence with calculations by (3) and a(4). But there is a small number of works up to now, where this model (3) has been used for making comparison with experiments.

In dissertation [42] the graphs were drawn, on which Royal made a comparison of local heat transfer coefficients in steam condensing inside a horizontal tube of d=13,8 mm and l=3.5 m with the model (3). The experiments were made at the values of vapor pressure, which were close to those in [22]. However, the values of the local heat transfer coefficients α_{exp} are lower by 30÷80% than the calculated values α_{calc} by formula (3).

As shown in [38], the calculations by (3) showed a discrepancy of 25% and less for condensation of R134a and from 35% to 50% for condensation of R404a.

The method of Thome at al. (5) is often used by researchers for comparison with different experiments. In contrast to the Boyko and Kruzhilin's model (3), the method (5), as well as methods of Cavallini at al. [18], [26], are used for the annular and intermediate flow of the phases (under existing influence of shear stress) and for the stratified phase flow.

As shown in [28], there is the good convergence (~20%) between the experiments on condensation of carbon dioxide inside the horizontal tube of d=3,42 mm and l=3.5 m at G=200÷800 kg/(m^2·s) with calculations by (5). It was also shown in [43] that there is the discrepancy in 172% between the experimental data α_{exp} on condensation of carbon dioxide inside the horizontal tube of d=5,15 mm at G=600÷1000 kg/(m^2·s) with the calculated data α_{calc} by (5).

In [10] the experimental data of Cavallini et al. [18] are compared with the method (5) for two cases: first, without taking into account the influence of waves–parameter f_i in (5); secondly, when the parameter f_i is taken into account. According to Figs. 6 and 7 from [10], the account of f_i has an influence on heat transfer only for R32. And this influence is slight.

The discrepancy in more than 100% between the experimental data α_{exp} and the calculated data α_{calc} by (5) was shown in Fig. 29 from dissertation [16].

VI. The Improvement of Design Methods

A. The Substantiation of Improvement of Design Methods

The analysis of the works, where the experimental data on condensation inside tubes were compared with different design methods, showed great difference among the results of different authors in the case of using the same relationships for heat transfer prediction for all kinds of fluids. The discrepancy in the same design methods for the same refrigerants is 100% and more.

When employing theoretical solutions, the main reason for attaining different results can be first of all the use of different methods for prediction of the friction coefficient C_f. Then, it is very important to know the range of application for one or another relationship in order to compare them with theoretical solutions as well as with experimental correlations. Many semi-empirical relationships were obtained over a short range of the changes of G, x and physical properties. For this reason, semi-empirical relationships can only be used in conditions which are close to experimental ones.

B. The Substantiation of the Range of Application for Boyko's Relationship

The convergence of experimental heat transfer coefficients α_{exp} with the calculated data α_{calc} on (3) within the limits of ±20% for all experiments from [22] at G=80÷1700 kg/(m^2·s) and x=0.2÷0.83 were shown in [19], [20]. It is necessary to note that a multiplier $\mathrm{Re}_{lo}^{0.8}\left[1+x(\rho_l/\rho_v-1)\right]^{0.5}$ in (3) is more than 10^4, $\mathrm{Re}_l \geq 3000$, $\mathrm{Fr}_l \geq 10^4$, when the lowest value of Re_v is close to 7000. It means, that in all experiments [22], the turbulent film flow of vapor and liquid exists almost along the full length of the tube, which corresponds to the Boyko's model (3).

In Fig. 4, it is drawn the comparisons of all experimental data α_{exp} from [22] at $x=(x_1+x_2)/2$ with calculations by (3). It is seen that this formula is in the good agreement with experiment.

In Table I, it is drawn the date on steam condensation inside the vertical tube at p=361 kPa from [44]. In this work Treputnev has investigated steam condensation inside an experimental section of l=120 mm and d=18 mm. This section consists of two copper thick-walled cylinders, which are intended to measure the heat transfer coefficients and pressure drop ΔP. In the experiment, steam came to the vapor cooler for cooling, further it came to the experimental section, after that it came into the condenser and the condensate was drained into the drainage through measuring tanks. The vapor cooler is designed to decrease the mass vapor quality of working fluid to the given value. The heat exchange surface of the experimental section was divided into the parts to the length, in such a way that the mass vapor quality changes Δx of working fluid were low. So the measured values of q, α and ΔP can be considered to be quasilocal.

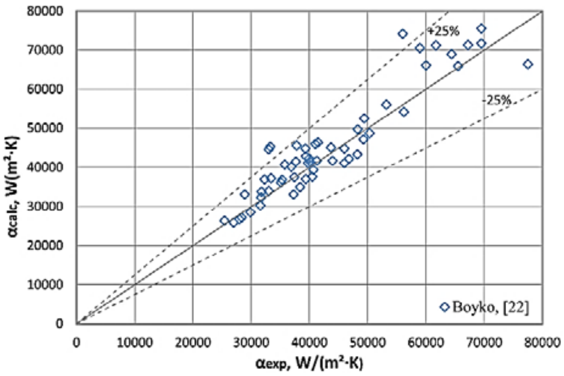

Fig. 4 Application of the model (3) to Boyko [22] data

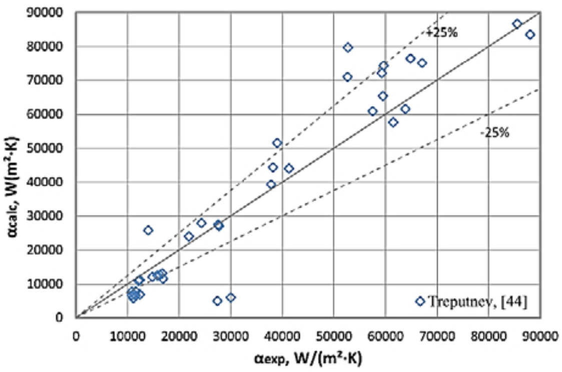

Fig. 5 Application of the model (3) to Treputnev [44] data

TABLE I
EXPERIMENTAL DATA ON STEAM CONDENSATION INSIDE A VERTICAL TUBE FROM [44]

№	x	G, kg/(m^2·s)	q, kW/m^2	α_{exp}, W/(m^2·K)	Re_l	α_{cacl}, W/(m^2·K)	№	x	G, kg/(m^2·s)	q, kW/m^2	α_{exp}, W/(m^2·K)	Re_l	α_{cacl}, W/(m^2·K)
1	0.5	209	1198	37800	18716	39306	23	0.70	318	782	59500	28478	65351
2	0.51	23	707	11300	2087	6897	24	0.71	22	642	11600	1979	7796
3	0.51	27	629	10800	2418	7760	25	0.71	40	576	15800	3573	12507
4	0.53	19	682	11100	1657	5846	26	0.72	40	587	16300	3573	12629
5	0.54	21	605	11600	1890	6555	27	0.73	13	963	27400	1128	5043
6	0.55	40	442	12100	3573	10973	28	0.73	105	844	27600	9403	27501
7	0.56	405	1119	52600	36269	70955	29	0.74	373	1221	64900	33403	76383
8	0.58	21	657	10800	1872	6741	30	0.75	101	878	27700	9045	27021
9	0.59	39	455	12400	3510	11216	31	0.74	373	1221	64800	33403	76383
10	0.60	43	558	14800	3806	12096	32	0.80	39	611	16700	3510	13087
11	0.62	98	729	21900	8776	23990	33	0.81	15	591	30000	1316	6008
12	0.62	98	729	21900	8776	23990	34	0.84	98	715	24300	8776	27911
13	0.62	407	1227	67100	36448	75061	35	0.86	32	632	16900	2866	11535
14	0.63	252	1025	39000	22567	51519	36	0.86	358	1001	52700	32060	79614
15	0.64	206	1130	41300	18448	44089	38	0.90	248	1150	57500	22209	60815
16	0.64	396	1209	59600	35463	74253	39	0.97	221	1317	61500	19791	57592
17	0.64	206	1130	41300	18448	44089	40	0.97	370	1279	85500	33134	86577
18	0.65	375	1163	59300	33582	72026	41	0.97	221	1317	61500	19791	57592
19	0.66	20	564	12400	1791	6941	42	0.97	370	1279	85500	33134	86577
20	0.69	100	463	14000	8955	25715	43	0.98	239	1291	63800	21403	61503
21	0.70	19	700	12400	1719	6918	44	0.99	348	1320	88000	31164	83492
22	0.70	197	1032	38200	17642	44395	45	0.99	348	1268	88000	31164	83492

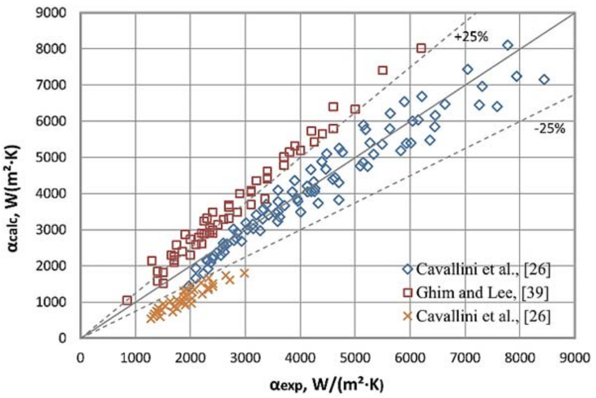

Fig. 6 Application of the model (3) to Cavallini et al. [26], Ghim and Lee [39] data

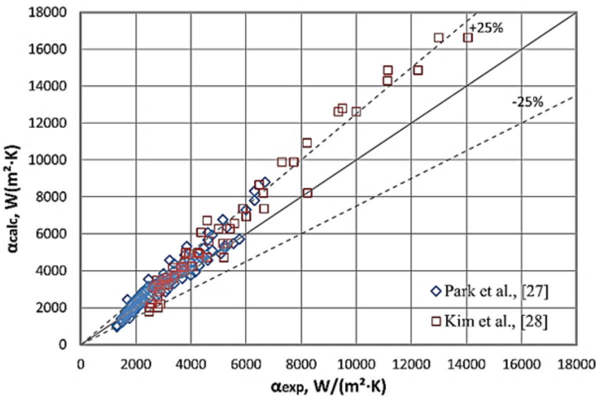

Fig. 7 Application of the model (3) to Park et al. [27] and Kim et al. [28] data

In Fig. 5, it is shown the comparison of the experiments with the calculation by (3). The good convergence takes place at $Fr_l > 10^4$ ($\beta > 5$) when the influence of shear stress on heat transfer exists. It is necessary to pay special attention to convergence of the experiments inside the copper tube with calculation by (3), where the coefficient c is equal 0.024. It proves that heat transfer is independent of wall material.

The discrepancy in experiments is observed when $\mathrm{Re}_l < 10^3$, i.e. in the region where Nu decreases along with the growth of Re_l (Figs. 1-3). Thus, Boyko's model (3) does not work in this region.

In Fig. 6, it is shown the comparison of Cavallini et al. [26] data on condensation of R134a, R125, R32, R410A, R236ea with calculation by (3). The experimental data, marked by the sign "×" are less than the calculated data by 25%. These data correspond with R134a at G=65÷100 kg/(m^2·s) and x=0.29÷0.71; R236ea at G=100 kg/(m^2·s) and x<0.5; R125 at G=100÷200 kg/(m^2·s) and x=0.25÷0.66; R32 at G=100 kg/(m^2·s) and x=0.34÷0.69; R410a at G=100 kg/(m^2·s) and x=0.32÷0.7.

In Fig 6, it is also drawn the data on fluids R245fa, NOVEC649 and HFE-7000 from [39].

The analysis of all experiments showed that the discrepancy between the experimental data α_{exp} and calculated α_{calc} is more than 25% when the multiplier $\mathrm{Re}_{lo}^{0.8}\left[1+x(\rho_l/\rho_v-1)\right]^{0.5} < 5000$ and parameter $\beta > 5$. This data mainly corresponds to the regime of phase flow which is close to stratified one.

In Fig. 7, it is shown the comparison of the experimental data on condensation of R22, propylene, propane, DME and isobutene form [27] and on condensation of carbon dioxide from [28]. The analysis showed that the discrepancy is observed when the multiplier $\mathrm{Re}_{lo}^{0.8}\left[1+x(\rho_l/\rho_v-1)\right]^{0.5} < 4000$.

C. Improved Semi-Empirical Relationships

The theory of film condensation under existing influence of vapor velocity shows that Nu is a function of three parameters β, Re_l, Pr_l (Figs. 1–3). As seen in Figs. 1–3, the theory predicts different influence of power of β, Re_l, Pr_l, depending on values of β, Re_l, Pr_l. For example, in the region of laminar and laminar-wave film flow, the influence of Pr_l decreases along with the decreases Re_l till full absence.

The accuracy of the calculation of Nu in the region of the influence β depends on the accuracy of the friction coefficient C_f determination. There is no substantial procedure to calculate C_f up to now. So, in [16] it was reviewed more than 20 formulas for calculation of ΔP and C_f, respectively. None of them has enough accuracy (within ±40%), when they are compared with experiments.

Isachenko [25] made the measurements of local heat transfer coefficients in steam condensing at p=0.1 MPa. The author represented the correlations which contain numbers Fr_l, Re_l, Pr_l, not including parameter β.

Rifert et al. [19], [20], [31] generalized the experimental data on local heat transfer coefficients α_φ in such form as $Nu = f\left(C_f, Fr_l, \mathrm{Re}_f, \mathrm{Pr}_l\right)$. In [19], [20], [31], the experiments were made at $\mathrm{Re}_f < 200$ ($\mathrm{Re}_l < 800$), for this reason, as in Isachenko's [25] formula, the power of Re_f is negative that corresponds with graphs in Figs. 1–3. The analysis of Figs. 1–3 shows that the influence of Re_l on Nu cannot be taken into account when β>10 (when the influence of vapor velocity on heat transfer exists), Pr_l=1÷3 and Re_l changes from $8\cdot10^2$ to $2\cdot10^4$.

The boundary of changes Re_l from $8\cdot10^2$ to $2\cdot10^4$ are typical for all Freons (R22, R134a, R125, R32, R410a), hydrocarbon refrigerants (dimethyl ether, propylene, propane, isobutene) and carbon dioxide at G=200÷800 kg/(m^2·s) and $\beta\geq5$. The slight decrease of Nu along with the growth of Re_l takes place only at high vapor velocity (β>100). These features of theoretical calculations became the basis for generalization by correlations Nu=f(Fr_l, Pr_l) of experimental data taken from many works on condensation inside horizontal and vertical tubes.

The absence of C_f in these correlations is explained in such way. The value of Fr_l increases in proportion to w_v^2 . At the same time, C_f decreases in proportion to $w_v^{0.2\div0.25}$ depending on Re_l. For this reason, the influence of Fr_l (vapor velocity) power must also include the influence of C_f.

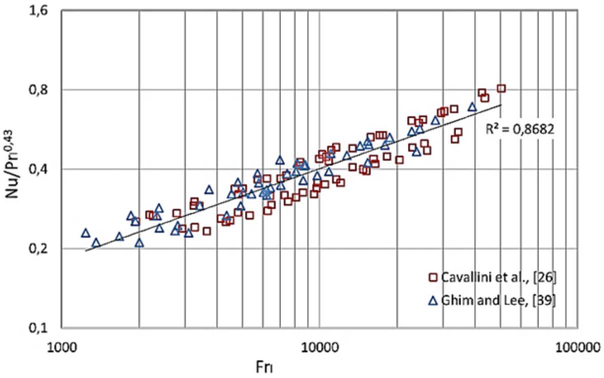

Fig. 8 Approximation of Cavallini et al. [26], Ghim and Lee [39] experimental data in the form $Nu = 0.017 Fr_l^{0.343}\,\mathrm{Pr}_l^{0.43}$

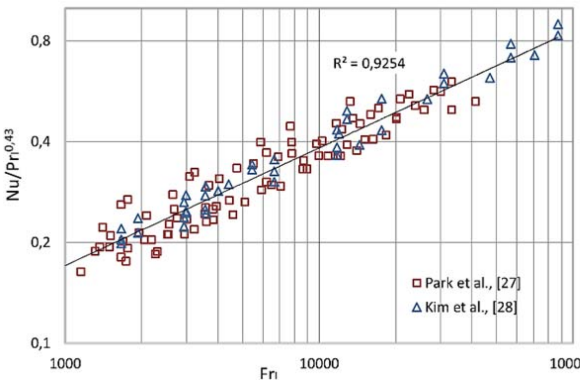

Fig. 9 Approximation of Park et al. [27] and Kim et al. [28] experimental data in the form $Nu = 0.0151 Fr_l^{0.351}\,\mathrm{Pr}_l^{0.43}$

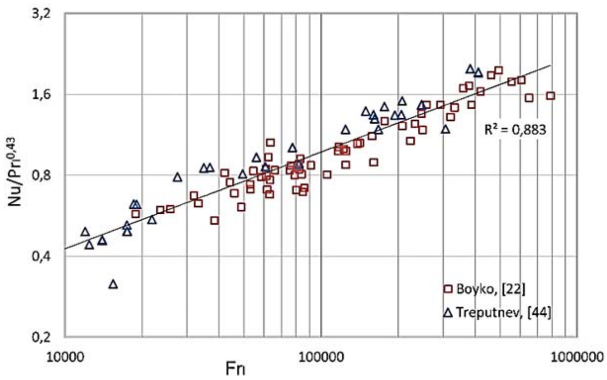

Fig. 10 Approximation of Boyko [22] and Treputnev [44] experimental data in the form $Nu = 0.0156 Fr_l^{0.359}\,\mathrm{Pr}_l^{0.43}$

In this work, the following data were generalized: Cavallini et al. [26] data on condensation of R134a, R125, R32, R410A, R236ea and Ghim and Lee [39] data on condensation of R245fa, NOVEC649 and HFE-7000 (Fig. 8); Park et al. [27] data on condensation of propylene, propane, DME and isobutene and Kim et al. [28] data on condensation of carbon dioxide (Fig. 9); data on steam condensation inside horizontal [22] and vertical [44] tubes (Fig. 10). Of all experiments of the authors, mentioned above, the experimental data at such values of G and x, when according to [10] the annular or intermediate regime takes place, were chosen for generalization. As usual it occurs at β>5, when $Fr_l > 500$.

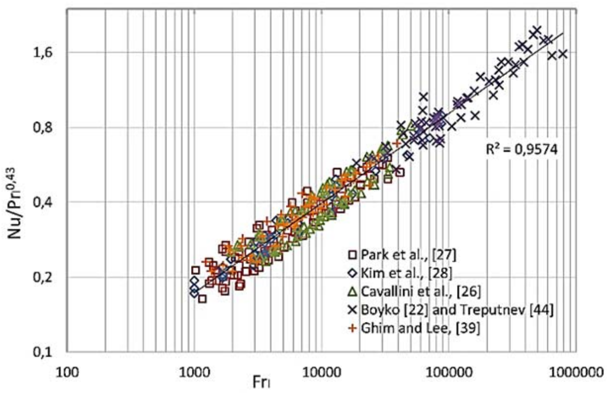

Fig. 11 Approximation of the experimental data of different authors in the form $Nu = 0.0144 Fr_l^{0.36}\,\mathrm{Pr}_l^{0.43}$

In Figs. 8-10 the data of different authors are generalized by the following relationships.

1. Cavallini et al. [26] data on condensation of R134a, R125, R32, R410A, R236ea and Ghim and Lee [39] data on condensation of R245fa, NOVEC649 and HFE-7000 are generalized by the formula:

$$Nu = 0.017 Fr_l^{0.343}\,\mathrm{Pr}_l^{0.43}. \tag{17}$$

The approximation adequacy is $R^2 = 0.8682$.

2. Park et al. [27] data on condensation of propylene, propane, DME and isobutene and Kim et al. [28] data on condensation of carbon dioxide are generalized by the formula:

$$Nu = 0.0151 Fr_l^{0.351}\,\mathrm{Pr}_l^{0.43}. \tag{18}$$

The approximation adequacy is $R^2 = 0.9254$.

3. The data on steam condensation inside horizontal [22] and vertical [44] tubes are generalized by the formula:

$$Nu = 0.0156 Fr_l^{0.359}\,\mathrm{Pr}_l^{0.43}. \tag{19}$$

The approximation adequacy is $R^2 = 0.883$.

As seen in (17)-(19), the difference between the exponent values at Fr_l for all kinds of fluids is slight.

In Fig. 11 the data of authors mentioned above are drawn. These data are generalized by the correlation:

$$Nu = 0.0144 Fr_l^{0.36}\,\mathrm{Pr}_l^{0.43}. \tag{20}$$

The approximation adequacy is $R^2 = 0.9574$.

Equation (20) generalizes all experimental data with the error less than 25%.

VII. Conclusion

The analysis of existing theoretical and semi-empirical design methods for heat transfer prediction in condensing inside tubes under existing influence of vapor velocity was made and the improved design method was suggested.

The results of the investigation are the following:

1. The accuracy of heat transfer prediction in theoretical solutions depends on the knowledge of shear stress (or friction coefficient C_f) for which there are no correct relationships up to now, especially for intermediate phase flow regime.
2. The accuracy of heat transfer prediction on semi-empirical relationships depends on their correct use, first of all the knowledge of the range of their application.
3. It was shown the good convergence of experimental data on condensation of steam, propane, isobutene, propylene, DME, carbon dioxide, R22, R134a, R125, R32, R410A, R245fa, NOVEC649 and HFE-7000 with the calculation by Boyko's formula (3), when the multiplier $\mathrm{Re}_{lo}^{0.8}\left[1+x(\rho_l/\rho_v-1)\right]^{0.5}>4000$ and $\mathrm{Re}_l>800$.
4. The new correlation for heat transfer prediction was obtained. It is based on the theoretical model of turbulent condensation and it is drawn in the form $Nu=0.0144Fr_l^{0.36}\,\mathrm{Pr}_l^{0.43}$. This equation generalizes the large quantity of the experimental data on condensation inside horizontal and vertical tubes of different fluids.
5. The calculation by the suggested formula does not need a correct estimation of phase flow regimes, condensate accumulation and a friction coefficient C_f.
6. It should be taken into account two restrictions while using the suggested relationship: first, Re_l must be more then 800; secondly, β must be more then 5.

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