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IMPROVED METHOD OF CALCULATION OF AIR COOLERS FOR SHIP REFRIGERATION SYSTEMS

УДОСКОНАЛЕНИЙ МЕТОД РОЗРАХУНКУ ПОВІТРООХОЛОДЖУВАЧІВ СУДНОВИХ СИСТЕМ РЕФРИЖЕРАЦІЇ

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Abstract. The refined methodology of thermal calculation of refrigerant air coolers, taking into account the peculiarities of operation of ship refrigeration systems, is proposed.
Peculiarities of river and marine operating conditions of the Ukrainian fleet impose increased requirements to the refrigeration system providing optimal modes of perishable cargo transportation, storage of foodstuffs for passengers and crews of ships and creation of comfortable conditions in ship premises. The efficiency of refrigeration system operation largely depends on energy losses from external irreversibility caused by finite temperature differences in cooling devices – refrigerant air coolers, and the latter – on the intensity of heat transfer between the cooled air and boiling refrigerant.
The analysis of operating conditions of low-temperature air coolers, their design practice, results of theoretical and experimental studies of heat transfer in low-temperature air coolers has shown that one of the main trends in the improvement of low-temperature air coolers is: improvement of design and manufacturing quality through research work, improvement of calculation methods, application of new technologies.
The aim of the study is to summarising their own experiences data on heat transfer on the ribbed surface of air coolers and at in-tube low-temperature boiling of the refrigerant and refinement of the method of air coolers thermal calculation, taking into account peculiarities of operation of ship refrigeration systems.
Estimated dependencies for determination of heat transfer coefficient from air to tube-plate surface with fin spacing of 3...6 mm, typical for ship refrigeration systems. Based on the results of own experimental studies, the equation for determining the heat transfer coefficient at low-temperature boiling of refrigerant in horizontal tubes of shipboard refrigeration systems is specified and a semi-empirical dependence generalising the experimental data and taking into account the influence of oil on heat transfer at boiling is proposed. The proposed equations for calculation of external (from air to the external finned surface) and internal (at low-temperature boiling of the refrigerant) heat transfer are used in the refined methodology of thermal calculation of air coolers, taking into account the peculiarities of operation of ship refrigeration systems (increased fin spacing and reduced boiling temperature).
Key words: refrigerant air cooler; ship refrigeration systems; heat transfer; thermal calculation method.

Анотація. Запропоновано уточнену методику теплового розрахунку хладонових повітроохолоджувачів, що враховує особливості експлуатації суднових систем рефрижерації.
Особливості умов експлуатації річкового та морського флоту України висувають підвищені вимоги до системи рефрижерації, що забезпечує оптимальні режими перевезення вантажів, що швидко псуються, зберігання продуктів харчування для пасажирів та екіпажів суден та створення комфортних умов у суднових приміщеннях. Ефективність роботи систем рефрижерації багато в чому залежить від енергетичних втрат від зовнішньої незворотності, зумовлених кінцевими різницями температур в охолоджувальних приладах – хладонових охолоджувачах повітря, а останні – від інтенсивності теплоперенесення між повітрям, яке охолоджується і киплячим хлодоном.
Аналіз умов експлуатації низькотемпературних охолоджувачів повітря, практики їх проектування, результатів теоретичних та експериментальних досліджень теплообміну в низькотемпературних охолоджувачах повітря показав, що однією з основних тенденцій у їх вдосконаленні є: поліпшення якості проектування та виготовлення шляхом проведення дослідницьких робіт, удосконалення методів розрахунку, застосування нових технологій.

Метою дослідження є узагальнення власних дослідних даних по тепловіддачі на ребристій поверхні охолоджувачів повітря і при внутрішньотрубному низькотемпературному кипінні холодоагенту та уточнення методики теплового розрахунку повітроохолоджувачів, що враховує особливості експлуатації суднових систем рефрижерації. Запропоновано розрахункові залежності для визначення коефіцієнта тепловіддачі від повітря до трубно-пластинчастої поверхні з кроком ребер 3...6 мм, характерним для охолоджувачів повітря суднових систем рефрижерації. На основі результатів власних експериментальних досліджень уточнено рівняння для визначення коефіцієнта тепловіддачі при низькотемпературному кипінні хладону в горизонтальних трубках суднових охолоджувачах повітря та запропонована напівемпірична залежність, що узагальнює дослідні дані та враховує вплив масла на тепловіддачу при кипінні. Запропоновані рівняння для обчислення зовнішнього (від повітря до зовнішньої ребристої поверхні) та внутрішнього (при низькотемпературному кипінні холодоагенту) теплообміну використані в уточненій методиці теплового розрахунку охолоджувачів повітря, що враховує особливості експлуатації суднових систем рефрижерації (підвищений крок ребра і знижена температура кипіння).
Ключові слова: хладоновий охолоджувач повітря; суднові системи рефрижерації; тепловіддача; методика теплового розрахунку.

FORMULATION OF THE PROBLEM

Peculiarities of river and marine operating conditions of the Ukrainian fleet impose increased requirements to the refrigeration system (RS) providing optimal modes of perishable cargo transportation (in refrigerator and vegetable holds), storage of foodstuffs for passengers and crews of ships (in provision storerooms, commercial and household refrigerating tanks) and creation of comfortable conditions in ship premises (living, service, public). The performance of these ships as a whole largely depends on the reliable and efficient operation of RS [1, 4].

The efficiency of RS operation largely depends on energy losses from external irreversibility caused by finite temperature differences in cooling devices – refrigerant air coolers (AC), and the latter – on the intensity of heat transfer between the cooled air and boiling refrigerant. Intensification of heat transfer and, as a consequence, reduction of temperature difference, on the one hand, provide increase of boiling temperature and, accordingly, reduction of power consumption by compressors (at a given air temperature), and, on the other hand, decrease of air temperature (at unchanged boiling temperature), which makes it possible to reduce energy costs for its circulation, i.e. for the drive of air fans. In both cases, the reduction of power consumption contributes to the rational consumption of fuel and energy resources by the ship as a whole and by the ship power plants in particular [7].

The analysis of operating conditions of low-temperature air coolers (LTAC), their design practice, results of theoretical and experimental studies of heat transfer in LTAC has shown that one of the main trends in the improvement of LTAC is: improvement of design and manufacturing quality through research work, improvement of calculation methods, application of new technologies [2].

ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

In [5, 8, 9] the results are presented experimental studies of heat transfer on the external surfaces of the AC, where the dependence of the heat transfer coefficient on the fin spacing and the velocity of the impinging air flow has been experimentally established. However, in order to be able to apply these results in design practice, it is necessary to present them in the form of calculated dependences.

PURPOSE OF THE STUDY

Summarising their own experiences data on heat transfer on the ribbed surface of AC and at in-tube low-temperature boiling of the refrigerant and refinement of the method of AC thermal calculation, taking into account peculiarities of operation of ship refrigeration systems.

MAIN MATERIAL

Heat transfer at external flow around the outer surface of finned tubes with continuous plate finning was described by the criterion equation in the form

$$Nu_{air} = C Re_{air}^{m} Pr_{air}^{n}, \quad (1)$$

where C is a constant; m, n – degree indices; $Nu_{air}=\alpha_p d_e/\lambda_v$ – Nusselt criterion; $Re_{air}=w_{air}d_e/\nu_{air}=w_{air}\rho d_e/\mu_{air}$ – Reynolds criterion; $Pr_{air}=\mu_{air}C_r/\lambda_{air}$ – Prandtl criterion; α_p – heat transfer coefficient referred to the total external surface of the AC, W/(m^2·K); λ_v – air thermal conductivity coefficient, W/(m·K); w_{air} and ($w_{air}\rho_{air}$) – velocity and mass velocity of air in the living section of the AC (between ribs and tubes), m/s and kg/(m^2·s); ν_{air} and μ_{air} – coefficients of kinematic and dynamic viscosity of air, m^2/s and Pa·s; d_e – equivalent diameter of the living section, m

Fig. 1 shows the dependence of Nu on Re at different d_e (S_p) for the tested ACs. We have generalized the results of investigations for surfaces with fins pacing of 3, 4.3, 5.3 and 6.2 mm (d_e is $4.72\cdot10^{-3}$, $6.44\cdot10^{-3}$, $7.61\cdot10^{-3}$, $8.57\cdot10^{-3}$ m, respectively), typical for ship refrigeration systems. As a result, the following values of degree in dices for Re, Pr and c were obtained: $m = 0.65$; $n = 0.4$; $c = 0.155$.

Substituting instead of the similarity criteria in (1) the corresponding physical quantities and expressions for c, m and n, we obtain in the final form

$$\alpha_p = 0{,}155\frac{\lambda(\rho w)^{0,65}}{d_e^{0,35}\mu^{0,65}}Pr^{0,4} \quad (2)$$

Fig. 2,a shows the comparison of heat transfer coefficients calculated by expression (2) $\alpha_{p.c}$ and obtained from the experiment $\alpha_{p.ex}$. The figure shows that the scatter does not exceed ±10%.

In [3, 6], analytical and experimental studies were carried out to determine the heat transfer coefficient α_a for in-line refrigerant boiling, the equation for calculating α_a was selected, and the dependence of α_a on q_{Fa} and oil concentration ξ was shown. However, when calculating the AC of refrigeration systems, it is necessary to reflect analytically the dependence of α_a on ξ and to present in a final form the equation by which α_a should be determined.

These curves were approximated by the equation in the form of a polynomial of the second degree

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

In Fig. 3,*a* approximating curves are plotted as three dashed lines for oil concentrations of 2; 3.5 and 7 %. The approximation error ranges from 3 to 10 %, with the larger error value (10 %) corresponding to q_{Fa} = 10 kW/m^2.

After obtaining the analytical dependence $\alpha_{a+oil}/\alpha_a = f(q_{Fa}, \xi)$, the calculation of heat transfer at in-line boiling is carried out as follows. Calculate the

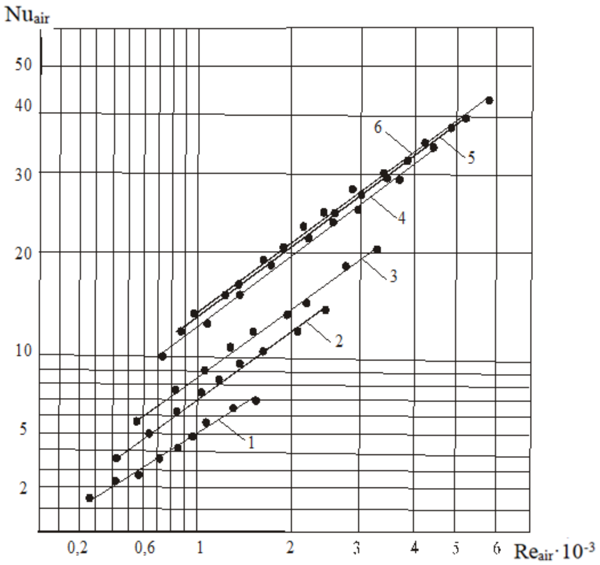

Fig. 1. Dependence of Nu_{air} on Re_{air} at different $d_e(S_p)$:
1 – $d_e = 2.2\cdot10^{-3}$ m (S_p= 1.4 mm); 2 – $3.53\cdot10^{-3}$(2.2);
3 – $4.72\cdot10^{-3}$(3.0); 4 – $6.64\cdot10^{-3}$(4.3); 5 – $7.61\cdot10^{-3}$(5.3);
6 – $8.57\cdot10^{-3}$ m (6.2 mm)

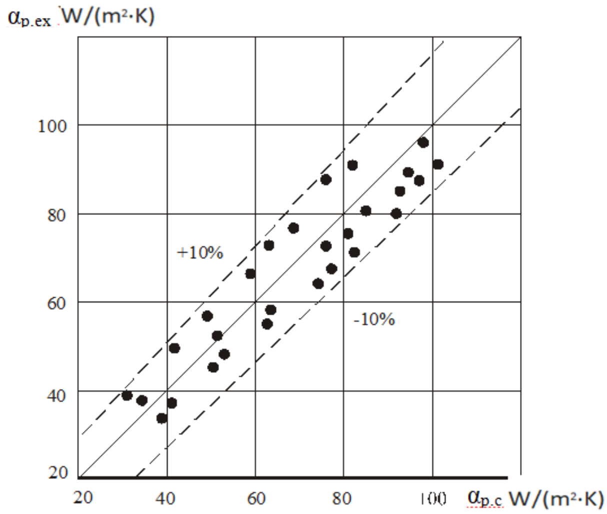

Fig. 2. Comparison of experimental $\alpha_{p.ex}$ and calculated $\alpha_{p.c}$ values heat transfer coefficients

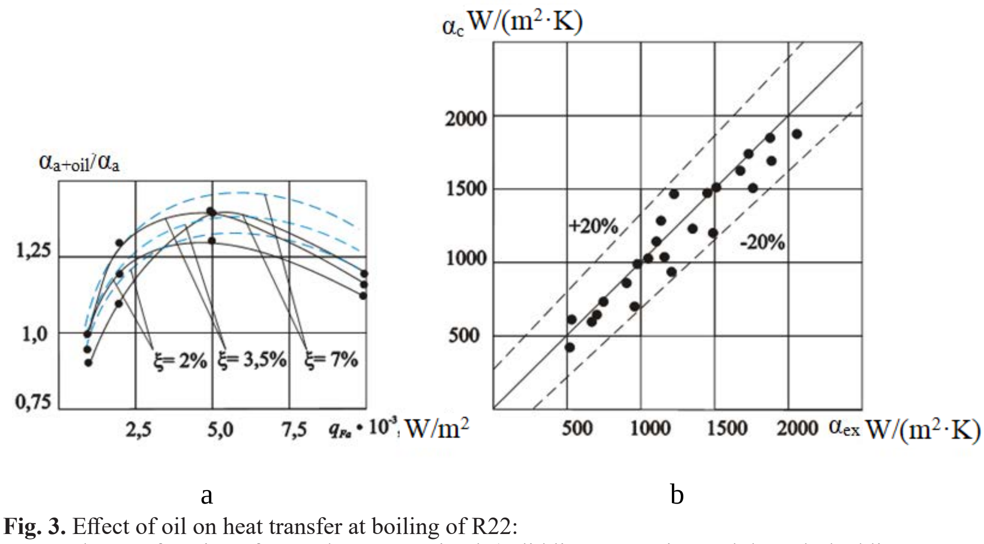

Fig. 3. Effect of oil on heat transfer at boiling of R22:
a – α_{a+oil}/α_a as a function of q_{Fa} and concentration ξ (solid line – experimental data; dashed line – according to equation (5.7)); b – comparison of α_c and α_{ex}

heat transfer coefficient α_a for pure refrigerant according to the formula of N.D. Danilova taking into account the convective component

$$\alpha_a = \alpha_{l.v.}(1 + 1{,}7N), \qquad (4)$$

where $\alpha_{l.v}$ is the heat transfer coefficient at boiling in a large volume,
$\alpha_{l.v.} = 7.5\, q_{Fa}^{0.7}(0.14 + 2.2\, P_o/P_{cr})$;
P_{cr} – critical pressure, Pa.

$N = Bo/K_r$ – complex, taking into account the convective component;

$Bo = w\rho r/q_{Fa}$; $K_r = p_0 b/\sigma$; r – heat of vapour formation, J/kg; p_0 – boiling pressure, Pa; σ – surface tension, N/m; g – acceleration of gravity, m/s; ρ', ρ'' – density of liquid and vapour refrigerant, kg/m^3;

$$b = \sqrt{\frac{\sigma}{g(\rho' - \rho'')}}$$

Then, given the value of q_{Fa} and ξ_m, using equation (3) calculate the value of the ratio $\alpha_{a+oil}/\alpha_a = K_m$ and find the heat transfer coefficient taking into account the influence of oil

$$\alpha_{a+\text{ oil}} = \cdot K_m\, \alpha_{b.o.}(1 + 1{,}7N), \qquad (5)$$

Fig. 3,*b* presents the results of comparison of experimental values of heat transfer coefficients α_{ex} and calculated α_c according to the above method. As can be seen from the graph, the deviations of individual experimental and calculated values of heat transfer coefficients at boiling do not exceed ±20%, which can be considered quite satisfactory.

The methodology of thermal calculation of low-temperature AC is based on the following basic provisions and assumptions:

1. The problem is stationary: $(\partial t/\partial \tau) = 0$, $(\partial q/\partial \tau) = 0$.
2. The boiling temperature of the refrigerant t_0 along the length of the pipe remains constant, i.e. $t_{0(L)} = \text{const}$.
3. Superheating of the refrigerant vapour Δt_v in AC relative to the liquid is assumed to be equal to zero, i.e. $\Delta t_v = 0$.

The boundary conditions of the third kind were considered, i.e. at known cooling capacity Q_0, air temperature t_{air1}, boiling temperature t_0, heat transfer laws (dependences for calculation of linear heat flux densities, temperature heads on heat conducting layers, heat transfer coefficients on the refrigerant side α_a and air side α_{air}) the heat exchange surface F_s (at full calculation) or the value of air temperature at the outlet (at verification calculation) was determined.

Input parameters: Q_0, t_{air1}, φ_{air1}, t_0, t_c, d_1, d_2, S_1, S_2, S_p, δ_p, δ_i, n_1, β.

Output parameters: F_s, L_p, n_2.

Boundary conditions: $(\partial t/\partial \tau) = 0$; $t_{0\,(x=0)} = t_0$, $t_{0\,(x=L)} = t_0$.

The heat transfer scheme in the air cooler is shown in Fig. 4. To simplify the calculation, the direct flow of heat-exchanging media (air and refrigerant) is assumed. The following order of air cooler calculation is accepted.

The diagram of heat exchange in the air cooler is shown in Fig. 4. To simplify the calculation, the direct flow of heat-exchanging media (air and refrigerant) is assumed. The following order of air cooler calculation is adopted.

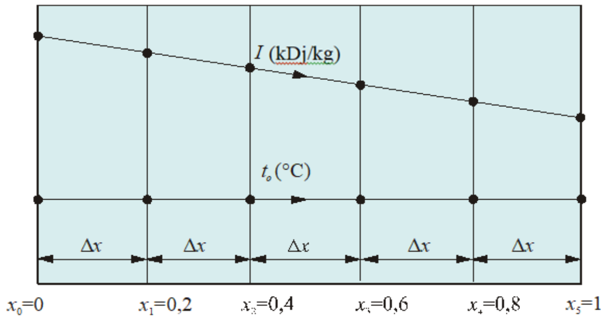

Fig. 4. Calculated diagram of heat transfer in the air cooler

The AC by heat input is divided (see Fig. 4), for example, into 5 equal sections Δ*x* (the more sections, the more accurate the calculation) each with heat input $\Delta x = Q/5$. Calculation is performed at relative values of heat removal: $x_0 = 0$ (air inlet); $x_1 = 0{,}2$; $x_2 = 0{,}4$; $x_3 = 0{,}6$; $x_4 = 0{,}8$; $x_5 = 1$ (air outlet). At the boundary of each section x_0, x_1, x_2, x_3, x_4 and x_5 (see Fig. 4), make expressions for linear heat flux densities separately for each heat conducting layer through its conductivity and temperature head:

$$q_{air} = (\alpha_p / C_v)\, F_{s.ef}\, (I - I_s),\ \ I_s = At_s^2 + Bt_s + C,$$

$$q_f = \frac{2\pi\lambda_f}{\ln\dfrac{d_2 + 2\delta_f}{d_2}}\theta_f,\ q_t = \frac{2\pi\lambda_t}{\ln\dfrac{d_2}{d_1}}\theta_t,\ \ q_a = K_m\alpha_a\pi d_1\theta_a,$$

$$t_{ws} - t_0 = \theta_f + \theta_t + \theta_a,$$

where (in addition to the already known designations) q_{air}, q_f, q_t, q_a – respectively linear densities of heat fluxes (fluxes related to the unit length of the tube) of air to the effective external surface $F_{s.ef}$, through the layer of frost, tube wall, from the inner wall of the tube to the refrigerant, W/m; t_{ws} – temperature of the wetted surface of AC, °C; $I_{s.air}$ – enthalpy of saturated air corresponding t_{ws}, kJ/kg; *A*, *B*, *C* – coefficients of the interpolation polynomial of the second degree $I_{s.air} = f(t_{ws})$; θ_f, θ_t, θ_a – temperature heads between the heat conducting layers, °C.

Equating linear heat flux densities of separate heat-conducting layers $q_{air} = q_f = q_t = q_a = q$ and adding to them the equation of balance of temperature heads $t_{ws} - t_0 = \theta_f + \theta_t + \theta_a$, a system of equations for determination of linear heat flux density at known enthalpy of the cooled air and boiling temperature at the considered boundary of sections is obtained.

Expressions for linear densities of heat fluxes on the air side and on the refrigerant side are non-linear. On the air side – because the heat transfer accompanied by moisture precipitation is determined by the difference of enthalpies of air in the flow and at the wall (wetted) layer. On the refrigerant side – since the heat transfer coefficient is not a constant value, but depends on the temperature head between the tube wall and the boiling refrigerant. Therefore, the method of successive approximations is applied.

The area of the external surface F_{sL}, related to 1m of the AC tube length, the equivalent diameter of the intercostal gap d_e, the blockage coefficient K_{bl}, the mass air velocity in the gap $\rho_{air}w_{air}$, the rib efficiency coefficient E_r, and the effective area of the external surface $F_{n.ef.}$ are determined according to the known formulas. According to the given cooling capacity Q_0, the refrigerant flow rate G_a and its mass velocity $\rho_a w_a$ are calculated.

Heat transfer coefficients on the air side are determined by formula (2). Heat transfer coefficient on the refrigerant side, taking into account oil, is calculated according to the formula of G.N. Danilova (4) and formulas (3), (5).

Since the above dependences include the value of linear heat flux density q, which is unknown in advance, the calculation procedure includes iterative cycles for q. At the same time, surface temperatures, air temperature decrease and moisture content change are determined for each heat removal section. Then, the total length of heat-exchange tubes of AC is determined by the numerical integration method.

CONCLUSIONS

1. Estimated dependencies (in the criterion-based form and in dimensional form) for determination of heat transfer coefficient from air to tube-plate surface with fin spacing of 3...6 mm, typical for ship refrigeration systems. The discrepancy between calculated and experimental data does not exceed ±10 %. 2. Based on the results of own experimental studies, the equation for determining the heat transfer coefficient at low-temperature boiling of refrigerant in horizontal tubes of shipboard refrigeration systems is specified and a semi-empirical dependence generalising the experimental data and taking into account the influence of oil on heat transfer at boiling is proposed. 3. The proposed equations for calculation of external (from air to the external finned surface) and internal (at low-temperature boiling of the refrigerant) heat transfer are used in the refined methodology of thermal calculation of AC, taking into account the peculiarities of operation of ship refrigeration systems (increased fin spacing and reduced boiling temperature).

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