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Restoration of correctness and improvement of a model for film condensation inside tubes

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This work is devoted to the experimental studies aimed at increasing the efficiency of horizontal tube condensers by strictly accurate evaluation of heat transfer and regime parameters in various condensing refrigerants in the horizontal tubes of such devices. The unique measurements of heat fluxes and heat transfer coefficients local by circumference were carried out during condensation of Freon R-22 and steam which varies over a wide range of the main regime parameters (G, x, q, Re_l). The improved model of film condensation inside the horizontal tubes for prediction of heat transfer with application of the results of numerical solutions of Bae *et al.* is suggested. In this model more precise definition of the interphase friction coefficient as the main parameter crucial for condensation is given. This more precise definition contains experimental substantiation of β_q – prediction for calculation of pressure losses by friction and correction β_q that takes into account surface suction at the interphase. Heat exchange predicted by the improved method was compared with the experimental data of various authors for 13 fluids (steam, R-22, R-123, R-134a, R-245fa, carbon dioxide, propylene, propane, ether, isobutene, refrigerants FC-72, Novec®649, HFE-7000) in annular and intermediate modes. Good agreement of the experiments with calculations (divergence within 25%) proves the correctness of the proposed method for both laminar flow of condensate film and turbulent flow.

Keywords: Film condensation, Heat transfer, Plain tube, Friction coefficient

INTRODUCTION

In modern air conditioning systems, refrigeration and heat pump plants, in the technology of seawater evaporation and power systems heaters, the process of vapour condensation is carried out mainly inside the horizontal tubes and channels. Heat exchange processes occurring in condensers of this type have a significant effect on the overall energy efficiency of these systems. The difference in temperature between the condensing and the cooling fluids and the loss of pressure of these fluids affects the rate of entropy production in the condenser, and hence, the exergy efficiency factor (EEF) of the apparatus.

Currently, the results obtained by the available methods and models for calculating heat transfer for condensation of two-phase flows in horizontal tubes differ by 50-70%. This inaccuracy is due to the presence of a large number of parameters that affect the heat exchange, the wide range of changes in these parameters and the lack of understanding of their influence on the laws of heat exchange. For example, geometrical sizes (length and diameter of tubes), thermophysical properties (heat conductivity, density, surface tension and other) of condensing fluids and operating parameters (pressure, flow, heat fluxes), vary 10-100 times in different heat exchangers. Inaccurate estimation of heat transfer can lead to the unjustified change in the size of the apparatus and pressure differences, which either increase or decrease, resulting in an efficiency decrease. Also, the lack of accuracy of heat transfer calculation leads to the inaccurate evaluation of the effectiveness of various methods of intensifying the heat transfer process during condensation in horizontal tubes.

In view of this, it is urgent to carry out new studies on the influence of regime parameters of the two-phase current on the regularities of local and average heat transfer during the film condensation of moving vapour in a horizontal tube. These studies will open up a possibility of developing a new method for calculating heat transfer during condensation of various refrigerants in horizontal tubes of heat exchangers. More precisely, the estimation of heat exchange and regime parameters will make it possible to increase the efficiency of the work of horizontal tube condensers.

Literature review

Nusselt [1] described the basis of heat transfer during laminar film condensation on a flat vertical surface. The Nusselt theory is used in many studies on condensation within vertical and horizontal tubes. Dukler [2], Bae *et al*. [3] and Traviss *et al.* [4] developed a model of film condensation inside tubes for laminar and turbulent flow of condensate.

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The results of the theoretical solutions in [2-4] are presented in the form of graphs $Nu_f=f(\beta, Re_l, Pr_l)$. Rifert and Sereda [5,6] analyzed more than 20 theoretical methods and correlations for heat transfer prediction during different modes of condensate film flow. In [5,6] it is shown that all theoretical solutions give results close to the theory of Bae *et al*. [3], despite the use of different models of turbulence. Rifert and Sereda [5,6] recommended the method of Bae *et al.* [3] as the most correct theoretical method for heat transfer prediction in annular flow of the phases.

Different authors have proposed a large number (over 60) of empirical dependencies for calculating heat transfer when condensing in tubes. Reviews of such dependences are given in many papers, in particular in [6-10]. In [10] Rifert *et al*. compared the most commonly used formulae of Thome *et al.* [11] and Shah [12] with the formula of Ananiev *et al.* [13]. As shown in [10], the model of Ananiev *et al.* [13] 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.

In the experimental studies of condensation inside the tubes, beginning with the work of Tepe and Mueller [14] up to present time, the coefficients of heat transfer are averaged over a certain length of the tube. Providing in the experiments relatively small changes in the vapour content Δx, the authors assume that they measure local heat transfer coefficients. This is true for a vertical tube, but not always suitable for condensation inside a horizontal one. In most of the experimental work there is no comparison with theoretical calculations [1-4]. Comparison of experimental data of different authors with different empirical dependencies was carried out by Rifert *et al*. in [6,10]. The authors [6,10] showed that over 60 empirical correlations for heat transfer prediction reveal significant discrepancies with experimental heat-transfer coefficients. The best convergence with the experiments of different authors have the correlations by Thome *et al.* [11], Shah [12] and Cavallini *et al.* [15]. In these correlations all included complexes are selected by intuition, without any theoretical or experimental substantiation.

Experimental coefficients of heat transfer are compared with the theory of film condensation only in a small number of the considered works. In these works, there is a large discrepancy between experimental data and theory. The reasons for such discrepancies are due to the incorrect determination of the boundaries of the film flow regimes and the inaccurate estimation of the tangential stress (friction coefficient). In this regard, various authors have proposed dozens of different models and dependencies for calculating heat transfer during condensation. In these cases, there is often no justification for new dependencies, based on the physical nature of the condensation process. Also, there is a disagreement between different dependencies with different experiments, and lack of limits of the suggested methods application.

Based on the nature of the condensation process, and new experimental data of the local heat transfer coefficients, the methods for determining the tangential stress on the vapour and film boundary are substantiated. That made it possible to improve the film condensation model and calculate the heat transfer inside the tubes under the effect of vapour velocity in the laminar and turbulent regime of the condensate film flow.

Test facility

The detailed description of the experimental apparatus and method of heat transfer investigation during film condensation of moving vapour inside horizontal tube can be found in Rifert's previous publication [16]. The apparatus included evaporator, test section, condenser to condense vapor downstream, inspection sections, pressure and temperature gauges at the inlet and outlet of all condensation sections, and condensate flow meters. The test section (Fig. 1) consists of two defining sections 1 and 2, two working sections 3 and 4, as well as sites for visual observation 5, and is designed to study the local coefficients of heat transfer over the length and perimeter of the normal cross-section of the horizontal tube during the condensation of different refrigerants in the tubes by the thick-walled method.

Defining sections 1 and 3 are intended to change the vapour content in the working sections, which allows modelling the process of condensation in tubes of different lengths at different massive vapour velocities. Both defining sections are made in the form of a heat exchanger "pipe in a pipe" with the length of l = 0.8 m. All sections are located on the same axis and have the same internal diameter of 17 mm.

The basis of working sections 2 and 4 shown in Fig. 2 is a segment of thick-walled tube with internal and external diameter of 17 and 80 mm respectively. The material used for the sections is brass brand LS-59. The section length is 96 mm. On the outer perimeter, the sections are surrounded by a ring cavity, in which water is supplied for cooling.

Changing the flow of cooling water allows getting different values of heat flows in the work areas. The condensation temperature at the input to the first defining section was measured using a thermocouple. The obtained value was controlled by a thermocouple installed in the second working section.

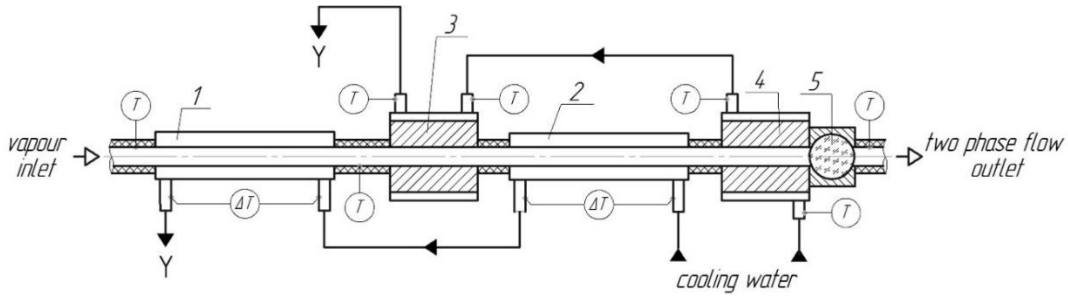

Fig. 1. Schematic view of the tested section: *1*, *2* – defining sections; *3*, *4* – brass working sections; *5* – inspection section; *T* – thermocouple

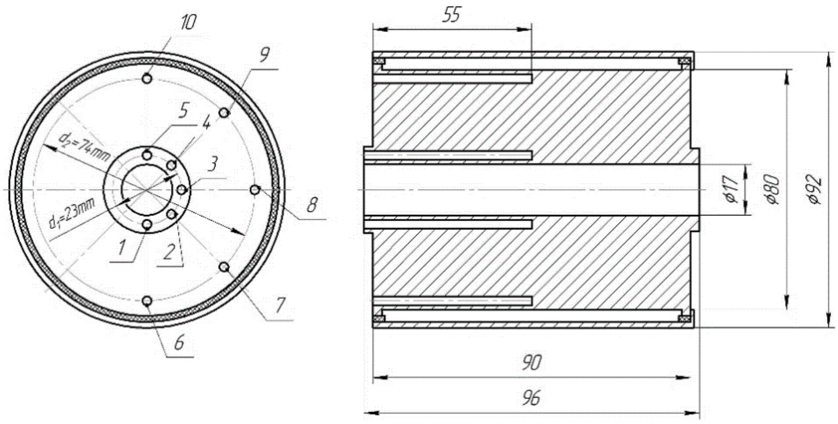

Fig. 2. Drawing of the brass working section: *1*…*10* – channels for locating thermocouples

To determine the temperature distribution in the wall of the work sections, chromel-collier thermocouples were used, which were placed inside the wall of the working sections at the diameters d_1 = 23 mm i d_2 = 74 mm in the axial channels with angular coordinates φ of 0, 45, 90, 135, 180 ° (Fig. 2).

The greater part of the experiments was carried out at local temperature differences in the wall and between the wall and the vapour above 8 ºC and 4 ºC, respectively. In this case, the temperature gradient in the wall in axial direction was much lower than that in adial direction. Such results indicate that the two-dimensionality does not affect the temperature distribution in the wall of the working sections. To calculate the local values of the coefficients of heat transfer α_φ local linear heat fluxes q_l, specific heat fluxes, assigned to the inner surface of the tube q_φ, and the temperature on the inner wall of the tube t_w were determined based on the dependencies:

$$q_l = \frac{2\lambda_b \pi\left(t_i - t_j\right)}{\ln\left(d_2/d_1\right)}; \quad q_\varphi = \frac{q_l}{\pi d};$$

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

where λ_b – coefficient of thermal conductivity of the working section; *i*, *j* – numbers of thermocouples on the diameters d_1 and d_2 respectively (Fig. 2).

Average by the perimeter, but local by length of the tube, the values of thermal flows $\overline{q_\varphi}$ and heat transfer coefficients $\overline{\alpha_\varphi}$ were determined by the formulae:

$$\overline{q}_\varphi = \int_0^\varphi q_\varphi d\varphi, \quad \overline{\alpha}_\varphi = \int_0^\varphi \alpha_\varphi d\varphi.$$

The basic parameters of the two-phase current in all sections of the experimental area and in the condenser were determined by solving the equations of the material and thermal balance recorded for each section in which steam condensation took place, as well as for the capacitor. The range of change of these parameters is shown in table 1.

Table 1. Main operating conditions during condensation tests.

Fluid	t_s, °C	G, kg/(m^2·s)	Local vapor quality x	ΔT, °C	q, kW/m^2
Steam	100	9–54	0.98–0.4	8–22	40–320
R-22	40	11–300	0.24–0.99	4–10	5–50

To prove the accuracy of measuring the local α_φ, the study was carried out for turbulent fluid flow inside a smooth tube provided that hydrodynamic and thermal stabilization were carried out in the adjacent layer. Under these conditions, the temperature of the walls of the working sections, the temperature fluctuations and local heat fluxes almost did not change along the perimeter of the tube, as it should be in the case of fluid flow through a complete section inside the tube.

The control and measuring equipment and the applied method of conducting experiments allowed determining the coefficients of heat transfer with a mean square error of ± 8%. The maximum imbalance of the material and thermal balance of the experimental setup for all conducted studies was 3 and 7%, respectively.

Substantiation of the choice of theoretical dependencies

For annular phase flow by Bae *et al.* [3] the, results of the calculations are represented in dimensionless terms as:

$$Nu_f = f\left(\beta, \mathrm{Re}_l, \mathrm{Pr}_l\right), \quad (1)$$

where $Nu_f = \alpha\left(v_l^2/g\right)^{1/3}/\lambda_l$, $\beta = C_f Fr_l/2$ and $\mathrm{Re}_l = 4\mathrm{Re}_f$ are given in [3].

These correlations are plotted in [3] for numbers Pr_l from 1 to 5. In Figs. 3 and 4 such diagrams are plotted for Pr_l = 1 and Pr_l = 5, respectively. The analysis of correlation (1) makes it possible to note the following features of heat exchange that could be laid down in the improved calculation method. In the region of low values of Re_l heat transfer decreases with increasing Re_l, while the degree of β and Re_l impact corresponds to Nusselt's theory of laminar film condensation [1].

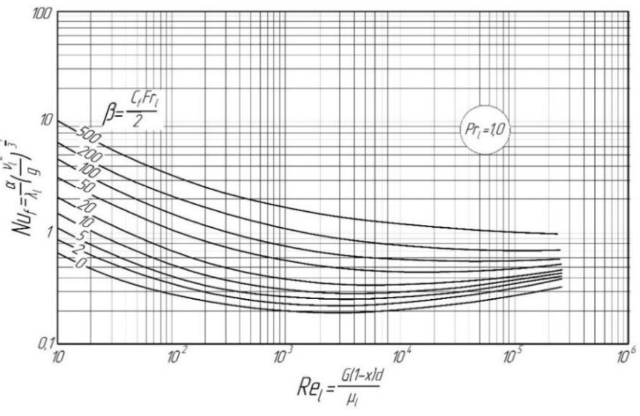

Fig. 3. Dimensionless local heat transfer coefficients from Bae *et al.* [3] at Pr_l=1.

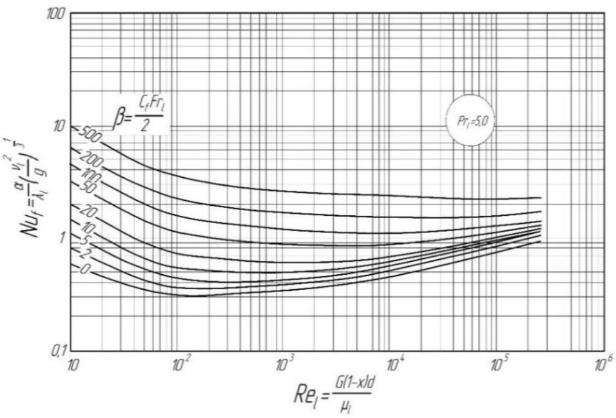

Fig. 4. Dimensionless local heat transfer coefficients from Bae *et al.* [3] at Pr_l=5.

As Re_l increases the heat transfer (Nu) decreases at the beginning and then depending on Pr_l value a wide (at low Pr_l numbers) or a narrow (at high Pr_l numbers) region of independence upon Re_l followed by increasing Nu is observed. In the region close to laminar flow of condensate film (Re_l < 100÷200) the effect of Pr_l is negligible and in some regimes (Re_l < 100 and β > 50÷100) is generally absent. It is true for a laminar flow. At turbulent condensate flow in accordance with the theory the heat transfer intensifies, as Pr_l grows.

The results of calculations by the formula (1) correctly reflect the nature of the condensation flow of the moving vapour inside tubes and channels, only when the influence of gravity on the film of condensation can be neglected. For horizontal tubes, these conditions correspond only to the annular mode of phase flow.

The conducted experimental studies showed that even minor asymmetry of the condensate flow in the upper part of the tube leads to a change in the wave and turbulent characteristics of the film and affects the distribution of local heat transfer coefficients, therefore the accuracy of the obtained results depends on the correct evaluation of the area with the annular mode of phase flow. The method of Rifert *et al*. [17] was used to determine the boundary conditions and the boundary value of the correlation of friction forces τ_f and the gravitational forces τ_f based on the obtained experimental data was corrected. The following criteria for determining the limits of flow regimes were obtained:

at $\tau_f/\tau_g > 10$ – annular flow; (2)

at $1 \le \tau_f/\tau_g \le 10$ – intermediate flow; (3)

at $\tau_f/\tau_g < 1$ – stratified flow, (4)

where $\tau_f = C_f \rho_v w_v^2/2$; $\tau_g = \rho_l g \delta$.

The value of the film thickness δ was calculated by the equation:

$$\delta^+ = \delta/\nu_l \left(\tau_f/\rho_l\right)^{0.5}, \quad (5)$$

where the dimensionless thickness of the film δ^+ depends on the value of Re_l number:

$$Re_l < 50, \quad \delta^+ = 0,7071 Re_l^{0,5}; \quad (6)$$

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$$50 < Re_l \le 1125, \quad \delta^+ = 0,4818 Re_l^{0,585}; \quad (7)$$

$$Re_l > 1125, \quad \delta^+ = 0,095 Re_l^{0,812}. \quad (8)$$

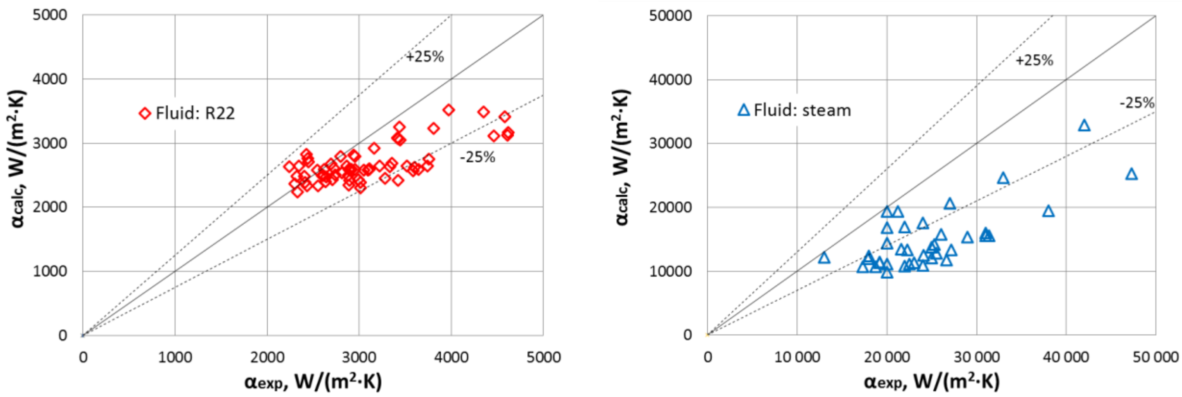
To compare the experimental values of the mean by the φ coefficients of heat transfer obtained in this paper on the basis of the theory of Bae *et al*. [3], sample points were selected which correspond only to the annular or intermediate phase flow modes according to (2)-(4).

The value of the theoretical number Nu_o was calculated on the basis of the interpolation of the graphs $Nu = f(\beta, Re_l, Pr_l)$ from the works [3] (in this work, spline interpolation was used by the values of Re_l, Pr_l i β_o in the Mathcad package). The parameter β_o was determined by the following formula for the single-phase flow:

$$\beta_o = C_{fo} Fr_l/2, \quad (9)$$

where $C_{fo} = 0,079/Re_v^{0,25}$ *at* $Re_v < 10^5$; $C_{fo} = 0,046/Re_v^{0,2}$ *at* $Re_v > 10^5$.

The results of this comparison are presented in Fig. 5.



a) Fluid: R22 b) Fluid: steam

Fig. 5. Calculated *vs*. experimental heat transfer coefficients: predictions by Bae *et al*. [3] theoretical model.

Table 2. Statistical comparison of Bae *et al*. [3] theoretical model with experimental data (in %)

Statistical comparison	R22 *G*=300÷119 kg/(m²s) *x*=0.99÷0.56; q=50÷5 kW/m²	Steam *G*=54÷9 kg/(m²s) *x*=0.98÷0.4; q=320÷40 kW/m²
e_A	13.9	38.5
e_R	-11.3	-37.6
σ_N	12.2	14.2
Percentage of predicted points lying within ±25% error bars	77	18

The statistical comparison of the theoretical model of Bae *et al*. [3] with the experimental data is summarized in Table 2 which presents the mean absolute deviation e_A, the average deviation e_R, and the standard deviation, σ_N, given in Eqs. (10)–(12), respectively, along with the percentage of predicted points lying within ±20% error bars.

$$e_A = 1/n \sum 100\left|\left(\alpha_{calc} - \alpha_{exp}\right)/\alpha_{exp}\right|, \quad (10)$$

$$e_R = 1/n \sum 100\left[\left(\alpha_{calc} - \alpha_{exp}\right)/\alpha_{exp}\right], \quad (11)$$

$$\sigma_N = \left[1/(n-1)\sum(e-e_R)^2\right]^{0.5}. \quad (12)$$

where $e = 100\left[(\alpha_{calc} - \alpha_{exp})/\alpha_{exp}\right]$, n – number of calculation points.

Figure 5 and table 2 show that the greatest discrepancy is observed for the case of steam condensation. For freon R22, part of the data at Re_l=2300÷970 exactly matches the theory.

Such a deviation of the experimental data from theoretical calculations can be explained as follows. The calculations by the theory of film condensation essentially depend on the method of determining the loss of pressure on friction, and, accordingly, the friction coefficient C_f, which is included in the dependence (1) for the calculation of heat transfer. As noted in [6,18], none of the available scientific works have a sufficiently precise idea of the nature of the effect of friction force condensation on the boundary between phase separation. The determination of the friction coefficient C_f for a single-phase flow by the formula (9) is valid only if the value of the Lokart-Martinelli X_{tt} parameter is approximately zero. In this case, as follows from the numerous calculated dependences of works [19,20], the friction coefficients in single-phase and two-phase streams will be the same. However, in annular and intermediate flow modes, the value of the parameter X_{tt} can reach nonzero values. In particular, in this paper, when performing experimental studies, the value of the parameter X_{tt} varies from 0.005 to 0.24. As the experiments have shown, with these values of X_{tt}, the two-phase flow begins to affect the friction coefficient C_f, which leads to the increase in the parameter β, and local and medium heat transfer coefficients, respectively. In addition, as a result of experimental studies of local heat transfer coefficients, the growth of α_φ with the increase in the heat flux was noted, which is not taken into account in formula (1) and other well-known methods for calculating film condensation [11,12,15].

Therefore, in order to improve the coordination of the theory [3] with the experimental data obtained, in this paper it is proposed to take into account the influence of the heat flux and the parameters of the two-phase flow on the parameter β, which characterizes the effect of the interfacial friction force τ_f on the heat transfer.

Investigating the influence of the two-phase flow parameters on average heat transfer

The two-phase flow is characterized by such basic parameters as the mass velocity of the vapour *G*, the vapour content *x*, the vapour density ρ_v, and the density of the liquid ρ_l. In this paper, the influence of these parameters on heat transfer is recommended to be taken into account by the parameter β_v:

$$\beta_v = \Phi_v^2 \beta_o \quad (13)$$

where Φ_v^2 – correction complex for taking into account the influence of the two-phase flow.

The dependence for the calculation of Φ_v^2 was selected, based on the analysis of experimental data on average heat transfer obtained when condensing refrigerant R22 to the annular and intermediate regimes. In these studies, the input mass speed of the couple ranged from 119 to 305 kg/(m²s), and of vapour content – 0.99 to 0.56. All experiments were performed for the same $\overline{q_\varphi}$, but for variable *G* and *x*. This procedure allows examining the specific impact on heat transfer of vapour velocity and vapour content at constant value or small changes of other characteristics.

To calculate the parameter Φ_v^2 four known formulae were used, which are presented in the works [21-24]:

from [21]: $\Phi_v^2 = 1 + 9,4X_{tt}^{0,62} + 0,564X_{tt}^{2,45}$; (14)

from [22]:

$$\Phi_v^2 = \left\{1 + 0,5\left[\frac{G}{gd\rho_v(\rho_l - \rho_v)^{0,5}}\right]^{0,75} X_{tt}^{0,36}\right\}^2; \quad (15)$$

from [23]: $\Phi_v^2 = \left(1 + 2,85X_{tt}^{0,523}\right)^2$; (16)

from [24]: $\Phi_v^2 = 1 + CX_{tt}^n + X_{tt}^2$, (17)

where

$$C = 21\left\{1 - e^{(1-0,28Bo^{0,5})}\right\}\left\{1 - 0,9e^{-0,02Fr_l^{1,5}}\right\}, \quad n = 1 - 0,7e^{-0,08Fr_l},$$

$$Bo = \frac{gd^2(\rho_l - \rho_v)}{\sigma}, \quad Fr_l = \frac{Gx}{\sqrt{gd\rho_v(\rho_l - \rho_v)}}.$$

The results of calculating parameter Φ_v^2 from the experimental data using the formulae (14)-(17) are shown in Fig. 6. It shows that the value of the Φ_v^2 complex significantly changes depending on the formula for its calculation and it is impossible to choose the best among them. Therefore, the experimental values of the mean values by φ heat transfer coefficients with the theoretical calculation (1) are compared. The number Nu is determined by the values Re_l, Pr_l and β_v. The coefficient β_v is calculated by the formula (13) using the dependences (14)-(17) to determine the complex Φ_v^2. The results of the comparison are shown in Fig. 7.

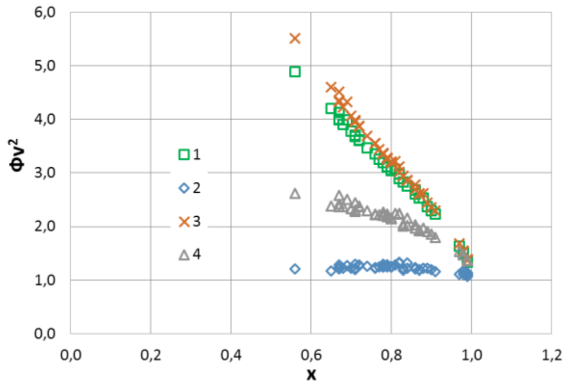


Fig. 6. Variation of Φ_v^2 with *x*: *1* – Φ_v^2 calculated by equation (14); *2* – by (15); *3* – by (16); *4* – by (17).

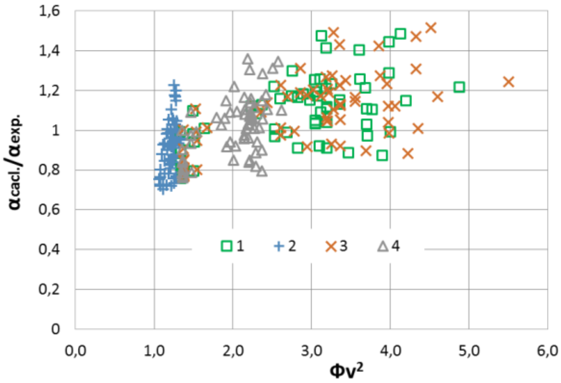


Fig. 7. Comparison of the experimental data for R22 condensation with calculations by (1): *1* – Φ_v^2 calculated by equation (14); *2* – by (15); *3* – by (16); *4* – by (17).

As it can be seen from Fig. 7, using the formulae (14) and (15), with the increase of parameter Φ_v^2 the values of heat transfer coefficients are overestimated by 50%. The calculation of Φ_v^2 by the formula (16) practically does not affect heat transfer. The best correlation between the experiment and the calculation is provided by the formula (17), which permits to summarize all data with an accuracy of ± 20%. Therefore, the dependence (17) is recommended to be used for determining the β_v coefficient by (13).

Consequently, the effect of the two-phase condensation current on the heat transfer is recommended to be taken into account by the correction of β_v by the formula (13), which makes it possible to use the theory (1) to calculate the heat transfer in the case of condensation of freon R22 not only for the annular, but also for the intermediate phase flow with sufficient accuracy (± 20 %).

Investigating the influence of heat flux on the average heat transfer

The effect of the heat flux on the hydraulic resistance and heat exchange in two-phase flows under phase transformations is theoretically justified in [25,**Error! Reference source not found.**] and is explained by the phenomenon of suction of the condensate mass into vapour in the adjacent layer. In [25] it is shown that at a relative rate of suction $j = q/rGx > 10^{-4}$, the hydraulic resistance of the friction coefficient C_f at the boundary of the phase separation increases in comparison with the resistance of the single-phase flow (C_{fo}) and is described by the formula:

$$C_f/C_{fo} = 1 + 17{,}5\,\mathrm{Re}_v^{0{,}25}\,j\,. \tag{18}$$

In [**Error! Reference source not found.**] the dependence for the definition of C_f has the following form:

$$C_f/C_{fo} = (1-0.25b)^2/(1+0.25b)^{0.2}\,, \tag{19}$$

where $b = -2q/rGxC_{fo}$.

In this case, there is a limitation of the use of the formula (19) with *b* less than minus 4, when the phenomenon of suction of the condensate mass does not affect the C_f/C_{fo} ratio.

The authors of [27] suggested much earlier than in [25,**Error! Reference source not found.**] the dependence for taking into account the influence of the heat flux on the C_f/C_{fo} ratio:

$$C_f/C_{fo} = b/(e^b - 1)\,. \tag{20}$$

In this paper, the effect of the heat flux on heat transfer is proposed to be taken into account by the correction β_q:

$$\beta_q = \Phi_q \beta_o\,, \tag{21}$$

where $\Phi_q = C_f/C_{fo}$ – correction factor to take into account the influence of heat flux.

The formula for determining the C_f/C_{fo} value was selected by analyzing the experimental data obtained for the case of steam condensation by the following parameters: t_s=102 °C, *G*=54÷9 kg/(m²s), *x*=0,4÷0,98, *q*=320÷40 kWt/m², which correspond to the annular and intermediate phase flow modes. All experiments were performed for the same *G* and *x*, but for variable $\overline{q_\varphi}$. This allowed to show the special effect of the heat flux on the heat transfer at a constant value or at a slight change in the other characteristics.

The results of the calculation of the Φ_q parameter are shown in Fig. 8. It is seen that calculations by formulae (18)-(20) provide values very close among themselves (difference within ± 10%), and to calculate the coefficient β_q for (21) one can use any of the formulae. In this work, the dependence (18) with the limitation of the suction effect on the boundary values of parameter *b* is used less than minus 4.

The analysis of the obtained experimental data showed that the effect of the heat flux on the heat transfer should be taken into account with β_q correction by the formula (21). This makes it possible to use the theory (1) to calculate the heat transfer in the case of steam condensation not only for the annular, but also for intermediate phase flow with adequate accuracy (± 25%).

Fig. 8. Variation of Φ_q with average by φ heat flux: *1* – Φ_q calculated by equation (18); *2* – by (19); *3* – by (20).

Comparing the proposed method with the obtained experimental data

The practical application of the developed method for calculating heat transfer under various condensation parameters is complicated by the necessity of the solution of the dependence (1) given in an implicit form. Solving the equation (1) is possible in two ways. Graphically, using the interpolation of graphs in Figs. 3 and 4, or numerically. However, the numerical solution (1), presented in [3,4] has a very cumbersome and uncomfortable look for permanent use. Therefore, the proposed method for calculating heat transfer in this paper is presented in the form of a special program, which is developed in the package Mathcad. With the help of information technology of "cloud" functions, the principle of which is described in detail in [28], the developed program is available on the Internet by link: http://twt.mpei.ac.ru/MCS/ Worksheets/Thermal/Heat-transfer-during-condensation-in-smooth-horizontal-tubes.xmcd and is available for all users.

The work of this program is based on the following algorithm:

1. Using the values of the Froude numbers Fr_l and the friction coefficient for the single-phase flow C_{fo}, the parameter β_o is determined by the formula (9).

2. The values of correction complexes Φ_v^2 and Φ_q are calculated from the relations (13) and (21), respectively.

3. The parameter β_{qv} which takes into account the influence of both the two-phase current and the heat flux on the frictional force τf is determined by the formula:

$$\beta_{qv} = \beta_o \Phi_v^2 \Phi_q \,. \qquad (22)$$

4. The values of the dimensionless Nu number are calculated on the basis of the interpolation of the graphs given in Figs. 3 and 4. To do this, the Mathcad package uses spline interpolation with values Re_l, Pr_l and β_{qv}.

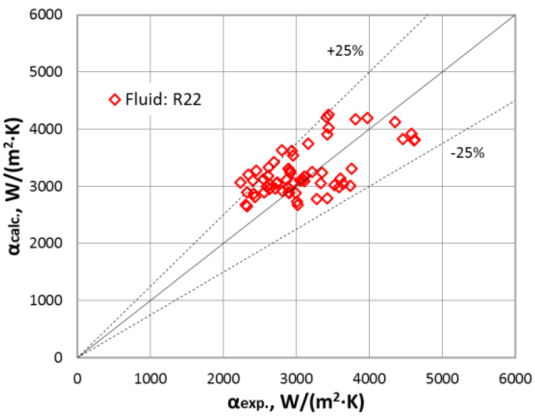
5. The values of heat transfer coefficient are based on the formula $\alpha = Nu_f \lambda_l \left(v_l^2/g\right)^{-1/3}$.

Table 3 shows a part of the experimental data by average heat transfer coefficients obtained for three different cases. The first is with the complex $\Phi_v^2 \approx 1{,}0$, and Φ_q>1.6. The second, on the contrary, is Φ_q<1.2 and $\Phi_v^2 > 1{,}6$. And the third case, when the value of both complexes is more than 1.6. Theoretical values of Nusselt numbers Nu_o, Nu_v and Nu_{vq}, are obtained using the developed program by the following parameters: for Nu_o: Re_l, Pr_l and β_o; for Nu_v: Re_l, Pr_l and β_v; for Nu_{vq}: Re_l, Pr_l and β_{qv}. It can be seen that under the values $\Phi_q << \Phi_v^2$ it is possible not to take into account the influence of heat flux on the heat transfer and, accordingly, not to take into account the effect of Φ_v^2, when Φ_q >> Φ_v^2. Considering these two amendments makes it possible to obtain a fairly accurate (up to ± 25%) reconciliation of calculation and experimental data.

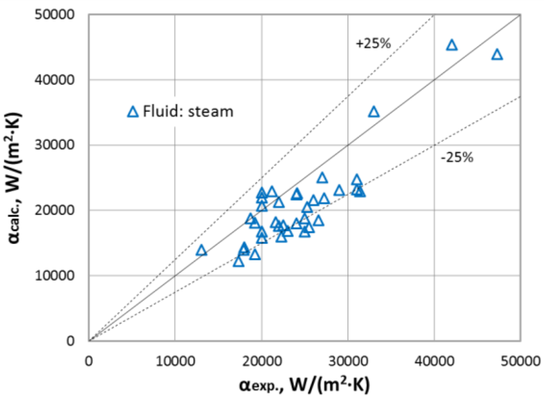
The comparison of all experimental data obtained with the proposed method for the annular and intermediate flow regimes is shown in Fig. 9. The statistical comparison of the suggested method with the experimental data is summarized in Table 4.

Table 3. Comparison of the experimental data for R22 and steam condensation with theoretical calculations

№	Fluid	$\frac{G,\ [kg/(m^2 s)]}{x}$	$\frac{\bar{q}\cdot 10^{-3}[W/m^2]}{\bar{\alpha}\cdot 10^{-3}[W/(m^2K)]}$	$\overline{Nu}_{exp}$	$Fr_l \cdot 10^{-3}$	Re_l	$C_{fo}\cdot 10^3$	$\frac{\beta_o}{Nu_o}$	$\frac{\Phi_v^2}{Nu_v}$	$\frac{\Phi_q}{Nu_{vq}}$
1	Steam	36 / 0.975	180 / 42	1.26	10	59	5.25	26 / 0.99	1.02 / 1.0	1.6 / 1.3
2	Steam	26 / 0.97	190 / 33	0.99	5.08	57	5.72	15 / 0.74	1.02 / 0.75	1.82 / 1.06
3	R22	192 / 0.9	5 / 2,81	0.43	3.57	2287	3.91	7 / 0.38	1.85 / 0.43	1.04 / 0.44
4	R22	232 / 0.86	18,2 / 3.35	0.,5	4.33	4134	3.35	8 / 0.4	2.1 / 0.47	1.2 / 0.49
5	R22	139 / 0.67	37.3 / 2.89	0.44	1.11	5512	4.39	2 / 0.35	2.4 / 0.39	1.8 / 0.44
6	Steam	36 / 0.48	275 / 19.2	0.57	2.4	1209	6.28	8 / 0.34	1.82 / 0.41	2.57 / 0.55



a) Fluid: R22



b) Fluid: steam

Fig. 9. Calculated *vs*. experimental heat transfer coefficients: predictions by the improved model

Table 4. Statistical comparison of the suggested method with experimental data (in %)

Statistical comparison	R22	Steam
	G=300÷119 kg/(m^{2}s); x=0.99÷0.56; q=50÷5 kW/m^2	G=54÷9 kg/(m^{2}s); x=0.98÷0.4; q=320÷40 kW/m^2
e_A	12.1	17.1
e_R	1.1	-10.9
σ_N	15	14.4
Percentage of predicted points lying with in ±25% error bars	92	95

It can be seen from Fig. 9 and Table 4 that the calculation for complexes Φ_v^2 and Φ_q to determine the correction β_{qv} by the formula (22) greatly improves the reconciliation of experimental data with the proposed method.

Comparing the proposed method for calculating heat transfer with other methods on the basis of experimental data of various authors

In order to confirm the accuracy of the developed method, its verification was performed with experimental data from the works of the following scientists: concerning the condensation of steam – Boyko [**Error! Reference source not found.**]; freons R-22, R-123 and R-134a – Yu *et al*. [30]; carbon dioxide –Kim and Jang [31]; propylene, propane, ether and isobutane – Park *et al*. [32]; refrigerant FC-72 –Lee *et al*. [33] and refrigerants R-245fa, Novec®649, HFE-7000 –

Ghim and Lee [34]. The results are shown in Figs. 10-15, which make it evident that the developed method of calculation with an accuracy of ± 25% generalizes all experimental data in the annular and intermediate modes of phase flow.

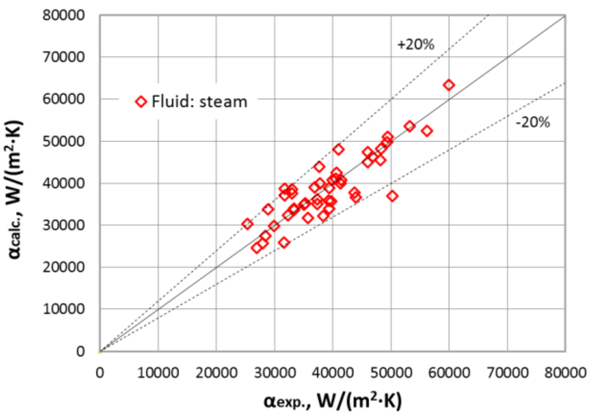


Fig. 10. Application of the improved model to Boyko [**Error! Reference source not found.**] data.

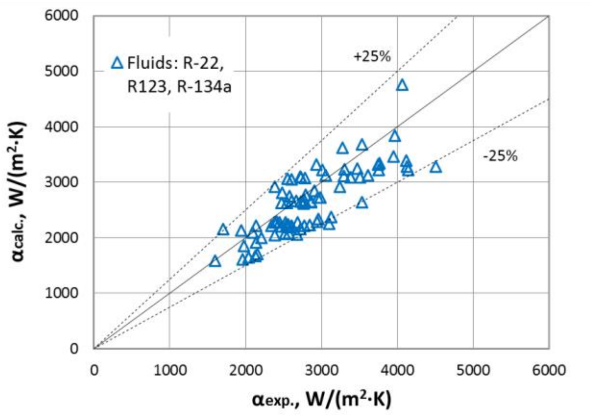


Fig. 11. Application of the improved model to Yu *et al.* [30] data.

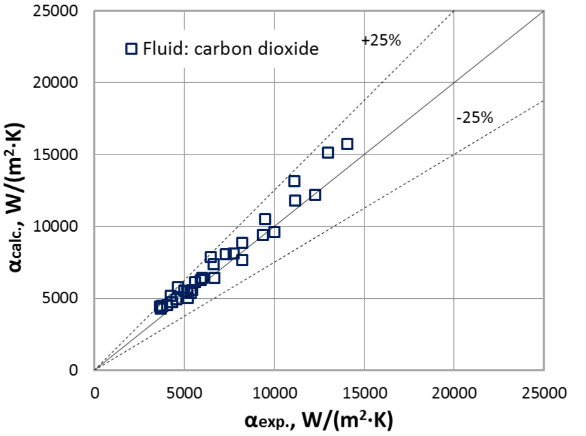


Fig. 12. Application of the improved model to Kim and Jang [31] data.

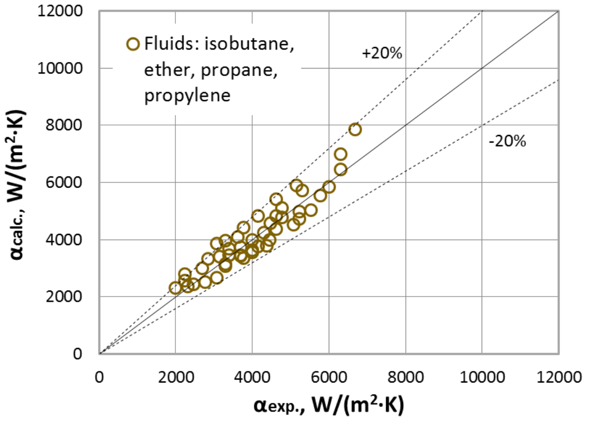


Fig. 13. Application of the improved model to Park *et al.* [32] data.

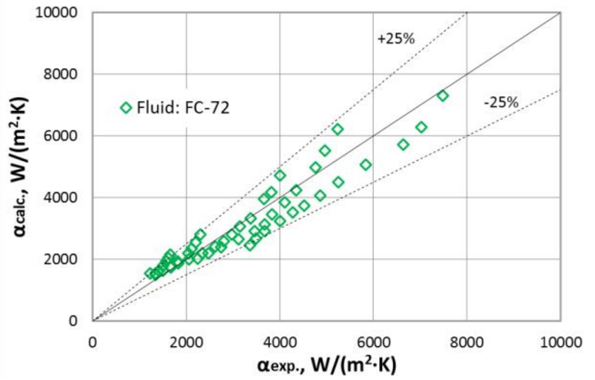


Fig. 14. Application of the improved model to Lee *et al.* [33] data.

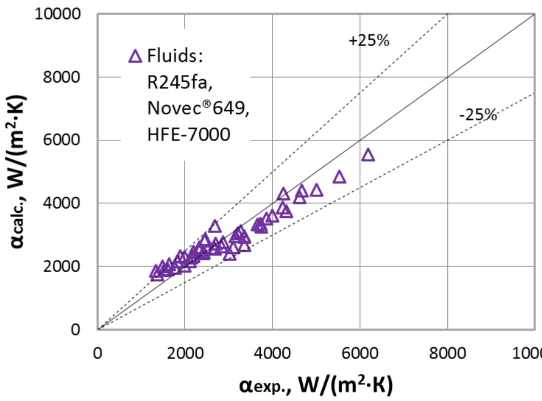


Fig. 15. Application of the improved model to Ghim and Lee [34] data.

CONCLUSIONS

As a result of the theoretical studies and the analysis of experimental values averaged by the perimeter of the tube of the heat transfer coefficients during the condensation of freon R22 and steam within the horizontal tube, we can formulate the following conclusions:

1. The application of the theory of film condensation [3] was proven to be correct when calculating the heat transfer during condensation in a horizontal tube for the annular phase flow regime. The necessity of more accurate estimation of influence, both of two-phase condensation current, and of heat flux on the laws of heat exchange is shown.

2. The influence of the two-phase condensation current on the heat transfer is recommended to be taken into account with the parameter $\beta_v = \Phi_v^2 \beta_o$. It is experimentally proven that the formula (17) for calculating the complex Φ_v^2 is correct.

3. The effect of heat flux on the heat transfer is expedient to be estimated by correction $\beta_q = \Phi_q \beta_o$. In this case, the application of formula (18) is used experimentally to calculate the parameter Φ_q.

4. The use of complexes β_v and β_q for the calculation of heat exchange by the theory of film condensation (1) allows to generalize with adequate accuracy (error ± 25%) both the experimental data obtained and the experimental data of other authors for the annular and intermediate modes of phase flow in a wide range of changes in regime parameters. Such calculation accuracy is not achievable for other, most commonly used methods [11,12,15].

5. The developed method for calculating heat transfer is presented as a program in the package Mathcad. With the help of information technology of "cloud" functions, this program is available on the Internet by the link: http://twt.mpei.ac.ru/MCS/Worksheets/Thermal/Heat-transfer-during-condensation-in-smoth-horizontal-tubes.xmcd and is available to all users.

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]
Nu_f – film Nusselt number, ($= \dfrac{\alpha}{\lambda_l}\left(\dfrac{v_l^2}{g}\right)^{1/3}$)
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_v – vapor Reynolds number ($= Gxd/\mu_v$)
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
ΔT – temperature difference (=t_s-t_w), [K]
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 – shear stress, [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
f – frictional factor
l – liquid
v – vapor/gas
exp – experimental
calc – calculated;
+ – non-dimensional symbol

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ВЪЗСТАНОВЯВАНЕ НА КОРЕКТНОСТТА И ПОДОБРЯВАНЕ НА МОДЕЛА ЗА ФИЛМОВА КОНДЕНЗАЦИЯ В ТРЪБИ

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(Резюме)

В статията са разгледани експерименталните изследвания, целящи увеличаване на ефективността на хоризонтални тръбни кондензатори чрез много точна оценка на топлопреноса и режимните параметри при кондензиране на различни охладители в хоризонталните тръби на устройствата. Уникалните измервания на топлинните потоци и коефициентите на топлопренос са проведени при кондензация на Freon R-22 и водна пара в широк интервал на режимните параметри (G, x, q, Re_l). Предложен е подобрен модел на филмова кондензация в хоризонталните тръби за предсказване на топлопреноса с използване на резултатите от цифровите решения на Bae *et al*. В този модел е дефиниран по-точно междуфазният фрикционен коефициент като основен параметър, определящ кондензацията. По-точната дефиниция включва експериментална обосновка на предсказването на β_q за изчисляване на загубите в налягането от триенето и корекцията на β_q, която взема пред вид повърхностното засмукване в междинната фаза. Топлообменът, предсказан от подобрения метод, е сравнен с експерименталните данни на различни автори за 13 течности (водна пара, R-22, R-123, R-134a, R-245fa, въглероден диоксид, пропилен, пропан, етер, изобутен, охладители FC-72, Novec®649, HFE-7000) в пръстеновидни и междинни режими. Доброто съвпадение между експериментите и изчисленията (различие в рамките на 25%) доказва коректността на предложения метод при ламинарни и турбулентни потоци на кондензатния филм.