DOI https://doi.org/10.15589/znp2022.1(488).4
УДК 621.51:628.84

MATHEMATICAL MODEL AND METHOD FOR CALCULATING THE TEMPERATURE LEVEL OF THE HERMETIC COMPRESSOR UNIT

МАТЕМАТИЧНА МОДЕЛЬ ТА МЕТОД РОЗРАХУНКУ ТЕМПЕРАТУРНОГО РІВНЯ ГЕРМЕТИЧНОГО КОМПРЕСОРНОГО АГРЕГАТУ

Olena V. Lytosh
Ph. D., Associate Professor
olena.lytosh@nuos.edu.ua
ORCID: 0000-0002-8109-4680

О. В. Литош,
канд. техн. наук, доцент

Admiral Makarov National University of Shipbuilding, Mykolaiv
Національний університет кораблебудування імені адмірала Макарова, м. Миколаїв

Abstract. The mathematical model and calculations method of temperature level of hermetic compressor unit (HCU) for marine self-contained air conditioners with calculation of condition work and of construction machine give an accounted.
Existing methods for calculating the temperatures of HCU elements (refrigerant, lubricating oil, parts and assemblies) using mathematical models do not allow the creation and improvement of modern HCU shipboard ACE, since they do not take into account their specific features. Literature sources show the results obtained mainly for small commercial HCUs or large air compressors. All this complicates the calculation of the temperature level of high-speed ship HCUs and requires additional research. A mathematical model and a method for calculating the temperature level of a ship high-speed HCU of the HGV type have been developed the author, which make it possible to assess their thermal state, determine the temperatures of the compressor elements, refrigerant and lubricating oil, depending on the operating conditions and design parameters of the machine.
The mathematical model and method for calculating the temperature level of the HCU is based on the certain basic provisions and assumptions. The procedure has been adopted for calculating the HCU temperature level. To check the adequacy of the mathematical model, the results of calculating the temperature level were compared HCU with empirical data. Calculated and experimental values of the temperatures of the main elements HCU, refrigerant and oil differ on average by 2…3 °C, maximum – 6…7 °C (deviation does not exceed 5…6 %).
Comparison of the results of calculation and experiment allows us to conclude that the developed mathematical model is adequate to the processes under study and can be recommended for implementation in the design practice of high-speed HCUs.
Using the developed method, the following studies were carried out: the rational thickness of the insulation of the discharge tract was determined HCU, various options of the cooling system were considered, the influence of the color of the outer surface of the casing was evaluated HCU to the temperature level of the machine. As a result, it has been found that cooling the HCU casing with water from the air conditioner evaporator, which is fed into a sump covering only the lower part of the casing, reduces the temperature level by 10…25 °C. 3. It has been shown that painting the HCU casing with black matte varnish reduces its temperature level by 6…7 °C (compared to current yellow color), and the rational thickness of the insulation of the discharge path is 2…2,5 mm.
Key words: hermetically compressor unit, temperature level, thickness of the insulation of the discharge tract, mathematical model, shipboard air conditioning equipment.

Анотація. Представлено математичну модель і метод розрахунку температурного рівня герметичного компресорного агрегату (ГКА) для суднових автономних кондиціонерів з урахуванням умов роботи та конструктивних параметрів машини.
Наявні методи розрахунку температур елементів ГКА (холодоагенту, мастила, деталей і вузлів) з використанням математичних моделей не дають можливість створювати й удосконалювати сучасні ГКА суднового ОКВ, тому що не враховують їхні специфічні особливості. У наукових джерелах наведено результати, отримані переважно для малих ГКА торгового призначення або великих повітряних компресорів. Усе це ускладнює розрахунок температурного рівня високооборотних суднових ГКА та потребує проведення додаткових досліджень.

Автором розроблені математична модель і метод розрахунку температурного рівня суднового високооборотного ГКА типу ХГВ, що дають змогу оцінити їхній тепловий стан, визначити температури елементів компресора, холодоагенту та мастила залежно від умов роботи і конструктивних параметрів машини.
Математична модель та метод розрахунку температурного рівня ГКА базуються на деяких основних положеннях і припущеннях. Утверджено порядок розрахунку температурного рівня ГКА. Для перевірки адекватності математичної моделі зіставлено результати розрахунку температурного рівня ГКА з дослідними даними. Розрахункові й експериментальні значення температур основних елементів ГКА, холодоагенту та мастила відрізняються в середньому на 2…3 °C, максимальні – на 6…7 °C (відхилення не перевищує 5…6 %).
Зіставлення результатів розрахунку та експерименту дає змогу зробити висновок, що розроблена математична модель адекватна досліджуваним процесам і може бути рекомендована до впровадження у практику проєктування високооборотних ГКА.
За допомогою розробленого методу виконано такі дослідження: визначено раціональну товщину ізоляції нагнітального тракту ГКА, розглянуто різні варіанти системи охолодження, оцінено вплив кольору зовнішньої поверхні кожуха ГКА на температурний рівень машини. У результаті встановлено, що охолодження кожуха ГКА водою з випарника кондиціонера, яка подається в піддон, що охоплює тільки нижню частину кожуха, дає можливість знизити температурний рівень на 10…25 °C. Визначено, що фарбування кожуха ГКА чорним матовим лаком дає змогу знизити його температурний рівень на 6…7 °C (порівняно з нинішнім фарбуванням у жовтий колір), а раціональна товщина ізоляції нагнітального тракту становить 2…2,5 мм.
Ключові слова: герметичний компресорний агрегат, температурний рівень, товщина ізоляції нагнітального тракту, суднове обладнання кондиціювання.

FORMULATION OF THE PROBLEM

Reliability of high-speed hermetic compressor unit (HCU) shipboard air conditioning equipment (ACE), as well as their volumetric and energy indicators to a large extent depend on the temperature level, which is usually determined empirically in the process of creating and improving HCU. There are practically no reliable methods for calculating the temperatures of HCU elements, refrigerant and lubricating oil, and practical recommendations apply only to machines similar to those tested. Due to the peculiarities of the design forms and operating conditions of ship HCU, the principles of their design differ significantly from those adopted for open (stuffing box) machines and require their own development [8].

ANALYSIS OF RECENT RESEARCH AND PUBLICATIONS

Recently, there have been no significant publications on the development and use of mathematical models and cybernetic methods for research, design, adjustment and measurement of processes in the creation and improvement of HCU. Such publications date back to 70–80 years of the last century. This can be explained by the fact that foreign HCU manufacturers, developing and using mathematical models for research, optimization and calculation of compressors, have stopped publishing on this topic or publish descriptive (advertising) articles without setting out the main content of the methodological material concerning the essence of the mathematical models they use [1; 6; 7].

In domestic textbooks, manuals and journals, including those published in recent years, descriptions of a number of modern HCUs and some of their characteristics are given, but, unfortunately, there are no design recommendations [4; 5; 7; 8].

Existing methods for calculating the temperatures of HCU elements (refrigerant, lubricating oil, parts and assemblies) using mathematical models do not allow the creation and improvement of modern HCU shipboard ACE, since they do not take into account their specific features. Literature sources show the results obtained mainly for small commercial HCUs or large air compressors. All this complicates the calculation of the temperature level of high-speed ship HCUs and requires additional research [1; 6; 8].

Purpose of the study is a presentation of the mathematical model and method for calculating the temperature level of a high-speed HCU marine ACE depending on design and operating parameters.

Main material. The mathematical model and method for calculating the temperature level of the HCU is based on the following basic provisions and assumptions:

1. Stationary problem: $(\partial q/\partial \tau) = 0$, $(\partial t/\partial \tau) = 0$.
2. The thermodynamic system is homogeneous, i. e. gas pressure at all points of the cavities of the compressor unit on the suction side (p_{suc}) и discharge (p_{dis}) at every moment of time constantly.
3. Suction pressure and temperature (p_{suc}, t_{suc}) and pressure discharge (p_{dis}) constant, i. e. p_{suc} = const, t_{suc} = const, p_{dis} = const.
4. Simulated processes are reversible and equilibrium.
5. The continuity of the environment is observed, i. e. l / L <<1 (here l – molecule mean free path, L – characteristic system size).
6. The change in the potential and kinetic energy of the gas is negligible.
7. The simultaneous change in the parameters of the refrigerant is observed throughout the volume.

Boundary conditions of the third kind were considered, i. e. with known performance G_0, G_1, input (p_{suc}, t_{suc}) and output (p_{dis}) the parameters of the refrigerant, the values of the contact surfaces of the HCU elements, refrigerant and oil, the laws of conservation of energy and heat transfer, the numerical values of the temperatures of the HCU elements, refrigerant and oil were determined by the joint solution of the heat balance and heat transfer equations.

Settlement and constructive diagram of a sealed compressor unit and a diagram of energy flows in it are shown in figure 1.

The equation for calculating temperatures is

$$C_i \frac{dT_i}{d\tau} = \sum_k^i Q_{ki} + \sum_l^i Q_{li}\,, \tag{1}$$

whare C_i – heat capacity of element material HCU; T_i – element temperature; τ – time; Q_{ki} – the amount of heat transferred to an element k thermal conductivity or heat transfer; Q_{li} – the amount of heat given up (received) by the refrigerant l, passing through the element HCU.

This equation describes heat fluxes during non-stationary operating modes of the HCU, which occur, in particular, during its cyclical operation in automated installations. When designing an HCU, first of all, it is necessary to determine the maximum values of the temperatures of the HCU elements, oil and refrigerant in the steady state, on which the HCU's performance significantly depends.

In the case of calculating the steady-state thermal mode of the SCU

$$dt/d\tau = 0,\ C(dt/d\tau) = 0$$

and equation (1) is simplified

$$\sum_k^i Q_{ki} + \sum_l^i Q_{li} = 0\,.$$

In accordance with the diagram of energy flows in the HCU (see fig. 1, b) for the main elements of the HCU: the casing, the electric motor, the block crankcase, the cylinder, the shaft, the discharge cavity, the receiver, the discharge tube, the lubricating oil and for the states of the refrigerant: the casing, the electric motor, the cylinder, the discharge cavity, the receiver and the discharge tube, – the balance equations of the incoming and outgoing energy flows have been drawn up. The number of basic elements and balance equations is determined, on the one hand, by the need to take into account all the main heat fluxes, and, on the other hand, by the possibility of determining with sufficient accuracy the corresponding geometric parameters and heat transfer coefficients.

The balance equations describing the thermal processes in the HCU in the steady state have the form for:

Block crankcase

$$Q_{em,\,bc^{\cdot}} + Q_{c,\,bc} - Q_{bc,\,2} \pm Q_{bc,\,3} = 0; \tag{2}$$

Electric motor

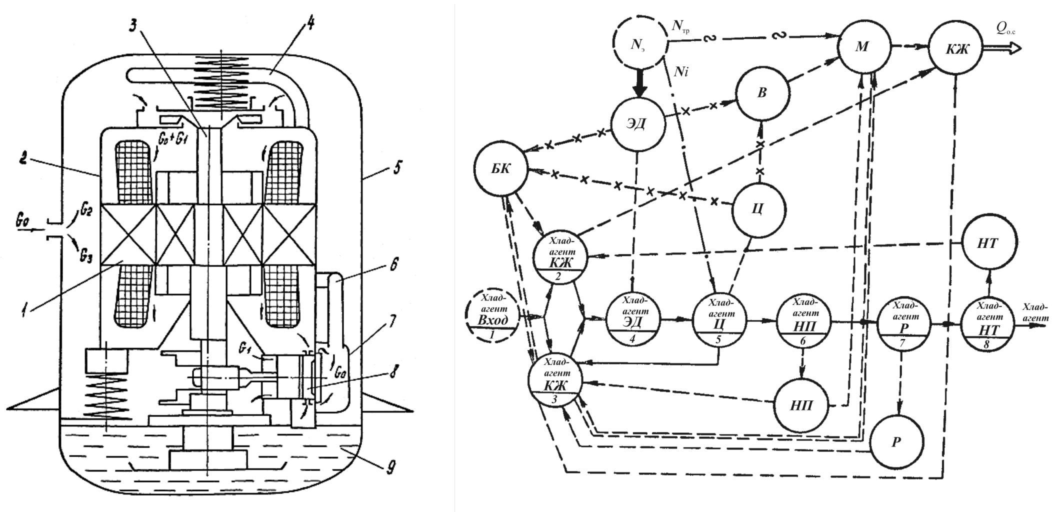
$$Q_{em.^{\cdot}} - Q_{em,\,bc} - Q_{em,\,s} - Q_{em,\,4} = 0; \tag{3}$$

Shaft

$$Q_{em,\,s^{\cdot}} + Q_{c,\,s} - Q_{s,\,o} = 0; \tag{4}$$

Discharge tube

$$Q_{8,\,dt} - Q_{dt,\,2} = 0; \tag{5}$$



a *b*

Fig. 1. Design and design diagram of HCU (***a***) and diagram of energy flows in it (***b***):
1 – electric motor; 2 – block crankcase; 3 – shaft; 4 – discharge tube; 5 – casing; 6 – receiver; 7 – discharge cavity; 8 – cylinder; 9 – oil;
$N_{el.m.}$, N_{fr}, N_i – respectively, the power consumed by the electric motor, friction and indicator; Q_e – heat release to the environment; EM – electric motor; BC – block crankcase; S – compressor unit shaft; O – crankcase oil; CH – compressor unit housing; C – cylinder; DC, DT – discharge cavity and the tube; R – receiver; 1…8 – condition DRP in the corresponding elements compressor unit

Casing

$$Q_{2,c\cdot} + Q_{3,c} + Q_{o,\,c} - Q_e = 0; \tag{6}$$

Receiver

$$Q_{7,\,r} - Q_{r,\,3} = 0; \tag{7}$$

Cylinder

$$Q_{5,\,c\cdot} - Q_{c,\,s} - Q_{c,\,bc} = 0; \tag{8}$$

Discharge cavity

$$Q_{6,\,dc\cdot} - Q_{dc,3} - Q_{dc,\,o} = 0; \tag{9}$$

Oil

$$Q_{s,.o\cdot} + Q_{dc,\,o} + N_{fr} \pm Q_{o,\,3} - Q_{o,\,c} = 0; \tag{10}$$

refrigerant at the top casing

$$G_2 i_1 + Q_{bc,\,2\cdot} + Q_{dt,\,2} - Q_{2,\,c} - G_2 i_2 = 0; \tag{11}$$

at the bottom of the casing

$$G_3 i_1 + G_1 i_{np} \pm Q_{bc,\,3\cdot} \pm Q_{o,\,3} + Q_{dc,\,3\cdot} + Q_{r,\,3} - Q_{3,\,c} - (G_1 + G_3) i_3 = 0; \tag{12}$$

In the electric motor

$$(G_0 + G_1) i_{fr} + Q_{em,\,4} - (G_0 + G_1) i_4 = 0; \tag{13}$$

In the cylinder

$$(G_0 + G_1) i_4 + N_i - Q_{5,\,c} - G_1 i_{pc} - G_0 i_5 = 0; \tag{14}$$

In the discharge cavity

$$G_0 i_5 - Q_{6,\,dc} - G_0 i_6 = 0; \tag{15}$$

In the receiver

$$G_0 i_5 - Q_{7,\,r} - G_0 i_7 = 0; \tag{16}$$

In the discharge tube

$$G_0 i_7 - Q_{8,\,dt} - G_0 i_8 = 0, \tag{17}$$

where Q – heat flux (letters and numbers in the indices correspond to the designations adopted in fig. 1, ***b***);
$i_1 \ldots i_8$ – enthalpy of the refrigerant in the respective elements HCU;
i_{fr}, i_{pc} – enthalpy of the coolant in the piston-cylinder clearance and in front of the electric motor.

The amount of heat released in the electric motor was found by the formula

$$Q_{em} = (1 - \eta_{em}) N_e, \tag{18}$$

where η_{em} – electric motor efficiency; N_e – the power consumed by the electric motor. Heat flow between two HCU elements (k and n)

$$Q_{kn} = \frac{\lambda}{\delta_{kn}} F_{kn} \left(t_k - t_n\right), \tag{19}$$

where λ – coefficient of thermal conductivity; $\delta_{кn}$ – distance between geometric centers of elements k and n; F_{kn} – heat exchange surface between elements k and n; t_k and t_n – average element temperatures k and n.

Heat flow from the refrigerant to the element, for example n, compressor unit

$$Q_{ln} = \alpha_{ln} F_{ln} (t_l - t_n), \tag{20}$$

where α_{ln} – heat transfer coefficient from gas to element HCU; F_{ln} – heat exchange surface between refrigerant and HCU element; t_l – average temperature of the refrigerant.

The amount of heat given up (received) by the refrigerant passing through the HCU element,

$$Q_{l\,i} = G_i (i_l - i_i), \tag{21}$$

where G_i – mass, refrigerant flow; i_l, i_i – enthalpy of the refrigerant at the outlet and inlet of the element.

The values of temperatures (enthalpies) of the refrigerant, oil, HCU elements, mass flow rates of the refrigerant, power, losses to the environment included in equations (2)…(21) were determined empirically [2].

The heat transfer coefficients from the coolant to the HCU elements, from them to the coolant and oil in the crankcase were found by the formula

$$\alpha_{l,n} = \frac{Q_{l,n}}{F_{l,n}\left(t_l - t_n\right)}. \tag{22}$$

Heat flows Q_{ln}, included in expression (22), were calculated from the heat balance equations (2)…(21).

Figure 2 shows an enlarged block diagram of the iterative calculation of the temperatures of HCU elements, refrigerant and oil based on the algorithm for solving the system of balance equations (2)…(22).

Initial data for the calculation:

design parameters: p_{suc}, t_{suc}, p_{dis}, $F_{em\text{-}bc}$, $F_{em\text{-}s}$, $F_{em\text{-}drp}$, $F_{c\text{-}bc}$, $F_{bc\text{-}drp}$, $F_{c\text{-}s}$, $F_{s\text{-}o}$, $F_{c\text{-}drp}$, $F_{o\text{-}c}$, $F_{drp\text{-}c}$, $F_{em\text{-}bc}$, $F_{dc\text{-}o}$, $F_{o\text{-}drp}$, $F_{dc\text{-}drp}$, $F_{r\text{-}drp}$, $F_{dt\text{-}drp}$, c_{p1}, c_{p2};

experimental data: η_{em}, N_e, G_0, G_1, Q_e, N_{fr}, N_i, $\alpha_{l,n}$.

As a result of the calculation, it is determined: G_2, G_3, t_{em}, t_{bc}, t_s, t_c, t_o, t_c, t_{dc}, t_r, t_{dt}, $t_{drp\text{-}2} \ldots t_{drp\text{-}8}$.

The following procedure has been adopted for calculating the HCU temperature level (see fig. 2).

In block 1, design parameters, experimental data, initial conditions, operating mode, counting step and other numerical material are entered. The calculation of the individual constants required in the calculation process is performed in block 2. In block 3, there is an increase and calculation of the amount of refrigerant flow rates. The determination of the coefficients included in the equations of thermal processes is carried out in block 4. Block 5 calculates the temperatures of the HCU elements, refrigerant and oil.

The HCU thermal balance check is carried out in block 6

$$\Delta_Q = Q_e + G_0\, c_{p2} t_8 - G_0\, c_{p1} t_1 - N_e,$$

where G_0 – refrigerant mass flow; c_{p1}, c_{p2} – suction and discharge refrigerant heat capacity; t_1, t_8 – temperatures before the suction and after the discharge pipes; N_e – power consumed by HCU; Q_e – heat given to the environment; Δ_Q – unbalance value.

In block 7, the condition is checked $\Delta_Q \le \Delta_{Q.per}$. Here $\Delta_{Q.per}$ = (A) – value of permissible unbalance. When this condition is met, the calculation is considered complete and the results are printed. Otherwise, the new value of G_2 is accepted and the calculation is repeated. To check the adequacy of the mathematical model, the results of calculating the HCU temperature level were compared with experimental data.

The adequacy of the model was checked in various operating modes of the HCU, with a change in the diametral clearance 2Δ between the piston and the cylinder and environmental conditions. In fig. 3 shows the temperature values in the compressor unit FGV-2,2. It can be seen from the figures that the calculated and experimental values of the temperatures of the main HCU elements, refrigerant and oil differ on average by 2…3 °C, maximum – 6…7 °C (deviation does not exceed 5…6 %).

Comparison of the results of calculation and experiment allows us to conclude that the developed mathematical model is adequate to the processes under study and can be recommended for implementation in the design practice of high-speed HCUs.

Using the developed method, the following studies were carried out: the rational thickness of the insulation of the HCU discharge path was determined, various options for the cooling system were considered, the effect of the color of the outer surface of the HCU casing on the temperature level of the machine was evaluated.

Isolation of the discharge path reduces the HCU temperature level by reducing heat fluxes from the discharge cavity, tube and receiver. However, as a result of experimental studies, the rational insulation thickness has not been determined. The availability of a method for calculating the temperature level made it possible to solve this problem.

In fig. 4 shows the calculated dependences of the temperatures of the windings of the electric motor t_{em}, oil t_0 and discharge t_{dis} HCU FGV-2,2 from insulation

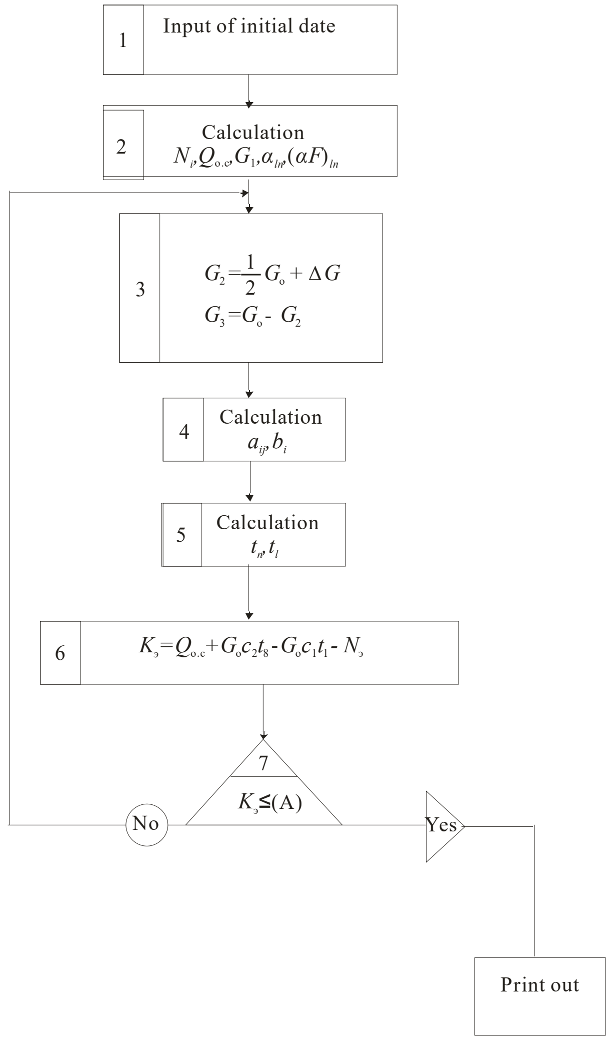


Fig. 2. Enlarged flowchart for calculating the temperature level of a sealed compressor unit

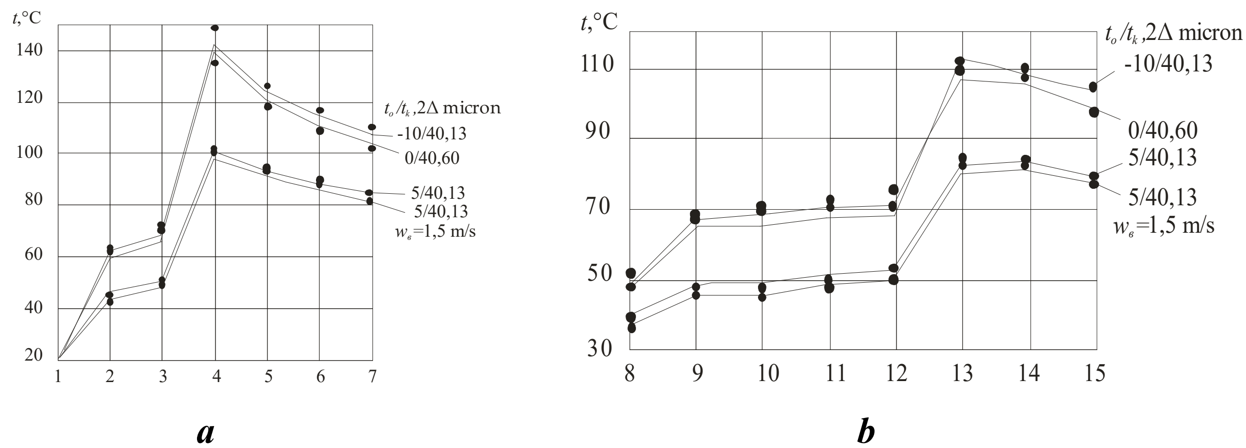


Fig. 3. Temperature change:
a) refrigerant in HCU: *1 – in front of the suction port; 2 – in the casing; 3 – after the electric motor; 4 – at the exit from the cylinder; 5 – in the discharger cavity; 6 – in the receiver; 7 – after the discharge pipe;*
b) SCU elements and oil: *8 – casing; 9 – block crankcase; 10 – oil; 11 – electric motor; 12 – cylinder; 13 – discharge cavity; 14 – receiver; 15 – discharge tube;*
2Δ – diametric clearance piston-cylinder, micron; $w_в$ – air velocity at the SCU casing, m/s; solid line – calculation; points – experiment

thickness δ_{is} discharge path at boiling temperatures t_0 = –10 °C and condensation t_c = 40 °C. From the figure it follows that with an insulation thickness of more than 2 mm, a decrease in temperature t_{em}, и t_0 practically stops and they reach values, respectively 63 and 57 °C.

Thus, for this type of HCU, the rational thickness of the discharge line insulation is 2.0…2.5 mm. In this case (in comparison with an uninsulated discharge path), the temperature of the motor windings decreases by 5 °C, oil – by 9 °C, and the discharge temperature increases by 10 °C.

Studies of HCU ***heat exchange*** with the ***environment*** have shown that the total coefficient of heat transfer from the casing to stationary air, depending on the operating mode, is 12…15 Wh/(m^2·K) [3]. The share of radiant heat transfer in this case is 30…40%. Therefore, one of the ways to enhance heat exchange with the environment can be to increase the heat radiation as a result of discoloration of the outer surface of the SCU casing. For this purpose, for the HCU FGV-2,2 operating in the t_0 = –10 °C and t_c = 40 °C, were calculated temperatures of parts, refrigerant and oil depending on the color of the casing.

The calculation results are shown in fig. 5.

The picture shows that the HCU casing is painted with black matt lacquer (degree of blackness ε = 0,98) allows you to reduce its temperature level by 6…7 °C compared to current yellow color (ε = 0,80).

In ship air conditioning systems with dynamic maintenance principles use air-cooled condenser or water-cooled condenser and cooling tower air conditioners. In this case, the condensation temperature can reach 55…65 °C, which significantly increases the temperature level of the HCU and, consequently, lowers its energy and operational performance. To improve the operating conditions of the HCU in air conditioners operating in a wide range of loads and ambient temperatures, a special

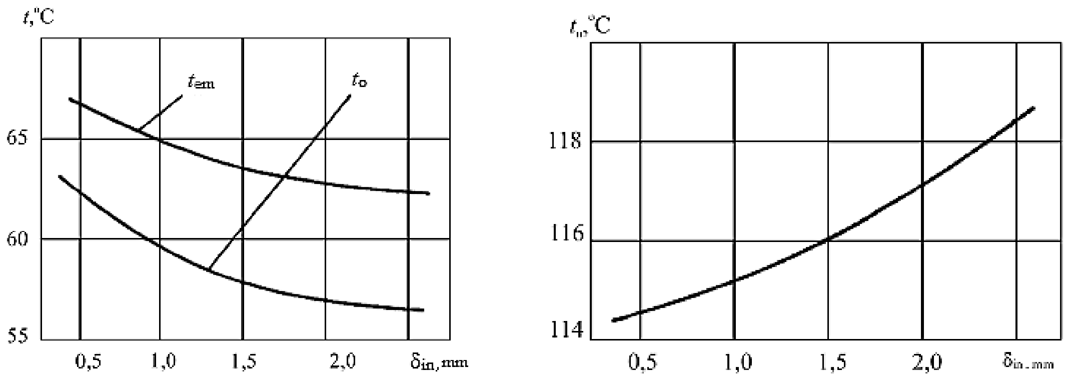


Fig. 4. Dependences of the temperatures of the windings of the electric motor t_{em}, oil t_0 and discharge t_{dis} from insulation thickness δ_{in} discharge path

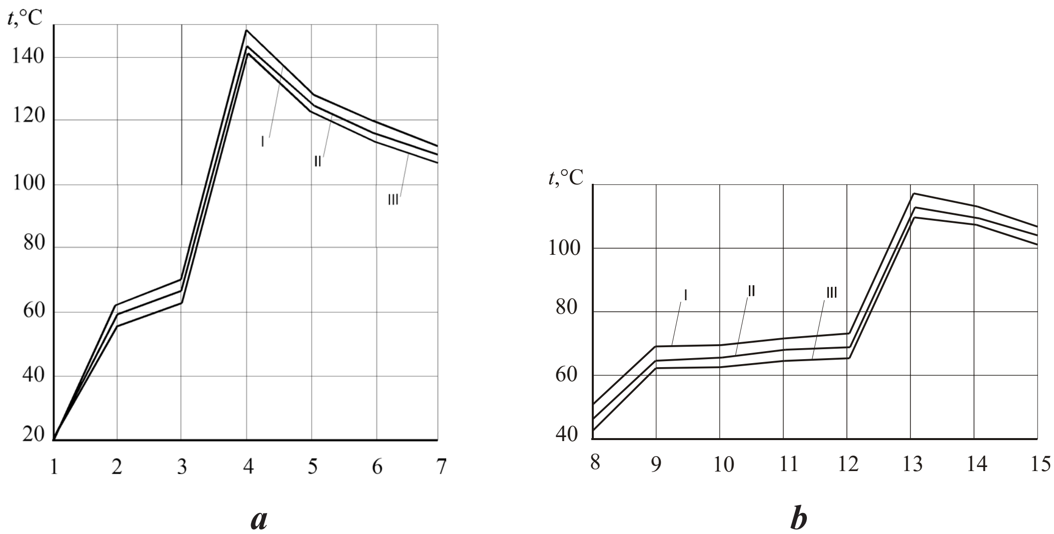


Fig. 5. Temperature change when the outer surface of the HCU is painted yellow (I), grey colour (II), black matte varnish (III):
a) refrigerant in HCU: *1 – in front of the suction port; 2 – in the casing; 3 – after the electric motor; 4 – at the exit from the cylinder; 5 – in the discharge cavity; 6 – in the receiver; 7 – after the discharge pipe;*
b) HCU elements and oil: *8 – casing; 9 – block crankcase; 10 – oil; 11 – electric motor; 12 – cylinder; 13 – discharge cavity; 14 – receiver; 15 – discharge tube*

cooling system is used – cooling of the casing of the sealed compressor unit water or condensate supplied from the evaporator (air cooler) to a special sump that covers the entire HCU or only its lower part [5].

The following cooling system options were investigated:

1) water at a temperature of 15 °C was fed into a sump covering the lower part of the HCU casing;

2) water with a temperature of 15 °C washed the entire HCU casing;

3) water at 35 °C (summer seawater temperature) was fed into a sump covering the lower part of the HCU casing.

The results of calculating the temperatures of the elements, refrigerant and oil of the compressor unit FGV-2.2 are shown in fig. 6. The figure shows that the use of a special cooling system allows you to reduce the temperature level of the HCU by:

In the option *1* – 15…18 °C;

In the option *2* – 20…23 °C;

In the option *3* – 6…8 °C.

According to the author, ***option 1*** is preferable as it is simpler and more technologically advanced in application. The temperature level in this case decreases by about the same amount (10…25 °C), as when the HCU casing is blown with air at a speed 8 m/s [3].

During the operation of the compressor unit KHGV-14 in mode t_0 = 0 °C and t_c = 65 °C with cooling of the lower part of the casing by condensate from the evaporator temperature of the electric motor windings t_{em} and lubricating oil t_o were respectively 58 and 51 °C. In the case of application for cooling the jacket of water with a temperature 35 °C HCU temperature level increased by 10…12 °C, but did not exceed the permissible values.

CONCLUSIONS

A mathematical model and a method for calculating the temperature level of a ship high-speed HCU of the HGV type have been developed, which make it possible to assess their thermal state, determine the temperatures of the compressor elements, refrigerant and lubricating oil, depending on the operating conditions and design parameters of the machine.

It has been found that cooling the HCU casing with water from the air conditioner evaporator, which is fed into a sump covering only the lower part of the casing, reduces the temperature level by 10…25 °C.

It has been shown that painting the HCU casing with black matte varnish reduces its temperature level by 6…7 °C (compared to current yellow color), and the rational thickness of the insulation of the discharge path is 2…2,5 mm.

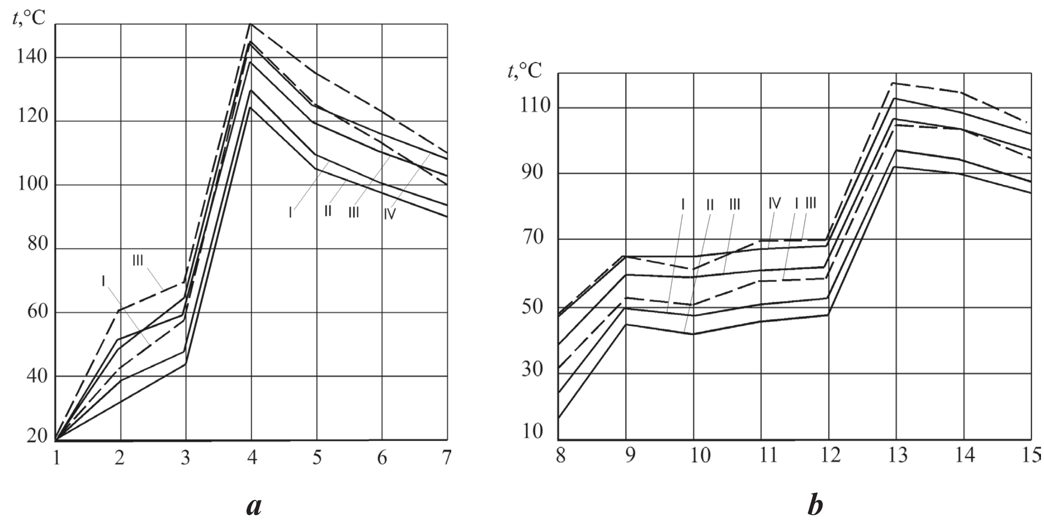


Fig. 6. Temperature change with different HCU cooling options:
I – option 1; II – option 2; III – option 3; IV – standard execution;
solid line – t_0 = –10 ºC; t_c = 40 ºC; dotted – t_0 = 0 ºC; t_c = 65 ºC; (for other designations see fig. 5)

REFERENCES

[1] Vasil'ev, V.D., Solozhentsev, E.D. (1978). *Kiberneticheskie metody pri sozdanii porshnevykh mashin [Cybernetic methods in the creation of piston machines]*. Moscow: Mashinostroenie [in Russian].

[2] Radchenko, N.I., Dorosh, V.S., Radchenko, A.N., Radchenko, R.N., Lytosh, E.V. (2011). *Germetichnye parokompressornye kholodil'nye mashiny sudovykh sistem konditsionirovaniya i refrizheratsii: monografiya [Hermetic vapor-compressor refrigerating machines of ship air-conditioning and refrigeration systems: monograph]*. Mykolaiv: NUK [in Russian].

[3] Dorosh, V.S., Shchesyuk, O.V. (1983). Issledovanie teploobmena vysokooborotnykh germetichnykh kompressorov s okruzhayushchey sredoy [Study of heat exchange of high-speed hermetic compressors with the environment]. *Kholodil'naya tekhnika – Refrigeration technology*, no. 2, pp. 43–46 [in Russian].

[4] Zahoruiko, V.O., Holykov, O.A. (2002). *Sudnova kholodylna tekhnika [Ship refrigeration technology]*. Kyiv: Naukova dumka [in Ukrainian].

[5] Zakharov, Yu.V. (1994). *Sudovye ustanovki konditsionirovaniya vozdukha i kholodil'nye mashiny [Ship air conditioning and refrigeration units]*. Saint Petersburg: Sudostroenie [in Russian].

[6] Plastinin, P.I. (1987). *Teoriya i raschet porshnevykh kompressorov [Theory and calculation of piston compressors]*. Moscow: Agropromizdat [in Russian].

[7] Talyzin, M.S., Gorokhov, S.Yu. (2012). Porshnevye kompressory Stream: novye standarty v kompressorostroenii [Piston compressors Stream: new standards in compressor engineering]. *Kholodil'naya tekhnika – Refrigeration technology*, no. 1, pp. 86–88 [in Russian].

[8] Yakobson, V.B. (1977). *Malye kholodil'nye mashiny [Small refrigeration machines]*. Moscow: Pishchevaya promyshlennost' [in Russian].

[9] Plastinin, P.I. (2006). *Porshnevye kompressory: uchebnoe posobie dlya vuzov [Piston compressors: textbook for universities]*. 3rd ed. Moscow: Kolos [in Russian].

BIBLIOGRAPHY

[1] Васильев В.Д., Соложенцев Е.Д. Кибернетические методы при создании поршневых машин. Москва : Машиностроение, 1978. 120 с.

[2] Герметичные парокомпрессорные холодильные машины судовых систем кондиционирования и рефрижерации : монография / Н.И. Радченко, В.С. Дорош, А.Н. Радченко, Р.Н. Радченко, Е.В. Лытош. Николаев : НУК, 2011. 400 с.

[3] Дорош В.С., Щесюк О.В. Исследование теплообмена высокооборотных герметичных компрессоров с окружающей средой. *Холодильная техника*. 1983. № 2. С. 43–46.

[4] Загоруйко В.О., Голиков О.А. Суднова холодильна техніка. Київ : Наукова думка, 2002. 576 с.

[5] Захаров Ю.В. Судовые установки кондиционирования воздуха и холодильные машины. Санкт-Петербург : Судостроение, 1994. 504 с.

[6] Пластинин П.И. Теория и расчет поршневых компрессоров. Москва : Агропромиздат, 1987. 271 с.

[7] Талызин М.С., Горохов С.Ю. Поршневые компрессоры Stream: новые стандарты в компрессоростроении. *Холодильная техника*. 2012. № 1. С. 86–88.

[8] Якобсон В.Б. Малые холодильные машины. Москва : Пищевая промышленность, 1977. 368 с.

[9] Пластинин П.И. Поршневые компрессоры : учебное пособие для вузов. 3-е изд. Москва : Колос, 2006. 465 с.

Дата надходження статті до редакції: 07.04.2022
Дата затвердження статті до друку: 19.04.2022