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Prediction of effective heat transfer coefficients for vapour condensation inside horizontal tubes in stratified phase flow

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In modern condensers of air conditioning systems, heat pumps, evaporators of seawater desalination systems, and heaters of power plants, the process of vapour condensation is carried out mainly inside the horizontal tubes and channels. Heat transfer processes occurring in condensers have a significant effect on the overall energy efficiency of the mentioned systems.

In this paper, the experimental investigation of heat transfer during condensation of freons R22, R406a, and R407c in the plain smooth tube with d = 17 mm were carried out with the following parameters: t_s = 35–40°C, G = 10–100 kg/(m^{2}s), x = 0.8–0.1, q = 5–50 kW/m^2, ΔT = 4–14 K. The unique measurements of circumferential heat fluxes and heat transfer coefficients were carried out with the thick wall method during different condensation modes. It can be inferred that with the increase of the heat flux, at the top part of the tube the thickness of the condensate film increases, which leads to the decrease in heat transfer. At the bottom of the tube, the increase in the heat flux enhances heat transfer coefficient, that is characteristic of the turbulent liquid flow in the tube.

The obtained results allowed improving the prediction of effective heat transfer coefficients for vapour condensation, which takes into account the influence of condensate flow in the lower part of the tube on the heat transfer. This method generalises with sufficient accuracy (error ± 30%) the experimental data on condensation of freons R22, R134a, R123, R125, R32, R410a, propane, isobutene, propylene, dimethyl ether, carbon dioxide, and methane under stratified flow conditions. Using this method for designing heat exchangers, which utilise such types of fluids, will increase the efficiency of thermal energy systems.

Keywords: condensation, heat transfer, heat exchanger, plain tube, stratified flow

INTRODUCTION

Currently, the available methods and models for calculating heat transfer for condensing two-phase flows in horizontal tubes differ. Applicability and accuracy of these methods may vary depending on certain parameters of heat exchangers. Geometric sizes (length and diameter of tubes), thermophysical properties (thermal conductivity, density, surface tension, etc.) of condensing fluids and operating parameters (pressure, velocity, heat flux) vary 10 to 100 times in different heat exchangers. Inaccurate estimation of heat transfer can lead to an unjustified change in the size of the apparatus and pressure differences, resulting in the 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 crucial to study influence of regime parameters of two-phase flow on the regularities of heat transfer during the film condensation of moving vapour in a horizontal tube. These studies will open up the possibility of developing a new method for calculating heat transfer during condensation of various refrigerants in horizontal tubes of heat exchangers. A more precise estimation of heat transfer and regime parameters improves the efficiency of the horizontal tube condensers.

LITERATURE REVIEW

Characteristic features of the condensation process

During film condensation in horizontal heat exchangers, three different modes of condensate film flow and phase distribution can occur (Fig. 1). The extent of the modes depends upon physical properties of vapour and condensate, tube diameter and orientation, vapour velocity and heat flux. These modes are the following:

1. At the tube inlet, there is an annular flow of the phases. The condensate film has a minimum thickness and flows laminarly. Under a high vapour velocity, the condensate separation into vapour flow is possible. In this case, mist flow of the phases occurs. Under a decreasing vapour velocity, more liquid forms on the tube wall, film thickness increases, and transition from laminar flow of condensate film to turbulent flow takes place.

2. During condensation, at a certain distance from the tube inlet, an asymmetric flow of condensate film takes place. This happens because, due to gravity, the condensate film is flowing down under a definite angle to horizon. At the bottom of the tube, the condensate film flow is turbulent, while it is laminar at the top part. In this case, the intermediate or semi-annular flow of the phases occurs.

3. When gravity exceeds the shear stress of vapour, at low values of x, the stratified flow of the phases takes place. In this case, condensate flows down under the angle close to 90° to horizon at the top of the tube, while at the bottom of the tube, it flows as a stream due to the gradient of static pressure. Under a low vapour velocity at the tube inlet, this regime may take place along the full length of the tube.

In accordance with such classification of the phase flow mode, the methods of heat transfer prediction are developed.

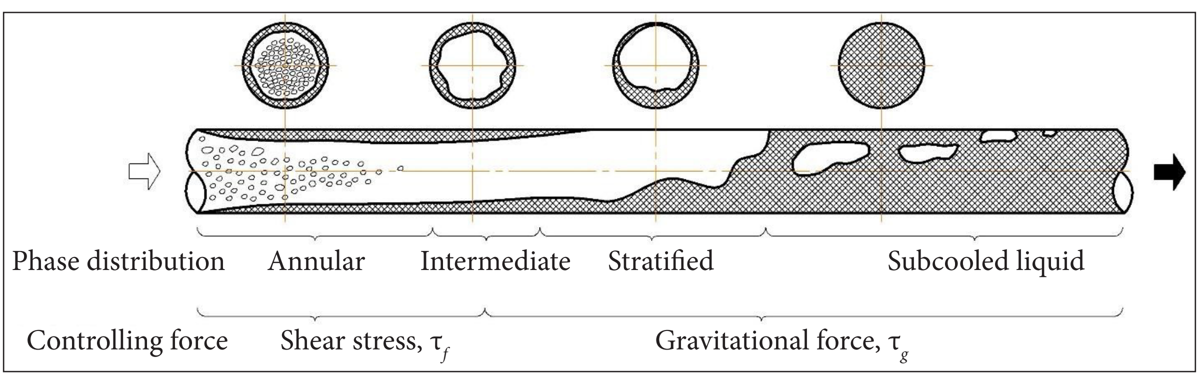


Fig. 1. Modes of condensate film flow and phase distribution according to [1]

Methods of heat transfer prediction

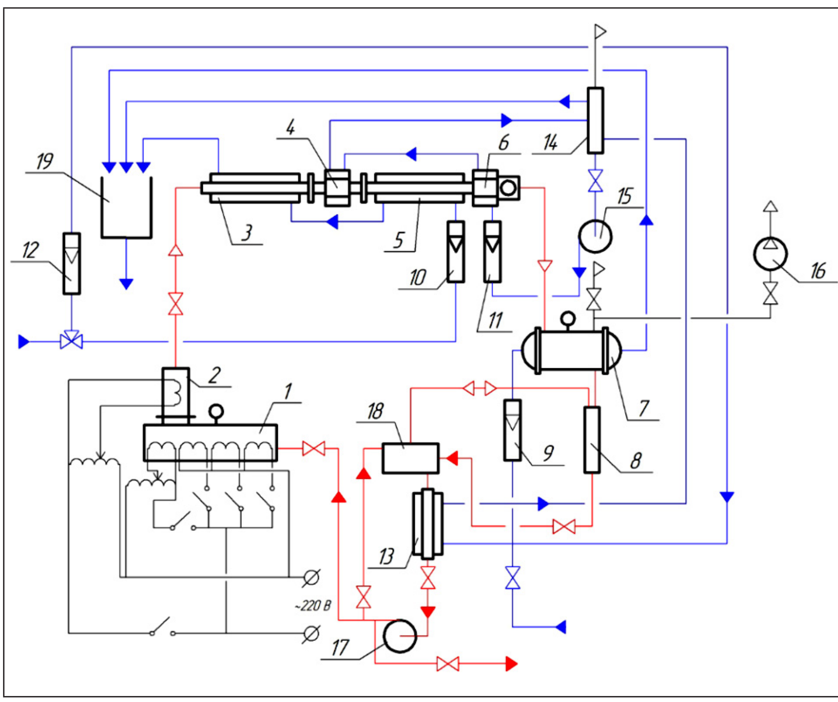
So far, hundreds of studies on condensation inside plain tubes and channels have been published. On the basis of a comprehensive review of the publications, the methods suggested by Thome et al. [2], Cavallini et al. [3], Shah [4], Rifert et al. [5, 6] are recommended for the accurate prediction of effective heat transfer coefficients in the case of condensation at annular and intermediate flow of the phases. However, in the case of the stratified mode of phase flow, there is no clear certainty about the use of different calculated dependencies. In most cases, an analogy is adopted between the processes taking place inside and outside of a horizontal tube. The heat transfer in this case is calculated by the Nusselt formula (1) obtained for condensation on the outer surface of the horizontal tube with or without heat transfer consideration in the condensate stream (see Fig. 1). The heat transfer in the stream is calculated by the formulae for convective heat transfer during the turbulent flow of the liquid.

$$\alpha_{strat} = 0.655\left[\frac{\lambda_l^3 \rho_l \left(\rho_l - \rho_v\right) gr}{\mu_l dq}\right]^{1/3}. \qquad (1)$$

It is shown in [7] that the authors of most publications have not investigated the influence of heat flux on the nature of heat transfer but studied the effect of mass velocity G and vapour content x instead. Also, there are no measurements of the heat transfer coefficients in the condensate stream, where their values are much lower than at the top of the tube and there are no clear recommendations regarding the limits of applying the proposed calculation dependencies.

DESCRIPTION OF EXPERIMENTAL INSTALLATION

Figure 2 shows the diagram of experimental installation. The experimental installation included a steam generator *1*, a steam superheater *2*, the first presection *3*, the first experimental section *4*, the second presection *5*, the second experimental section *6*, a condenser *7*, a rotameter *8* for measurement of condensate flow rate, rotameters *9*, *10*, and *11* for measuring the cooling water flow rate in the condenser *7*, presections *3* and *5*, and experimental sections *4* and *6*, respectively, and a rotameter *12* for measuring the cooling water flow rate supplied into the experimental sections. Presections *3* and *5* gave the possibility to create corresponding



1 – electric steam generator;
2 – separator-superheater;
3 – the first presection;
4 – the first experimental section;
5 – the second presection;
6 – the second experimental section;
7 – terminal condenser;
8 – volumetric condensate gauge;
9, *10*, *11*, *12* – rotameters;
13 – condensate overcooler;
14 – mixer;
15 – circulation pump;
16 – vacuum pump;
17 – condensate pump;
18 – receiver;
19 – drain collector

Fig. 2. Diagram of the experimental installation

modes of phase flow in the experimental sections *4* and *6*, respectively. All sections were located on one longitudinal axis. The inside diameter of the tubes in all sections was 17 mm, the length of both pre-sections was 0.8 m and of experimental sections 110 mm. All tubes were made of brass.

The experimental sections (Fig. 3) had 80-mm outer diameter. In the middle of each of the sections at the diameters d_1 = 23 mm and d_2 = 74 mm, five chromel-copel thermocouples were laid down at the points with φ = 0°, 45°, 90°, 135°, and 180°.

The measured wall temperatures at the points served as boundary conditions for solving the heat conduction equation. The experimental values of the local heat flux q_φ and heat transfer coefficient α_φ were determined by using the temperatures measured at these points and the following equations:

$$q_l = \frac{2\lambda_b \pi \left(t_i - t_j\right)}{\ln\left(d_2/d_1\right)}, \tag{2}$$

$$t_w = t_i + \frac{q_l}{\pi}\frac{1}{2\lambda_b}\ln\frac{d_1}{d}, \tag{3}$$

$$q_\varphi = q_l / (\pi d), \tag{4}$$

$$\alpha_\varphi = q_\varphi / \Delta T, \tag{5}$$

where q_l is the linear density of heat flux, W/m^2; λ_b is the thermal conductivity coefficient of brass experimental sections, W/(m·K); *i*, *j* are the numbers of the thermocouples at diameters d_1 and d_2, respectively (see Fig. 3).

The greater part of the experiments was carried out at local temperature differences in the wall and between the wall and the vapour above 9°C and 2°C, respectively. In all experiments, the temperature gradient in the wall in the axial direction was an order of magnitude smaller than the temperature gradient in the radial direction.

Saturation temperature t_s was measured by the thermocouple located at the inlet into the first presection and evaluated due to the thermocouple installed directly after the second experimental section. The maximum obtained relative uncertainty of the heat transfer coefficient was equal to 6%. Residual imbalance of the heat spent for heating water, which cooled all the sections and condenser, was determined by the quantity of condensate and did not exceed 2%.

The basic parameters of two-phase flow in all the sections of the experimental area and in the condenser were determined by solving the equations of the mass and heat balance recorded for each section, in which vapour condensation took place, and for the condenser. The range of change of these parameters is shown in Table 1. Local heat flux q_φ was varied, while *G* and *x* were kept constant throughout all experiments.

DETERMINING THE LIMITS OF THE FLOW REGIMES

The conducted experimental studies have shown that even minor asymmetry of the condensate flow

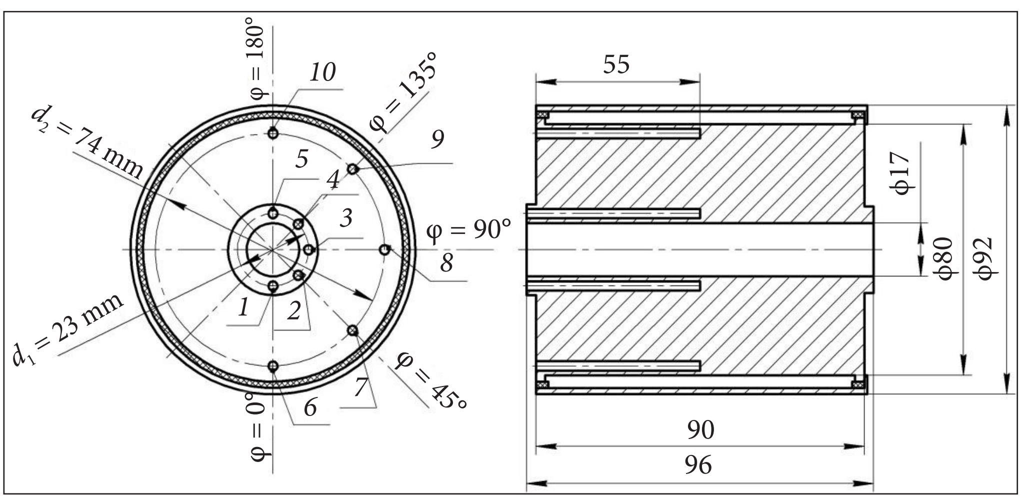


Fig. 3. Drawing of the brass working section: *1, 2, 3, 4, 5* – the channels for locating thermocouples at d_1 = 23 mm; *6, 7, 8, 9, 10* – the same, at d_2 = 74 mm

Table 1. **Main operating conditions during condensation tests**

Fluid	t_s, °C	G, kg/(m²·s)	Local vapor quality x	ΔT, °C	q_φ, kW/m²
R22	40	10–100	0.1–0.8	4–10	5–50
R406a	35			5–10	5–35
R407c	35			2–14	8–25

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 stratified mode of phase flow. The method of Rifert et al. [5, 6] was used to determine the limits of phase flow regimes. The boundary value of the shear stress τ_f and the gravitational force τ_g are calculated in this way:

at $\tau_f / \tau_g > 10$ – annular flow; (6)

at $1 \le \tau_f / \tau_g \le 10$ – intermediate flow; (7)

at $\tau_f / \tau_g < 1$ – stratified flow, (8)

$$\tau_f = C_f \rho_v w_v^2/2, \tag{9}$$

$$\tau_g = \rho_l g \delta. \tag{10}$$

The value of the film thickness δ is calculated from the equation:

$$\delta^+ = \delta/\nu_l(\tau_f / \rho_l)^{0.5}, \tag{11}$$

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

$$\mathrm{Re}_l < 50,\ \delta^+ = 0.7071\mathrm{Re}_l^{0.5}, \tag{12}$$

$$50 < \mathrm{Re}_l \le 1125,\ \delta^+ = 0.4818\mathrm{Re}_l^{0.585}, \tag{13}$$

$$\mathrm{Re}_l > 1125,\ \delta^+ = 0.095\mathrm{Re}_l^{0.812}. \tag{14}$$

The authors [5, 6] suggest determining the friction coefficient of the two-phase flow C_f in formula (9) according to the dependence (15). Equation (15) allowed evaluating both the effect of the two-phase flow (parameter Φ_v^2) and vapour suction at the interphase (parameter Φ_q) on the friction coefficient C_f and, accordingly, on the shear stress τ_f.

$$C_f = C_{fo}\, \Phi_v^2 \Phi_q, \tag{15}$$

where $C_{fo} = 0.079 / \mathrm{Re}_v^{0.25}$ at $\mathrm{Re}_v \le 10^5$ or $C_{fo} = 0.046 / \mathrm{Re}_v^{0.2}$ at $\mathrm{Re}_v > 10^5$.

Parameters Φ_v^2 and Φ_q are determined by the equations (16)–(17):

$$\Phi_v^2 = 1 + CX_{tt}^n + X_{tt}^2, \tag{16}$$

where

$$C = 21\left[1 - e^{\left(1-0.28Bo^{0.5}\right)}\right]\left[1 - 0.9e^{-0.02Fr^{1.5}}\right],$$

$$Fr = \frac{Gx}{\sqrt{gd\rho_v\left(\rho_l - \rho_v\right)}}, n = 1 - 0.7e^{-0.08Fr}.$$

$$\Phi_q = 1 + 17.5\,\mathrm{Re}_v^{0.25}\frac{q}{rGx}. \tag{17}$$

DEVELOPMENT OF THE HEAT TRANSFER METHOD

The influence of the circumferential heat flux on the local values of the heat transfer coefficients varies depending on φ. In all the experiments, the value of α_φ in the upper tube segment (φ > 90°) decreases with the increase of q_φ (Fig. 4). It is explained by the fact that with the increase in heat flux, the thickness of the condensate film also increases, which leads to the decrease in heat transfer. For φ < 90° at the bottom of the tube (in the condensate stream), α_φ increases under condition of increasing q_φ (Fig. 4). Such an impact of heat flux on heat transfer is characteristic of the turbulent liquid flow in the tube.

Experimental values α_φ in the upper part of the tube (φ > 90°) are larger then in the lower part (φ < 90) due to the large difference in condensate film thickness in the top and bottom parts of the tube. Figure 4 shows that the experimental values α_φ in the upper part of the tube (φ > 90°)

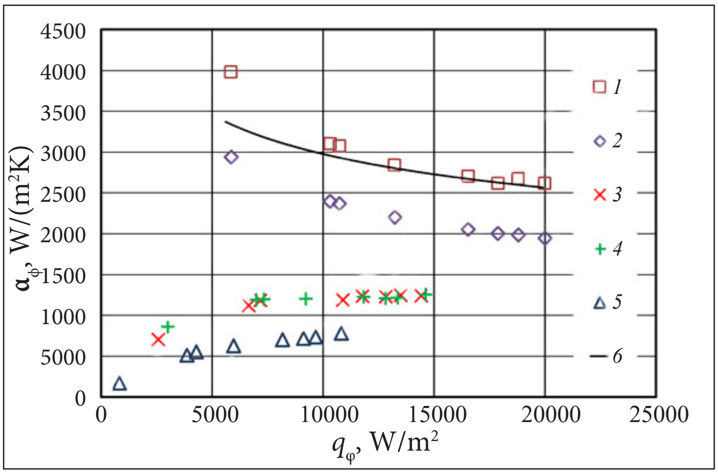


Fig. 4. The influence of heat flux q_φ on α_φ at $G = 43$ kg/(m²s); $x = 0.24$: *1* – $\varphi = 0°$; *2* – $\varphi = 45°$; *3* – $\varphi = 90°$; *4* – $\varphi = 135°$; *5* – $\varphi = 180°$; *6* – calculated by eq. (1) at $\varphi = 180°$

are quite consistent with the calculation according to the Nusselt formula (1). Such a result complies with the theory of film condensation and coincides with the data of [2, 8–10].

To calculate the heat transfer in the lower part of the tube, it is necessary to know the geometric parameters of the condensate stream (Fig. 5).

The authors of [11] performed comparative calculations of the angle of the tube flooding φ_{str} and that of the stream height h_{str} using different theoretical and empirical dependencies. In [11], it is shown that the most acceptable formula for calculating the angle of the tube flooding with the condensate stream is the dependence by Consetov (18). The geometric parameters of the condensate stream in this work are determined by dependencies (19)–(22):

– the angle of the tube flooding, rad:

$$\varphi_{str} = 2 \cdot \arccos\left[1 - 4.2 Fr_l^{0.33}\left(\frac{\sigma}{\rho_l g d^2}\right)^{0.25}\right], \quad (18)$$

– the height of the condensate stream, m:

$$h_{str} = \frac{d}{2}\left(1 - \cos\frac{\varphi_{str}}{2}\right), \quad (19)$$

– the tube area that is flooded with condensate, m²:

$$S_{str} = d^2 / 8(\varphi_{str} - \sin\varphi_{str}), \quad (20)$$

– wetted perimeter, m:

$$P_{str} = d_{str} [\varphi_{str} / 2 + \sin(\varphi_{str} / 2], \quad (21)$$

– equivalent diameter of the stream, m:

$$d_{str} = 4S_{str} / P_{str}. \quad (22)$$

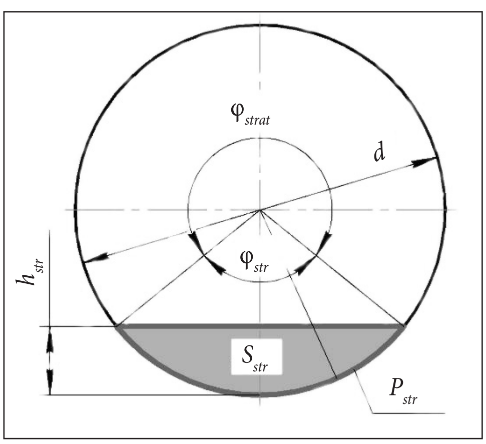


Fig. 5. Geometric parameters of the condensate stream under stratified conditions

The dependence for the calculation of convective heat transfer in the lower part of the tube ($0 \le \varphi \le \varphi_{str}$, Fig. 5) α_{bot} is the following:

$$\alpha_{bot} = \mathrm{Nu}_{bot}\frac{\lambda_l}{d_{str}} = c\,\mathrm{Re}_{str}^n\,\mathrm{Pr}_l^{0.5}\frac{\lambda_l}{d_{str}}, \quad (23)$$

where $\mathrm{Re}_{str} = w_{str} d_{str} / \upsilon_l$ – the Reynolds number in the condensate stream; w_{str} – the velocity of the condensate in the stream, m/s.

To determine the coefficients c and n in formula (23), the studies of freon condensation $R22$, $R406$a and $R407$c were performed precisely in the case of their stratified flow regime. In Fig. 6, the experimental values of α_φ in the lower part of the tube (φ = 0° and 45°, see Fig. 3) are generalized by the following equation:

$$Nu_{str} = 0.0161 Re_{str}^{0.842} Pr_l^{0.5}, \qquad (24)$$

the accuracy of approximation is $R^2 = 0.9521$.

Thus, formula (23) will have the following form:

$$\alpha_{bot} = 0.0161 Re_{str}^{0.842} Pr_l^{0.5} \frac{\lambda_l}{d_{str}}. \qquad (25)$$

Thus, the calculation of heat transfer in the case of the stratified regime of the phase flow is recommended to be performed by the following algorithm:

1. Calculate the angle of flooding of the lower part of the tube with the condensate stream φ_{str} and geometric parametres of the stream by formulae (18)–(22).
2. Determine the heat transfer coefficient in the upper part of the tube ($\varphi_{str} \le \varphi \le \varphi_{strat}$, Fig. 5) α_{top} by formula (1).
3. Calculate the value of the coefficient of heat transfer in the condensate stream ($0 \le \varphi \le \varphi_{str}$, Fig. 5) α_{bot}, by formula (25).
4. Calculate the effective heat transfer coefficients considering the values of heat transfer coefficients α_{top} and α_{bot}, as well as the angle of flooding φ_{str}:

$$\alpha_{strat} = \frac{\alpha_{top}(2\pi - \varphi_{str}) + \alpha_{bot}\varphi_{str}}{2\pi}. \qquad (26)$$

COMPARING THE PROPOSED METHOD WITH THE EXPERIMENTAL DATA OF VARIOUS AUTHORS

In order to confirm the accuracy of the developed method, its verification is performed with experimental data from the works of Cavallini et al. [12], Yu J et al. [13], Park et al. [14], Li et al. [15], and Zhuang et al. [16].

In [12], condensation of freons R22, R134a, R125, R32, and R410a occurred inside tubes with d = 8 mm and l = 1.0 m. Freon condensation temperature t_s = 40°C was the same for all the experiments. The mass velocity varied from 65 to 750 kg/(m²s), and the heat flux q, with the change in vapour quality Δx from 0.2 to 0.4, varied from 6 to 62 kW/m².

Yu et al. [13] investigated the condensation of freons R22, R123, and R134a inside the test section with a diameter of 8.4 mm. The total length of the horizontal tube was l = 6 m. The experiments were conducted under full vapour condensation by the temperature: t_s = 35–48°C for R22, t_s = 60–70°C for R123, and t_s = 48–49°C for R134a, accompanied by the change in mass velocity from 87 to 340 kg/(m²s).

Park et al. [14] investigated condensation of propylene, propane, dimethyl ether, and isobutane in tubes with the diameter d = 8.8 mm and

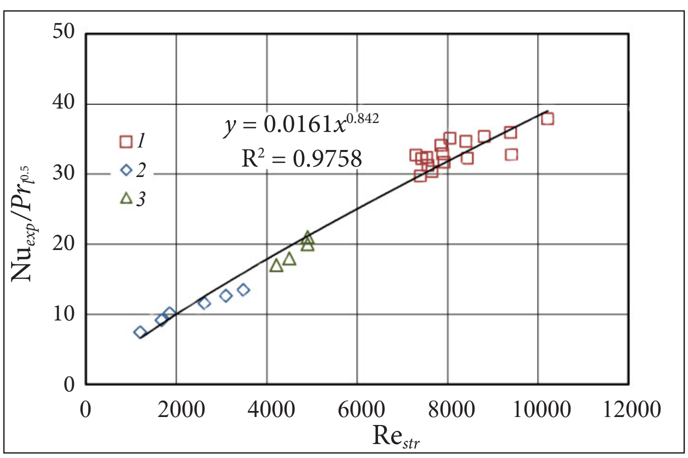


Fig. 6. Approximation of the experimental data on condensation: *1* – R22; *2* – R407c; *3* – R406a in the form $Nu_{str} = (Re_{str}, Pr_l)$

the length of 0.53 m at q = 7.3–7.7 kW/m^2 and t_s = 40°C.

In [15], the experimental data of carbone dioxide condensation heat transfer for mass fluxes G = 100–500 kg/(m^{2}s) inside a 4.73 mm diameter tube was presented at saturation temperatures t_s = 10 and 0°C under a wide range of vapour quality condition.

Zhuang et al. [16] investigated heat transfer in the course of methane condensation in a horizontal smooth tube with an inner d = 4 mm and length l = 200 mm under the following parameters: p_s = 2–3.5 MPa, G = 99–255 kg/(m^{2}s), and ΔT = 4.8–20.2 K.

For verification, we selected only the data that complied with the stratified mode of the phase flow according to equation (8). The results are shown in Fig. 7, which make it evident that the developed method of calculation with an accuracy of ±30% generalises all experimental data.

CONCLUSIONS

As a result of experimental research on the condensation of freons R22, R407c, and R406a inside the horizontal tube during the stratified regime of phase flow, we can formulate the following conclusions:

1. The different character of the influence of the heat flux on the heat transfer coefficient in the upper and lower parts of the tube is shown.

2. The method of heat transfer calculation in the case of condensation under the stratified phase flow regime is detailed. The obtained method, in contrast to other formulae, more accurately takes into account the effect of condensate flow in the lower part of the tube on the heat transfer. The verification showed it fits reasonably (±30%) the data on the condensation of various refrigerants in a wide range of changes in regime parameters.

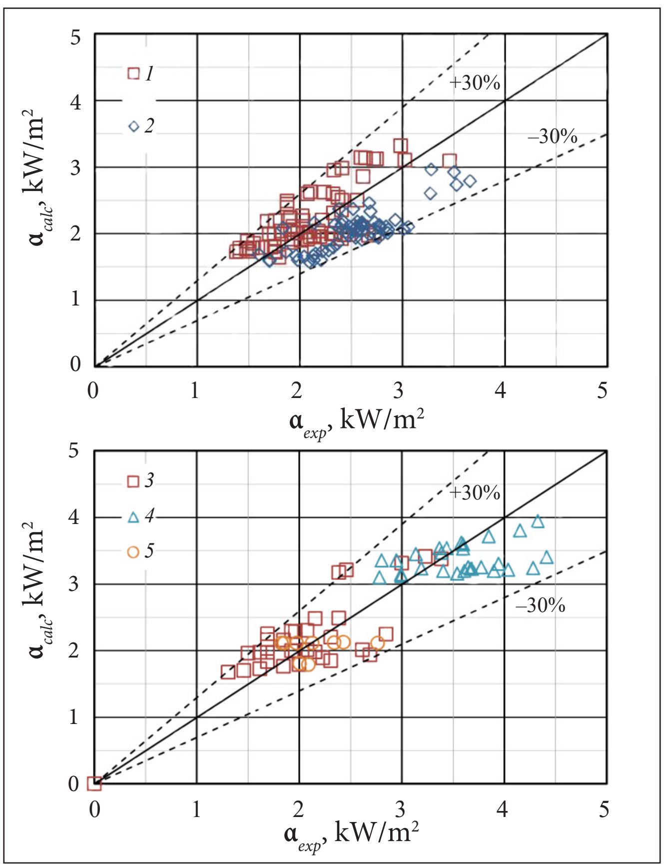


Fig. 7. Comparison of the suggested method with the experimental data on condensation of: *1* – freons R22, R134a, R125, R32, R410a, [12]; *2* – freons R22, R134a, R123, [13]; *3* – hydrocarbons, [14]; *4* – carbon dioxide, [15]; *5* – methane, [16]

3. The given method is recommended to be used in case of condensation of specific refrigerants (R22, R407c, R406a, hydrocarbons, carbon dioxide and methane) inside horizontal tubes. For other fluids, this model needs to be refined.

NOMENCLATURE

Bo – Bond number ($= gd^2 (\rho_l - \rho_v) / \sigma$)
C_f – two-phase flow friction coefficient
C_{fo} – one-phase flow friction coefficient
d – inner diameter of the tube, [m]
Fr_l – liquid Froude number ($= \frac{\left[G(1-x)\right]^2}{\rho_l^2 gd}$)
G – mass velocity, [kg/(m²s)]
g – gravitational acceleration, [m/s²]
l – length of the tube, [m]
Nu – Nusselt number ($= \alpha d / \lambda_l$)
Pr – Prandtl number
q – heat flux, [W·m⁻²]
q_φ – local heat flux along the tube perimeter, [W·m⁻²]
r – heat of vaporization, [J·kg⁻¹]
Re_l – liquid Reynolds number ($= G(1-x)d / \mu_l$)
Re_v – vapour Reynolds number ($= Gxd / \mu_v$)
t – temperature, [°C]
w – axial velocity, [m/s]
x – vapour quality
Greek symbols:
α – effective heat transfer coefficient, [W/(m²K)]
α_φ – local heat transfer coefficient along the tube perimeter, [W/(m²K)]
δ – thickness of the condensate film, [m]
ΔT – temperature difference ($= t_s - t_w$), [K]
λ – thermal conductivity, [W/(mK)]
μ – dynamic viscosity, [Pa·s]
ν – kinematic viscosity, [m²s⁻¹]
ρ – density, [kg/m³]
σ – surface tension coefficient, [N/m]
τ_f – shear stress, [Pa]
τ_g – gravitational 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
bot – bottom part of the tube
calc – calculated
exp – experimental
f – frictional
g – gravitational
l – liquid
s – saturated
strat – stratified
str – stream
top – top part of the tube
v – vapour
+ – dimensionless symbol

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EFEKTYVIŲ ŠILUMOS PERDAVIMO KOEFICIENTŲ PROGNOZAVIMAS GARŲ KONDENSACIJAI HORIZONTALIUOSE VAMZDŽIUOSE, ESANT SLUOKSNIUOTAM FAZINIAM SRAUTUI