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HEAT TRANSFER DURING REFRIGERANT BOILING IN SMOOTH HORIZONTAL TUBES OF MARINE LOW-TEMPERATURE AIR COOLERS

ТЕПЛООБМІН ПРИ КИПІННІ ХОЛОДОАГЕНТА В ГЛАДКИХ ГОРИЗОНТАЛЬНИХ ТРУБАХ СУДНОВИХ НИЗКОТЕМПЕРАТУРНИХ ПОВІТРООХОЛОДЖУВАЧІВ

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Abstract. The results of experiment for definition of heart transfer coefficient during refrigerant boiling in the horizontal tube of air coolers on the evaporating temperatures of –18 and –37 ºC were conducted, which on selected formulas for its calculations.
Purpose. To obtain experimental data on heat transfer during refrigerant boiling in a horizontal tube integrated into a refrigeration system, at temperatures as low as –40 °C, and to use these data to reasonably select calculation correlations for determining the heat transfer coefficient.
Method. In the design of air coolers (AC) for refrigeration systems, the accurate selection of calculation correlations for determining the heat transfer coefficient is crucial. An analysis of the equations describing heat transfer in ACs with in-tube boiling of refrigerants reveals contradictions in the choice of these correlations. Moreover, the average heat transfer coefficients calculated using different equations can vary significantly – by up to 75 %.
The available literature lacks experimental data on heat transfer coefficients that could serve as a reliable benchmark for selecting appropriate calculation correlations for determining heat transfer coefficients at boiling temperatures down to –40 °C. The value of the heat transfer coefficient depends not only on operating parameters but also on the presence of oil in the refrigerant, as well as other influencing factors. In studies of heat transfer to refrigerants boiling inside air cooler tubes, a horizontal tube wrapped with an electric heater has typically been used as a test evaporator.
Results. Equations and dependencies are obtained for calculating heat transfer during boiling of a refrigerant inside smooth horizontal copper pipes for zones of stratified and wave flow regimes, typical for the operating conditions of low-temperature air coolers.
Scientific novelty. As a result of research experimental data were obtained for heat transfer coefficients during in-tube refrigerant boiling at low temperatures (–37 °C and –18 °C). Based on the results, appropriate equations were selected, and a semi-empirical correlation accounting for the influence of oil on boiling heat transfer was proposed. The proposed equations are recommended for use in the design of low-temperature air coolers.
Practical importance. The proposed empirical and experimental correlations developed by the authors were applied in the design of hermetic refrigeration units for provision storerooms.
Key words: heart transfer; refrigerant boiling; heart transfer coefficient.

Анотація. Наведені результати експеримента по призначенню коефіцієнта тепловіддачі при кипінні холодоагенту у горизонтальній трубі повітроохолоджувача при температурах –18 і –37 ºC, по яким підібрани формули для його розрахунку.
Мета. Отримання дослідних даних з тепловіддачі при кипінні холодоагенту в горизонтальній трубі, включеної до схеми холодильної машини, при температурах до –40 ºC та на підставі цих даних обгрунтовано підібрати розрахункові залежності для визначення коефіцієнта тепловіддачі.
Методика. При проектуванні охолоджувачів повітря (ПО) систем рефрижерації важливим є правильний вибір розрахункових залежностей для визначення коефіцієнта тепловіддачі. Аналіз рівнянь, що описують теплооб-

мін у ПО з внутрішньотрубним кипінням холодоагентів, вказує на існування протиріч у їх виборі, а обчислені за різними рівняннями значення середніх коефіцієнтів тепловіддачі мають значну розбіжність (до 75 %).
У відомих літературних джерелах відсутні дані щодо експериментальних коефіцієнтів тепловіддачі, які могли б служити критерієм обґрунтованого вибору розрахункових залежностей для визначення коефіцієнтів тепловіддачі при температурі кипіння до –40 ºС. Значення коефіцієнта тепловіддачі залежить не тільки від режимних параметрів, а й від наявності у холодоагенті олії, і навіть інших чинників. При дослідженні процесів тепловіддачі до холодоагенту, що кипить усередині труб ПО, як опитний випарник зазвичай застосовували горизонтальну трубу з навитим на неї електронагрівачем.
Результати. Отримано рівняння та залежності для розрахунку тепловіддачі при кипінні холодоагенту всередині гладких горизонтальних мідних труб для зон розшарованого та хвильового режимів течії, характерних для умов роботи низькотемпературних охолоджувачів повітря.
Наукова новизна. В результаті досліджень отримані дослідні дані щодо коефіцієнтів тепловіддачі при внутрішньотрубному низькотемпературному (–37 і –18 ºС) кипінні холодоагенту, підібрано рівняння та запропоновано напівемпіричну залежність, що враховує вплив олії на тепловіддачу при кипінні. Запропоновані рівняння рекомендовані до застосування при проектуванні низькотемпературних охолоджувачів повітря.
Практична значимість. Запропоновані авторами розрахункові та дослідні співвідношення були використані при розробці герметичних холодильних машин для провізійних комор.
Ключові слова: теплообмін; кипіння холодоагенту; коефіцієнт тепловіддачі.

FORMULATION OF THE PROBLEM

In the design of air coolers (AC) for refrigeration systems, the accurate selection of calculation correlations for determining the heat transfer coefficient is crucial. An analysis of the equations describing heat transfer in ACs with in-tube boiling of refrigerants reveals contradictions in the choice of these correlations. Moreover, the average heat transfer coefficients calculated using different equations can vary significantly – by up to 75 % [2].

In [7], it is recommended that, when designing ACs with refrigerant boiling in smooth horizontal tubes, the Havlov equation be used to calculate heat transfer in the developed boiling region.

$$\alpha_a = 29\mathrm{Re}_L^{-0,3}\mathrm{Fr}_L^{0,2}\alpha_0 \ , \qquad (1)$$

$$\mathrm{Re}_L = \frac{\rho w(1-x)d_{in}}{\mu_L}\ ; \quad \mathrm{Fr}_L = \frac{(\rho w)^2(1-x)^2}{\rho_L^2 g d_{in}}\ ;$$

where Re_L and Fr_L – are the Reynolds and Froude numbers for the liquid phase, respectively; α_0 is the heat transfer coefficient during refrigerant boiling in a large volume, determined according to the expression obtained by Stephan [7].

$$\alpha_0 = \alpha_{St} = 0,071\frac{\lambda_L}{d_{B1}}\left(\frac{qd_{B1}}{T_0\lambda_L}\right)^{0,7}\left(\frac{d_{B1}T_0\lambda_L}{\sigma_L\nu_L}\right)^{0,3}\left[\frac{r\rho_п R_{sur}}{\rho_L d_{B1}^2\left(fd_{B1}^{0,5}\right)^2}\right]^{0,133}. \qquad (2)$$

Here $d_{в1}$ – is the bubble departure diameter, in meters:

$$d_{B1} = 0,511\left(\frac{\sigma_L}{g(\rho_L-\rho_S)}\right)^{0,5};$$

f – is the bubble departure frequency, in c^{-1}, calculated from the following relation:

$$fd_{B1}^{0,5} = 0,56\left(\frac{\rho_L-\rho_S}{\rho_L}g\right)^{0,5} = 1,75, \text{ м } 0,5/\text{с};$$

R_{sur} – is the surface roughness, in meters (for industrially manufactured tubes, $R_{sur} = (3...6)\cdot 10^{-6}$ m).

In [6], equations are presented for calculating heat transfer during refrigerant boiling inside a horizontal copper tube at low mass fluxes ($w_a\rho_a \le 100$ kg/(s·m²)) specifically for the stratified and wavy flow regimes that are typical under the operating conditions of low-temperature air coolers.

$$\alpha_a = \alpha_{l.v.}(1 + 1,7N), \qquad (3)$$

where $\alpha_{l.v}$ is the heat transfer coefficient during pool boiling (boiling in a large volume), determined using the equation proposed by G.N. Danilova,

$$\alpha_{l.v.} = 7,5\, q_{Fa}^{0,7}(0,14 + 2,2\, p_0/p_{кр}); \qquad (4)$$

p_{cr} – is the critical pressure, in Pa; $N = \mathrm{Bo}/\mathrm{K_p}$ – is a dimensionless group accounting for the convective component; $\mathrm{Bo} = \frac{w\rho r}{q_{Fa}}$; $\mathrm{K_p} = \frac{p_0 b}{\sigma}$; $b = \sqrt{\frac{\sigma}{g(\rho' - \rho'')}}$; r – is the latent heat of vaporization, in J/kg; p_0 – is the boiling pressure, in Pa; σ – is the surface tension, in N/m; g – is the gravitational acceleration, in m/s²; ρ', ρ'' – are the densities of the liquid and vapor phases of the refrigerant, in kg/m³.

ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

The available literature lacks experimental data on heat transfer coefficients that could serve as a reliable benchmark for selecting appropriate calculation correlations for determining heat transfer coefficients at boiling temperatures down to –40 °C.

The value of the heat transfer coefficient depends not only on operating parameters but also on the presence of oil in the refrigerant, as well as other influencing factors.

PURPOSE OF THE STUDY

The aim of this study is to obtain experimental data on heat transfer during refrigerant boiling in a horizontal tube integrated into a refrigeration system, at temperatures as low as – 40 °C, and to use these data to reasonably select calculation correlations for determining the heat transfer coefficient.

MAIN MATERIAL

In studies of heat transfer to refrigerants boiling inside air cooler tubes, a horizontal tube wrapped with an electric heater has typically been used as a test evaporator [1].

The experimental setup can be significantly simplified if the heater (evaporator) is a tube through which electric current is passed, generating heat in the tube proportional to the square of the current and the tube's electrical resistance. Since the evaporator tube in this case acts as an element with an internal heat source, the temperature distribution problem differs substantially from the classic case of heat transfer between two fluids across a separating wall. To determine the heat transfer coefficient, it is necessary to know the temperature distribution law within the tube wall.

Solutions to this problem are known for cases where the physical properties of the tube material (thermal conductivity and specific electrical resistance) are either constant or only the thermal conductivity varies with temperature. However, ignoring the temperature dependence of specific electrical resistance can lead to substantial errors – over 50 % – in solving the problem described above [3].

As a result of an analytical study of heat transfer during refrigerant boiling in a horizontal tube heated by electric current, equations have been derived (see detailed derivation in [3, 5]) to determine the temperature gradient in the tube wall, taking into account the temperature dependence of both the thermal conductivity and the specific electrical resistance of the tube material

$$t_2 - t_1 = \frac{1+(C+D)t_2}{C+D}[1-\psi], \qquad (5)$$

and the mean temperature difference between the tube wall and the refrigerant

$$\left(t_1 - t_a\right)_{\text{av}} = \frac{d}{b}, \qquad (6)$$

where t_a, t_1, t_2 − are the temperatures of the refrigerant, the inner tube wall, and the outer tube wall, respectively, in °C; C and D are the temperature coefficients of electrical resistivity and thermal conductivity, respectively, in K⁻¹.

$$\psi = \frac{J_1(m\cdot r_2)Y_0(m\cdot r_1) - J_0(m\cdot r_1)Y_0(m\cdot r_2)}{J_1(m\cdot r_2)Y_0(m\cdot r_2) - J_0(m\cdot r_2)Y_0(m\cdot r_2)},$$

J_0, J_1, and Y_0 are Bessel functions of the first and second kind of zero and first order, respectively;

$$m = \sqrt{(C+D)\left(\frac{q_V}{л}\right)_0};$$

r_2, r_1 – are the outer and inner radii of the tube, in meters;

λ, λ_0 are the thermal conductivity and its value at $t = 0$ °C, in W/(m·K);

q_v is the volumetric heat generation rate due to electric current, in W/m³;

$d = \frac{q_v}{\lambda}$; $b = \frac{2\pi\, r_1\, \alpha_a}{f\lambda}$; $f = \pi\left(r_2^2 - r_1^2\right)$ – is the cross-sectional area of the tube wall;

α_a – is the heat transfer coefficient, in W/(m²·K).

To derive the expression for the heat transfer coefficient α_a during refrigerant boiling inside the tube, we jointly solve equations (5) and (6), and, taking into account the temperature dependence: $q_v = q_{Fa}\frac{2r_1}{r_2^2 - r_1^2}$, we obtain:

$$\alpha_a = \frac{q_{Fa}}{(t_2 - t_a) - \left[1+(C+D)t_2\right]\frac{(1-\psi)}{(C+D)}}, \qquad (7)$$

where q_{Fa} – is the heat flux density, in W/m².

In determining the temperature difference between the outer surface of the evaporator tube and the refrigerant, the following assumptions were made: the refrigerant temperature is constant along the entire length of the tube; the rate of heat generation across the tube wall and along its length is uniform; the heat generated within the tube wall is transferred solely through the inner surface. The latter assumption is justified by the fact that the tube is enclosed in a thermally insulated shell (a calorimeter).

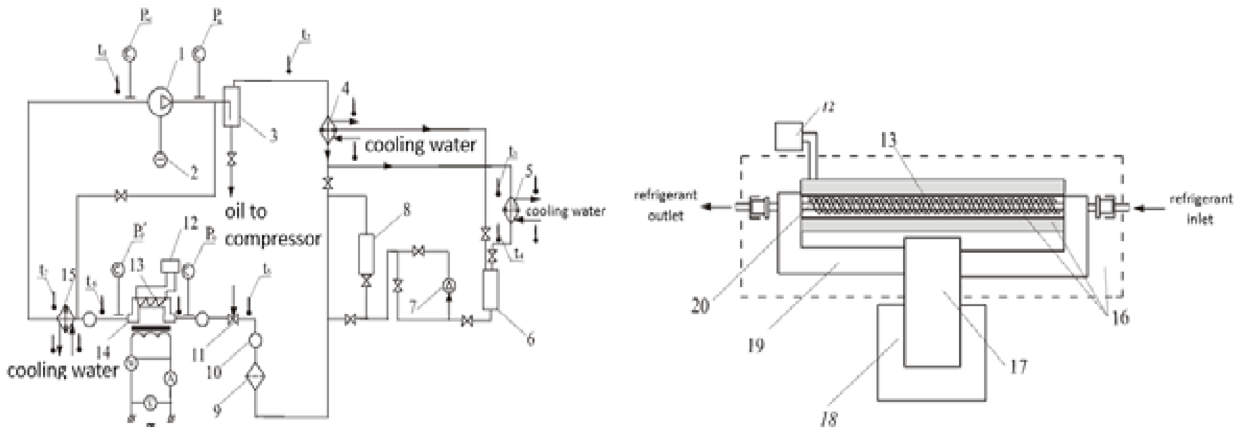
Equations (5) – (7) are more accurate compared to previously known expressions.

The obtained correlations were used in experiments investigating heat transfer during low-temperature boiling of refrigerant in a smooth horizontal tube.

Figure 1 shows the schematic diagram of the experimental setup.

The experimental setup (see Fig. 1) included a 2FV-4 compressor driven by a DC electric motor. The motor control system allowed smooth adjustment of the compressor shaft rotation frequency in the range of 2 to 50 s⁻¹ (120 to 3000 rpm), thereby regulating the compressor's capacity.

The experimental evaporator (see Fig. 1b) was a horizontal copper tube with a diameter of 16×1 mm and a length of 1 m, acting as the secondary short-circuited winding of a transformer. Electric current passed through the tube, heating it up, and the resulting heat was transferred to the refrigerant flowing inside.



a b

Fig. 1. Schematic diagram of the test facility (a) and structural diagram of the experimental evaporator (b):
1 – compressor; 2 – DC electric motor; 3 – oil separator; 4 – condenser; 5 – subcooler; 6 – calibrated tank; 7 – rotameter; 8 – oil concentration measurement device; 9 – filter-drier; 10 – sight glass; 11 – throttle valve; 12 – resistance meter; 13 – resistance thermometer; 14 – experimental evaporator; 15 – superheating evaporator; 16 – thermal insulation; 17 – magnetic core; 18 – primary winding of the transformer; 19 – short-circuited loop; 20 – copper tube – evaporator

The transformer's primary winding (see Fig. 1a) was connected to a voltage regulator through a wattmeter, ammeter, and voltmeter. By adjusting the supply voltage, the power – and thus the amount of heat transferred from the inner surface of the evaporator tube to the refrigerant – was controlled. The wattmeter measured the total active power consumed by the transformer, including magnetic and electrical losses. To account for these losses, the power consumption as a function of voltage was determined under both open-circuit and short-circuit conditions.

The temperature of the boiling refrigerant was determined using reference pressure gauges with scale divisions of 0.01 kgf/cm² and 0.025 kgf/cm² (corresponding to boiling temperatures of −37 °C and −18 °C, respectively) and cross-verified with laboratory thermometers.

The average temperature of the outer surface of the evaporator tube was measured using the resistance method, for which a wire resistance thermometer was wound around the tube. The specific heat flux was preliminarily calculated, set, and monitored via the wattmeter during the experiment.

Experimental Conditions:

- **Boiling temperature t_0, °C:**
  –18 and –37
- **Specific heat flux q_{Fa}, W/m²:**
  600; 1750; 3500; 5800; 8100; 10500
- **Refrigerant mass flow rate $Ga \cdot 10^3$, kg/s:**
  6.54; 13.5; 23.6 at t_0 = –18 °C
  6.67 at t_0 = –37 °C
- **Mass velocity $\rho_a w_a$, kg/(s·m²):**
  42.4; 87.7; 153.0 at t_0 = –18 °C
  43.3 at t_0 = –37 °C

The oil concentration in the refrigerant ranged from 4 to 6 % by volume (equivalent to 3–4 % by mass). The vapor quality of the refrigerant at the evaporator inlet was approximately 0.16 at t_0 = –18 °C, and about 0.21 at t_0 = –37 °C. The maximum vapor quality at the outlet of the experimental evaporator reached 0.65…0.70.

Based on the measurement results, the following parameters were determined:

1. Boiling t_0 (t_a) and condensation $t_к$ temperatures, determined using refrigerant saturation tables according to the absolute pressures at the inlet and outlet of the experimental evaporator and after the compressor.

The absolute pressure p_a of the refrigerant (in MPa) was calculated using the formula:

$$p_a = 0{,}0981(p + p_b / 735{,}6),$$

where p – measured pressure, in kg/cm²;

p_b – barometric pressure, in mmHg

2. Outer surface temperature of the evaporator tube, °C,

$$t_2 = \frac{R_h - R_c}{R_c}(235 + t_c) + t_c .$$

3. Heat transfer coefficient during refrigerant boiling inside the evaporator tube, in W/(m²·K).

To derive the final working formula, we substitute numerical values of the input variables into equation (7). Taking into account that

$$q_{Fa} = \frac{N - N_{\text{los}}}{\pi d_1 l} = 22{,}748(N - N_{\text{los}}),$$

we obtain:

$$\alpha_a = \frac{22{,}748(N - N_{\text{los}})}{(t_2 - t_a) - (1 + 0{,}005 t_2)(1 - \psi)200},$$

where

$$\psi = \frac{J_1(0{,}412 \cdot 10^{-2}\sqrt{N - N_{\text{los}}}) \cdot Y_0(0{,}361 \cdot 10^{-2}\sqrt{N - N_{\text{los}}}) - J_0(0{,}361 \cdot 10^{-2}\sqrt{N - N_{\text{los}}}) \cdot Y_0(0{,}412 \cdot 10^{-2}\sqrt{N - N_{\text{los}}})}{J_1(0{,}412 \cdot 10^{-2}\sqrt{N - N_{\text{los}}}) \cdot Y_0(0{,}412 \cdot 10^{-2}\sqrt{N - N_{\text{los}}}) - J_0(0{,}412 \cdot 10^{-2}\sqrt{N - N_{\text{los}}}) \cdot Y_0(0{,}412 \cdot 10^{-2}\sqrt{N - N_{\text{los}}})}$$

J_0, J_1, and Y_0 are Bessel functions of zero and first order of the first and second kind, respectively;

N, N_{los} – represent the total power consumption and transformer loss power of the experimental evaporator, in watts.

The relative error in determining the heat transfer coefficient ε_α varies from 2 % to 15 % depending on the heat flux density q_{Fa} and the refrigerant mass flow rate G_a. The highest error values (11–15 %) are observed at the lowest values of q_{Fa}. This can be explained by the fact that, at minimal q_{Fa} (600 W/m²), the relative error of the temperature difference $\varepsilon_{(t2-ta)}$ increases: while the absolute error $\Delta_{(t2-ta)}$ remains nearly constant, the temperature difference (t_2–t_a) becomes smaller. As q_{Fa} increases (from 1750 to 8000 W/m² and above), the value of ε_α decreases from 7.7 % to 2 %.

Figure 2 presents the experimental results as the dependence of the average heat transfer coefficient α_a on the heat flux density q_{Fa}.

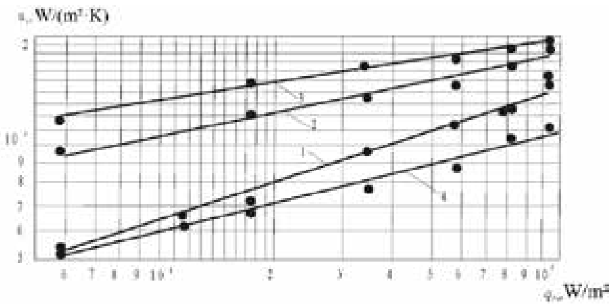


Fig. 2. Dependence of the average heat transfer coefficient α_a on the heat flux density q_{Fa}.
1 – $\rho_a w_a$ = 42.2 kg/(s·m²); 2 – $\rho_a w_a$ = 87.7 kg/(s·m²);
3 – $\rho_a w_a$ = 153 kg/(s·m²);4 – $\rho_a w_a$ = 43.3 kg/(s·m²);
Curves 1 to 3 correspond to a boiling temperature of –18 °C;
Curve 4 corresponds to a boiling temperature of –37 °C

An analysis of the obtained dependencies shows that, as the mass velocity ρawa increases from 42 to 153 kg/(s·m²) at a boiling temperature of –18 °C, the exponent in the power-law relation with q_{Fa} decreases from 0.35 to 0.18.

The degree to which the heat flux density influences heat transfer is related to changes in the heat transfer mechanism, which in turn are caused by restructuring of the vapor–liquid flow.

At $\rho_a w_a$ values between 42 and 88 kg/(s·m²) ($w_0 \leq 0.07$ m/s), a stratified flow regime is observed in the horizontal tube. Under these conditions, the heat transfer is mainly governed by nucleate boiling, which is strongly dependent on q_{Fa}.

At $\rho_a w_a$ = 153 kg/(s·m²), corresponding to $w_0 > 0.07$ m/s, a transitional (wavy-annular) regime is observed. In this case, heat transfer occurs partially due to convection, which reduces the sensitivity of the heat transfer coefficient α_a to changes in q_{Fa}.

At a boiling temperature of t_0 = –37 °C and mass velocity of 43 kg/(s·m²), the exponent in the power-law dependence on q_{Fa} is somewhat lower (0.255) compared to the same mass velocity at t_0 = –18 °C (0.35). This can be explained by the increase in refrigerant vapor quality at the evaporator inlet (from 0.16 to 0.21) as the boiling temperature decreases, due to deeper throttling. This changes the heat transfer mechanism: a larger portion of the tube surface is in contact with the vapor flow, which reduces the influence of q_{Fa} on the heat transfer coefficient.

The experimental results for heat transfer coefficients were compared with data from [1, 5] (see Fig. 3). The comparison was carried out at a refrigerant mass flow rate of G_a = 13.5 ·10-3 kg/s ($\rho_a w_a$ = 88 kg/(s·m²)), which corresponds to the flow conditions used in the present study. The data obtained by the authors agree satisfactorily (within a deviation of no more than 20 %) with the results of S.N. Bogdanov [1] and Bo-Pierre for the stratified and wavy flow regimes.

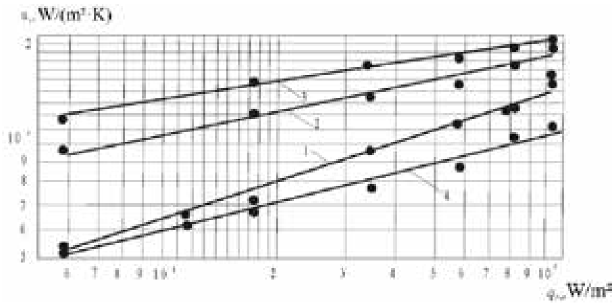


Fig. 3. Comparison of heat transfer coefficients:
▲ – according to S.N. Bogdanov [1];
■ – according to Bo-Pierre;
● – present study (authors' data)

To reasonably select equations describing heat transfer in the nucleate boiling region, the experimentally obtained values of average local heat transfer coefficients were compared with those calculated using equations (1), (2), and (3), (4). However, given that the experiments were conducted using a refrigerant containing 3–4 % oil by mass, while the correlations (1)…(4) were originally developed for pure refrigerants, it was necessary to assess the impact of oil on the heat transfer coefficient during boiling.

For this purpose, based on the analysis of experimental data from [6] obtained during boiling of refrigerant oil mixtures inside a horizontal copper tube in the stratified wavy flow regime the authors constructed plots (see Fig. 4, solid lines) showing the dependence of the ratio of the heat transfer coefficient for the refrigerant oil mixture to that of the pure refrigerant (α_{a+oil}/α_a) on heat flux density q_{Fa} and oil concentration ξ_o.

In the experiments described in [6], q_{Fa} ranged from 1 to 10 kW/m², and the oil concentration in each test series was 2 %, 3.5 %, and 7 % by mass.

According to the graphs (see Fig. 4, solid lines), at low heat fluxes (up to 1 kW/m²), the presence of oil in the refrigerant leads to a slight reduction in the heat transfer coefficient ($\alpha_{a+oil}/\alpha_a < 1$).

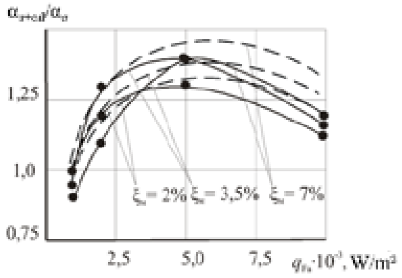
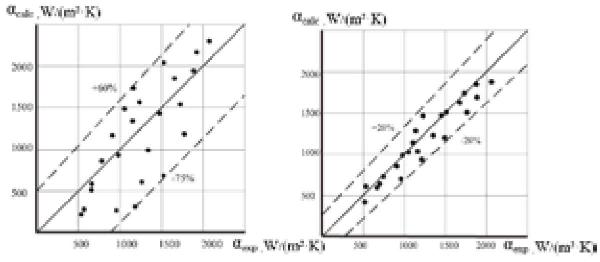


Fig. 4. Dependence of α_{a+oil}/α_a on q_{Fa} and oil concentration ξ_o during boiling of a refrigerant–oil mixture inside a horizontal copper tube in the stratified–wavy flow regime:
solid lines – experimental curves;
dashed lines – approximating curves



а б

Fig. 5. Comparison of experimental (αexp) and calculated (αcalc) heat transfer coefficients:
a – calculated using equations (1) and (2);
b – calculated using equations (3) and (4)

As the heat flux increases, the presence of oil in the refrigerant leads to an increase in the wetted surface area of the tube, which enhances heat transfer. The heat transfer coefficient rises accordingly, reaching its maximum at a heat flux of q_{Fa} = 5 kW/m², with an increase of 25–40 % compared to pure refrigerant.

With further increases in heat flux (up to 10 kW/m²), the heat transfer coefficient decreases slightly, yet still remains 10–20 % higher than that for the pure refrigerant.

The experimental curves (see Fig. 4, solid lines) were approximated by the following second-degree polynomial:

$$\alpha_{a+oil}/\alpha_a = (-0{,}2\xi_o - 14)10^{-3}q_{Fa}^{\,2} + (1{,}1\xi_o + 178)\,10^{-3}q_{Fa} + (0{,}03\xi_o + 0{,}72). \qquad (8)$$

In Fig. 4, the approximating curves are shown as three dashed lines corresponding to oil concentrations of 2 %, 3.5 %, and 7 %.

The approximation error ranges from 3 % to 10 %, with the highest error (10 %) observed at q_{Fa} = 10 kW/m².

The experimentally obtained average heat transfer coefficients were compared with those calculated using equations (1), (2), and (3), (4), modified using the correction from equation (8).

The comparison between the experimental (α_{exp}) and calculated (α_{calc}) heat transfer coefficients is presented in Figure 5.

As shown in Figure 5a, the discrepancy between the experimental data (α_{exp}) and the values of α_a calculated using equations (1) and (2) is quite substantial, ranging from +60 % to –75 %.

In contrast, the results calculated using equations (3) and (4) (see Fig. 5b) show a deviation not exceeding ±20 %, which can be considered reasonably acceptable.

Based on the above analysis, equations (3) and (4), in combination with the correction correlation (8), are recommended for calculating heat transfer during refrigerant boiling inside smooth horizontal copper tubes under stratified and wavy flow conditions, which are typical for the operation of low-temperature air coolers.

The proposed empirical and experimental correlations developed by the authors were applied in the design of hermetic refrigeration units for provision storerooms [4].

CONCLUSIONS

1. Experimental data were obtained for heat transfer coefficients during in-tube refrigerant boiling at low temperatures (–37 °C and –18 °C). Based on the results, appropriate equations were selected, and a semi-empirical correlation accounting for the influence of oil on boiling heat transfer was proposed. 2. The proposed equations are recommended for use in the design of low-temperature air coolers.

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