ТЕРМОДИНАМІЧНИЙ АНАЛІЗ ТА МОДЕЛЮВАННЯ

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Thermodynamic analysis of a real marine refrigeration system for cruise ship "Scarlet Lady"

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This paper presents an energy analysis of the marine refrigeration system as a component of the machinery system of the cruise liner "Scalet Lady". This type of ship is one of the most complex types from an energy point of view, which is associated with a wide variety of energy consumers. The study uses two methods of thermodynamic analysis of real cycles: the method of cycles and entropy-statistical method. For the analysis, the experimental data on the operating modes of the marine refrigeration system has been used. According to the results of processing the experimental data, a thermodynamic cycle has been formed and an energy calculation of the cycle has been performed using classical methods for two-stage refrigeration machines. With the help of the cycle model, the external irreversibility is determined by successively reducing the energy efficiency of the ideal cycle. The Lorentz cycle has been chosen as an ideal model, taking into account the variable temperatures of heat supply and removal during heat exchange "heat source – phase transformations of the refrigerant R407f". Along with the evaluation of efficiency, the distribution of losses by the components of the refrigeration system has been established, energy weaknesses that require improvement has been identified, and the most effective ways to reduce energy costs have been outlined. Analytical conclusions have a graphical interpretation of the thermodynamic efficiency assessment for the real refrigeration cycle. It has been established that in the considered real cycle of a marine refrigeration system, the greatest influence on energy efficiency is exerted by external irreversibility in the condenser, in particular, in the overheating zone (25.64% of the total energy consumption of the plant). The effect of irreversibility in the compressor is not fixed.

Keywords: *Real ship refrigeration system; Cycle method; Entropy-statistical method; Energy efficiency*

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Nomenclature:

h – specific enthalpy, kJ·kg^{-1};

$\dot{m}$ – mass flow rate, kJ·kg^{-1};

$\dot{Q}$ – heat rate, kW;

$\dot{Q}_e$ – refrigeration capacity, kW;

p – pressure, bar;

s – specific entropy, kJ·kg^{-1}·°C^{-1};

T – temperature, °C;

υ – specific volume, m^3·kg^{-1};

V_h – theoretical volumetric capacities of the compressor, m^3·kg^{-1};

w – specific work, kJ·kg^{-1};

$\dot{W}$ – power, kW;

y – refrigerant mass flow rate in the injection line, kg·s^{-1};

COP – coefficient of performance;

Δ – difference;

η_a – adiabatic efficiency of the screw compressor

Subscripts:

ac – actual;

aver – average;

aver.plan(c) – average planimetric (condenser);

aver.plan(e) – average planimetric (evaporator);

br – coolant (brine);

com – compressor;

cond – condenser;

ec – economizer;

ev, evap – evaporator;

exp – expansion;

in – inlet;

int – intermediate;

min – minimum;

out – outlet;

w – water;

1-6 – state of cycle

Superscripts:

a – adiabatic;

min – minimum;

real – real parameter or value

1. Introduction

Shipping is the lifeblood of the global economy. Ships are technically complex systems with a large amount of energy consumed. Various indicators based on different standards or principles are used to assess the energy efficiency of technical systems. Most of these indicators are used for equipment certification. At the global level, the International Maritime Organization (IMO) evaluates the energy efficiency of maritime transport using tools such as EEDI (Energy Efficiency Design Index), EEOI (Energy Efficiency Operational Index), and SEEMP (Shipping Energy Efficiency Management Plan) which are regulated by IMO circulars MEPC.1/Circ.682, MEPC.1/Circ.684, and MEPC.1/Circ.683, respectively [1]. A full performance evaluation takes into account rising bunker fuel prices. Thus, the maritime industry faces two problems: environmental and economic. The problem of increasing energy efficiency is solved by reducing fuel consumption on board ships.

One of the most complex types of ships from an energy point of view are cruise ships, which is associated with a wide variety of energy consumers. The efficiency of energy saving on ships is largely determined by the operating modes of machinery systems, which also include refrigeration systems. On the scale of such a large installation as a ship power complex, even a small percentage reduction in energy costs becomes a significant savings in absolute terms. Cruise ships use large refrigeration systems with multiple temperature levels of refrigeration production and varying operating conditions for the condensing heat rejection system. They ensure the functioning of refrigeration and freezing equipment, technological and comfortable air conditioning systems on the ship. Moreover, such systems have continuity of cooling processes in time. The energy consumption of ship refrigeration systems on passenger liners is 4% of the total ship energy consumption.

Increasing the energy efficiency of refrigeration systems is achieved by various methods: structural improvement of ccomponents, the use of advanced automatic control systems, the heat transfer intensification, and an increase in the thermodynamic efficiency of cycles. The latter method largely depends on the type of refrigerant used. Environmental requirements for maritime transport are constantly increasing in accordance with the International Convention for the Prevention of Pollution from Ships (MARPOL 73/78) [2]. At present, the refrigerant R407f is used as a medium-term alternative to HCF-type refrigerants used on ships classed by Lloyd’s Register of Shipping and GWP below 2500 [3]. This refrigerant is a non-azeotropic mixture in which phase transition processes occur at variable temperatures. Such features should be taken into account when determining the parameters of the system operation and operation of heat exchange equipment. In view of this, it is relevant to conduct a thermodynamic analysis using the experimental data of a real ship refrigeration system operating on a non-azeotropic mixture. Such an analysis will allow assessing the energy efficiency of operating modes and identifying ways to reduce the cost of maintenance and repair.

2. Literature review and problem statement

Recent publications on the subject matter contain the results of studies of the energy and economic efficiency of refrigeration systems on board ships. The authors propose various methods for improving the corresponding efficiency, which, in their opinion, may have prospects for their use in marine conditions.

Yan et al. carried out an analysis of ship air conditioning systems using modeling and simulation methods [4]. The technical requirements and the operation principle of marine air conditioning, as well as a mathematical model of cabin cooling in summer are described in detail. This model was tested on a particular example of the cabin of the senior officer of the ship "Yu Kun". It is proved that the presented mathematical model is reliable. Başhan and Parlak performed an exergy and economic analysis of the ship's refrigeration system paying attention to the problem associated with the variable sea water temperature [5, 6]. In the first study, it is proposed to use compressors with a variable shaft speed to improve energy efficiency [5]. The temperature of the refrigerant at the outlet of the condenser is chosen as the main parameter for frequency control. In the second study, an exergy analysis of a mechanical compression refrigeration system was carried out [6]. The characteristics of the system operation depending on the variable temperature of sea water are studied. The exergy losses in the refrigeration system are determined. It has been established that the greatest losses are observed in the compressor and expansion valve. Ovcharenko et al. provided a review of the methods used to improve the efficiency of the ship refrigeration plants [7]. Two methods are considered: the subcooling of the refrigerant after the condenser and the automatic control of compressors. The expediency of using these methods in the operation of ship refrigeration plants using refrigerants R22 and R717 is discussed.

A performance assessment of a cruise ship refrigeration plant using modern alternative refrigerants with low GWP was made by Pigani et al. [8]. Various cycles with the use of such refrigerants have been studied. The results of comparison with the parameters of existing systems based on R407f are given. It is concluded that the transition from modern technologies to systems using low GWP refrigerants entails a deterioration in energy performance and is not an effective strategy to reduce the overall environmental impact. Başhan and Kökkülünk studied the existing mechanical compression refrigeration systems on the example of the ship "Ince Ilgaz" in the case of variable sea water temperatures [9]. In this study were used he exergoeconomic and environmental analysis. On the base of the obtained results were proposed some new refrigeration systems with condensing heat recovery. Calculations were made for 15 different refrigerants. The author concluded that the use of condensation heat recovery in the refrigeration system can directly reduce fuel consumption and emissions to the atmosphere. A comparative analysis of the ship mechanical compression refrigeration system using refrigerant R435a as an alternative to refrigerant R134a has been carried out by Memet [10]. The analysis showed that the exergy efficiency of the system when operating with refrigerant R435a is 35% higher than this value for the system operating with refrigerant R134a.

When analyzing the literature review, it can be stated that there are no studies based on real data on the operation of existing refrigeration systems. The main parameters of the refrigeration system change over time and depend on many factors. Therefore, it is very important to conduct an experimental verification of the operation of the such system to establish the reliability of theoretical studies. Thus, it becomes relevant to continue research on ship refrigeration systems based on the available real experimental data using scientific tools to improve energy efficiency. The authors of studies [4-10] used exergy, thermoeconomic, and environmental analyzes to assess the energy efficiency of ship refrigeration systems. Such methods include many real cost and environmental parameters, which are very difficult to unify. This makes the thermodynamic calculations very time consuming. The use of such multi-criteria analyzes for local ship refrigeration systems is not advisable.

To estimate irreversibilities in the ship refrigeration systems in isolation from the entire power system of the ship, the authors of this paper propose to use entropy methods of thermodynamic analysis. Such methods allow, along with efficiency assessment, to obtain the distribution of losses over the components of the refrigeration cycle. This, in turn, allows to determine the weak point of the refrigeration system that requires perfection, and finding the most effective ways to reduce energy costs [11]. Literature review shows that the topic of this research is insufficiently studied and needs to be studied in detail to solve energy saving problems for real ship refrigeration systems.

3. Determination of characteristics of a of ship refrigeration cycle

The presented study has been carried out using data from a real refrigeration system (chiller) of the cruise ship "Scalet Lady" [12]. A schematic diagram of this refrigeration system is shown in Fig. 1. The presented refrigeration system provides storage of frozen goods in 7 provisional chambers. The intermediate coolant (brine) is ethylene glycol. The chiller consists of an oil-flooded screw compressor with an oil cooling system, a horizontal shell-and-tube condenser, a shell-and-tube economizer, a shell-and-tube evaporator with refrigerant boiling inside the pipes, a system of expansion valves and fittings. The refrigerant in the refrigeration system is R407f. The refrigeration unit operates on a two-stage compression cycle. Two-stage compression is carried out in one cavity of the compressor with injection of the saturated vapor at an intermediate pressure.

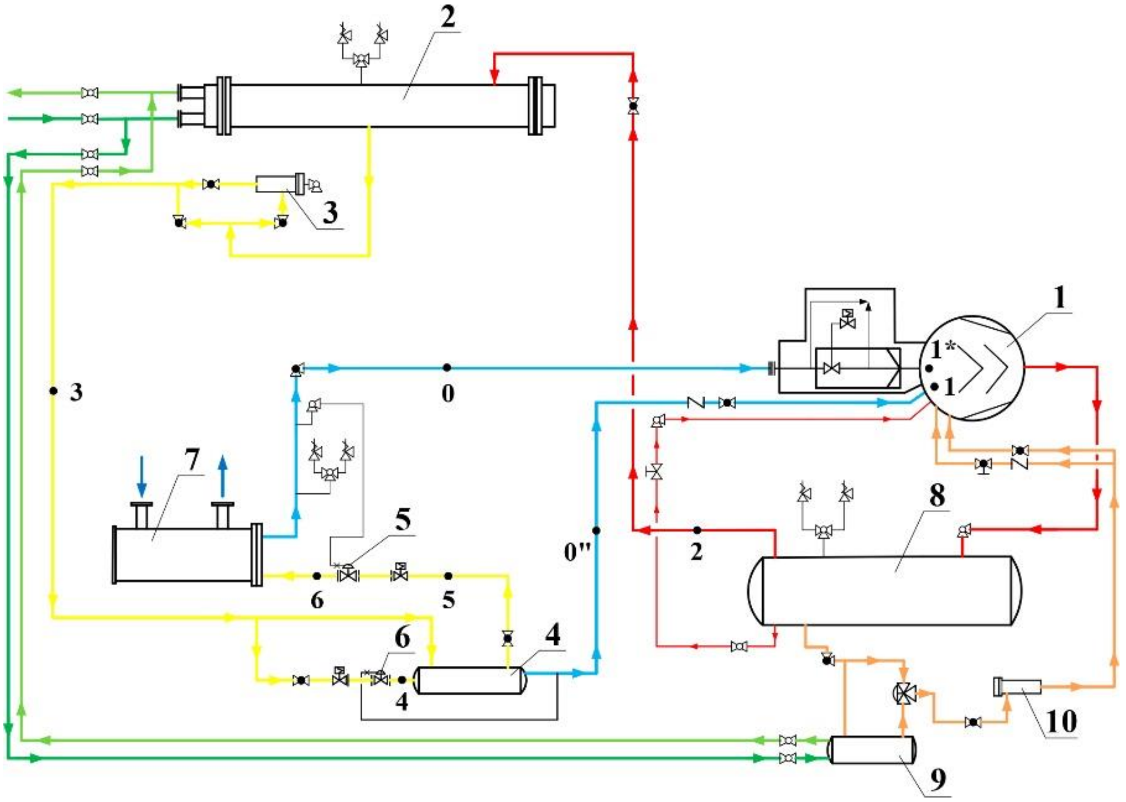


Figure 1 *– A schematic diagram of the ship refrigeration system: 1 – screw compressor; 2 – shell-and-tube condenser, 3 – filter drier, 4 – shell-and-tube economizer; 5 – expansion valve 1; 6 – expansion valve 2; 7 – shell-and-tube evaporator; 8 – oil separator, 9 – oil cooler 10 – mechanical filter*

For the thermodynamic analysis of the ship refrigeration system, the parameters and characteristics of the cycle obtained in real operating conditions during the ship movement were used (Fig. 2 and Table 1).

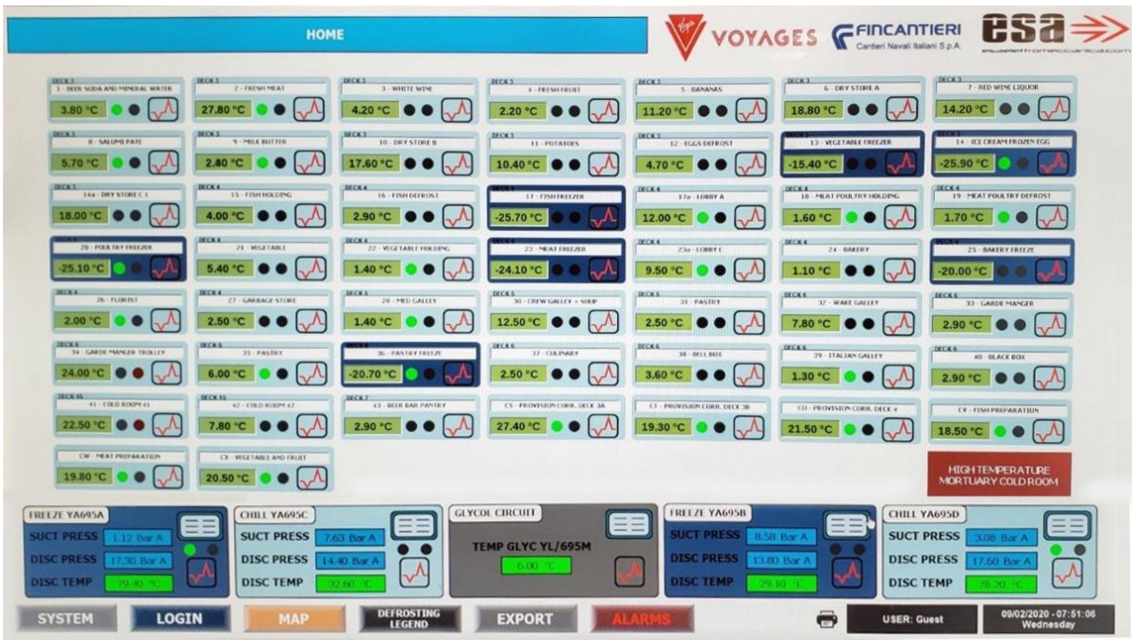


Figure 2 *– Parameters of the ship refrigeration system operation conditions in real time during the ship movement*

***Table 1** – Measured parameters of the refrigeration system for the storage of frozen goods*

Parameter	Value
Temperature at the compressor inlet T_1, °C	–31.9
Discharge temperature T_2, °C	79
Temperature of the coolant (brine) at the evaporator inlet $T_{br,in}$, °C	–33.4
Temperature of the coolant (brine) at the evaporator outlet $T_{br,out}$, °C	–38
Temperature of the water at the condenser inlet $T_{w,in}$, °C	34
Temperature of the water at the condenser outlet $T_{w,out}$, °C	36
Pressure at the compressor inlet p_1, bar	1.12
Discharge pressure p_2, bar	18.2
Intermediate pressure p_{int}, bar	1.33
Temperature at the evaporator inlet $T_{ev,in}$, °C	–42
Temperature at the economizer inlet $T_{ec,in}$, °C	38
Temperature at the economizer outlet $T_{ec,out}$, °C	6.5
Theoretical volumetric capacities of the compressor V_h, $m^3 \cdot h^{-1}$	705.60*
Actual volumetric capacities of the compressor $V_{h,ac}$, $m^3 \cdot h^{-1}$	507.96*

* In accordance with the ship's instructions

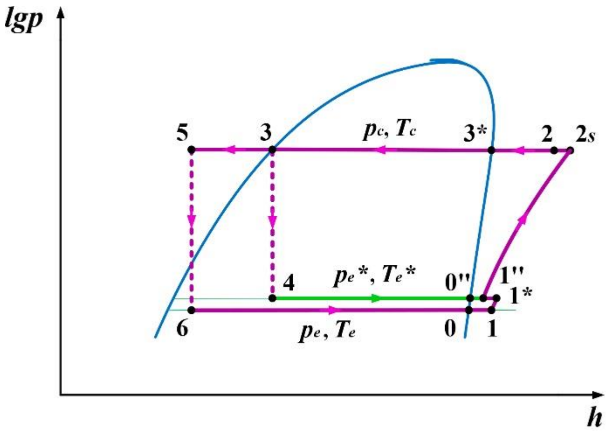
Based on the results of experimental data processing, a thermodynamic cycle has been formed and presented in the *lgp-h* and *T-s* diagrams (Fig. 3), the state parameters have been determined (Table 2), and the energy calculation of the cycle has been performed using classical methods for two-stage refrigeration machines presented in [14] (Table 3).

To carry out the energy calculation, it is conditionally assumed that 1 $kg \cdot s^{-1}$ of refrigerant circulates through the evaporator. Then the refrigerant mass flow rate in the compressor is (1 + *y*) kg s^{-1}, where *y* kg s^{-1} is the refrigerant mass flow rate in the injection line. The heat balance of the economizer can be written as:

$$1 \cdot h_3 + y \cdot h_4 = 1 \cdot h_5 + y \cdot h_{0''} \qquad (1)$$

From Eq. (1) the value of the refrigerant mass flow rate in the injection line *y* is define as follows:

$$y = \frac{h_5 - h_3}{h_4 - h_{0''}}. \qquad (2)$$



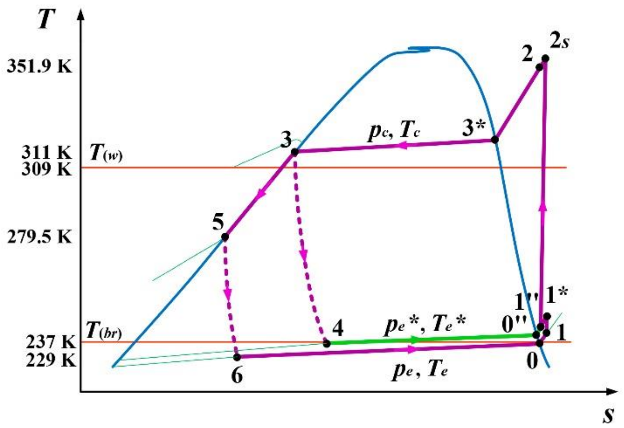


***Figure 3** – A real refrigeration cycle in lgp–h and T-s diagrams*

***Table 2** – Thermodynamic parameters for the states of the marine refrigeration system*

State	0	1	1*	1''	0''	2s	2	3	3*	4	5	6
p, bar	1.12	1.12	1.33	1.33	1.33	18.20	18.20	18.20	18.20	1.33	18.20	1.12
T, °C	–37.5	–31.9	–25.0	–30.0	–34.0	80.0	79.0	38.0	42.5	–40.0	6.5	–42.0
h, $kJ \cdot kg^{-1}$	390.0	395.7	400.0	397.9	392.8	475.7	469.0	260.0	418.5	260.0	208.0	208.0
υ, $m^3 \cdot kg^{-1}$		0.21		0.18		0.0161	0.0159					
s, $kJ \cdot kg^{-1} \cdot °C^{-1}$	1.837	1.851	1.851	1.836	1.826	1.836	1.834	1.214	1.721	1.294	1.036	1.067

Table 3 – Design characteristics of the real refrigeration cycle

Parameter	Value
Refrigeration capacity $\dot{Q}_e$, kW	122.29
Refrigerant mass flow rate in the evaporator $\dot{m}_{ev}$, kg·s^{-1}	0.671
Refrigerant mass flow rate in the condenser $\dot{m}_c$, kg·s^{-1}	0.934
Refrigerant mass flow rate in the economizer $\dot{m}_{ec}$, kg·s^{-1}	0.263
Oil mass flow rate in the economizer $\dot{m}_{oil}$, kg·s^{-1}	5.27
Condensing heat load $\dot{Q}_c$, kW	191.2
Economizing heat load $\dot{Q}_{ec}$, kW	34.8
Compressor power consumption $\dot{W}_{com}$, kW	83.84
Coefficient of performance *COP*	1.45

4. Analysis of the real marine refrigeration system operation by the entropy-statistical method

The actual processes occurring in the marine refrigeration system are non-equilibrium and irreversible, and are accompanied by an increase in entropy in all components of the system. The degree of irreversibility of the processes and the amount of work required to compensate for irreversibility is estimated using the Gouy-Stodola theorem (entropy generation in each component of the system). This principle is the basis of entropy-statistical analysis. This method makes it possible to determine the necessary energy costs to compensate for the entropy generation as a result of the irreversibility of work processes in various components of refrigeration systems and indicates ways to improve them [13].

The study of a real marine refrigeration system by the entropy-statistical method has been carried out using experimental data and calculations (Tables 1-3). Evaluation of the efficiency of the cycle is carried out according to the specific costs of electricity and the value of the of thermodynamic perfection parameter. Specific costs are related to 1 kg of refrigerant circulating through the evaporator.

The use of a non-azeotropic mixture in a refrigeration system leads to the fact that the isobaric processes of supply (boiling) and rejection (condensing) of heat in the heat exchangers are non-isothermic. In the presented system, the processes of heat supply in the evaporator (cooling of the coolant) and heat rejection in the condenser (heating of the water) also occur at variable temperatures. To fulfill the conditions of reversibility, the temperature of the working fluid must change in the same way as the temperatures of the coolant and water. The entropy-statistical analysis is based on the assumptions of the constancy of the temperatures of the coolant and water and in the processes of phase transitions of the refrigerant. Given this condition, to continue the study it is necessary to carry out the transition from variable to constant temperatures during heat transfer. To perform this transition, it is necessary to use the method of cycles [16]. As a reversible sample cycle for cycles with variable temperatures of heat supply and heat rejection, the Lorentz cycle is used. This cycle consists of two adiabatic processes and two processes with variable temperatures, the nature of which exactly follows the changes in source temperatures (Fig. 4, Cycle 1, Cycle 3). The cycles are built at the corresponding temperature levels: for sources of heat supply (brine) and heat rejection (water) – Cycle 1, and for the working fluid (refrigerant) of the refrigeration system – Cycle 3.

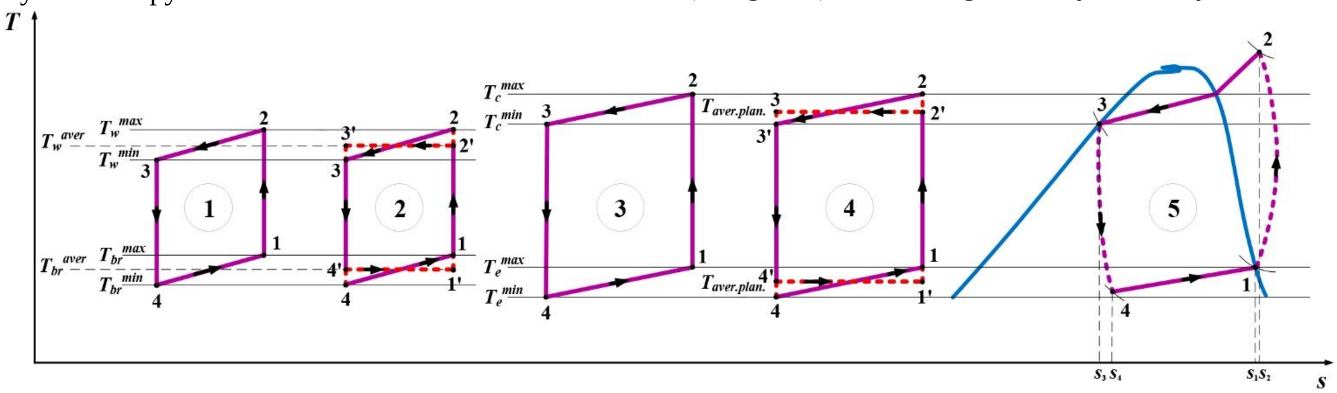


***Figure 4** – The method of cycles in determining the temperatures of heat supply and heat rejection of the real refrigeration cycle*

For any reversible Lorentz cycles, the only limitation is the need for constancy and equality of heat capacities in the processes of heat supply and heat rejection (lines 4-1 and 3-2 are equidistant). Skipping intermediate constructions, the real cycle with a real working fluid can be represented by Cycle 5 (Fig. 4). In the real cycle, the heat capacities in the processes of heat supply and heat rejection are not equal and not constant. For further analysis, it is necessary to replace the Lorentz cycles with equivalent Carnot cycles (Fig. 4, Cycle 2, Cycle 4), which have the same thermodynamic characteristic $COP_{LOR} = COP_{CAR}$. The temperature boundaries in the equivalent Carnot cycle are taken equal to the average planimetric temperatures in the processes of heat supply and heat rejection in the Lorentz cycle (Fig. 4, Cycle 2, Cycle 4).

The processes of heat supply and heat rejection in the presented refrigeration system are isobaric. In this case, it is not required to carry out planimetry, and the numerical value of temperatures in the equivalent Carnot cycle is determined by the values of enthalpies and entropies at the beginning and end of the process (Figure 4, Cycle 5).

To determine the design temperatures of the working fluids, it is necessary to use the following equations:

$$T_{aver.plan(c)} = \frac{h_{3^*} - h_3}{s_{3*} - s_3} \qquad (3)$$

$$T_{aver.plan(e)} = \frac{h_0 - h_6}{s_0 - s_6} \qquad (4)$$

The design temperatures of the brine and water are determined as the arithmetic mean values between the extreme temperatures, in view of the small heating and cooling values of the corresponding flows (Table 1):

$$T_{w,aver} = \frac{T_{w,in} + T_{w,out}}{2} \qquad (5)$$

$$T_{br,aver} = \frac{T_{br,in} + T_{br,out}}{2} \qquad (6)$$

Any pressure losses haven't been taken into account in heat exchangers and pipelines when carrying out the analysis by the entropy-statistical method. Initial data for analysis with considering Eqs. (3)-(6) are written as follows.

The average temperature of the fresh water in the condenser of the refrigeration system – $T_{w,aver}$ = 309 K.

The average temperature of the refrigerant (ethylene glycol) in the evaporator of the refrigeration system – $T_{br,aver}$ = 237 K.

Design evaporating temperature of the working fluid – $T_{aver.plan(e)}$ = 236 K.

Design condensing temperature of the working fluid – $T_{aver.plan(c)}$ = 312.62 K.

The sequence of analysis of the main components are written as follows.

The real specific cooling capacity of the refrigeration system is:

$$q_e^{real} = h_0 - h_6. \qquad (7)$$

The minimum specific work for cold production is:

$$w_{min} = q_e^{real} \cdot \frac{T_{w,aver} - T_{br,aver}}{T_{br,aver}}. \qquad (8)$$

The adiabatic specific work of compression is separated into two parts:

– for compression in the compressor from pressure p_e to p_e^*

$$w_{com1}^a = h_1 - h_{1*}. \qquad (9)$$

– for compression in the compressor from pressure p_e^* to p_c

$$w_{com2}^a = \left(h_{2s} - h_{1''}\right) \cdot \left(1 + y\right). \qquad (10)$$

The total adiabatic specific work of compression in the compressor is:

$$\sum w^a = w_{com1}^a + w_{com2}^a. \qquad (11)$$

The minimum required specific work of compression to compensate for the entropy generation in the condenser when the refrigerant vapor is cooled from temperature T_{2s} = 80ºC to saturation temperature T_c^{max} = 42.2ºC is defined as follows:

$$\Delta w_{cool.cond}^{min} = \left(h_{2s} - h_{3*}\right) \cdot \left(1 + y\right) - T_{w,aver} \cdot \left(s_{2s} - s_{3*}\right). \qquad (12)$$

The minimum required specific work of compression to compensate for the entropy generation in the condenser during the condensation of the refrigerant vapor is:

$$\Delta w_{cond}^{min} = T_{w,aver} \cdot \left(h_{3*} - h_3\right) \cdot \left(1 + y\right) \cdot \left(\frac{1}{T_{w,aver}} - \frac{1}{T_{aver.plan(c)}}\right). \qquad (13)$$

The minimum required specific work of compression to compensate for the entropy generation when throttling in expansion valve 1 is:

$$\Delta w_{exp1}^{min} = T_{w,aver} \cdot \left(s_6 - s_5\right). \quad (14)$$

The minimum required specific work of compression to compensate for the entropy generation when throttling in expansion valve 2 is:

$$\Delta w_{exp2}^{min} = T_{w,aver} \cdot \left(s_4 - s_3\right) \cdot y. \quad (15)$$

The minimum specific work of compression to compensate for the entropy generation in the evaporator is:

$$\Delta w_{evap}^{min} = T_{w,aver} \cdot q_e^{real} \cdot \left(\frac{1}{T_{aver.plan(e)}} - \frac{1}{T_{br,aver}} \right). \quad (16)$$

The minimum specific work of compression to compensate for entropy generation in the economizer is:

$$\Delta w_{econ}^{min} = T_{w,aver} \cdot \left[y \cdot \left(s_{0''} - s_4\right) - \left(s_3 - s_5\right) \right]. \quad (17)$$

The total value of the minimum specific work to compensate for the entropy generation in all components of the refrigeration system describes the design adiabatic specific work of compression:

$$w_*^a = w_{min} + \Delta w_{cool.cond}^{min} + \Delta w_{cond}^{min} + \\ + \Delta w_{exp1}^{min} + \Delta w_{exp2}^{min} + \Delta w_{evap}^{min} + \Delta w_{econ}^{min}. \quad (18)$$

The real specific work of compression can be found from Eqs. (19) and (20) taking into account the adiabatic efficiency of the screw compressor $\eta_a = 0.83$:

– for compression in the compressor from pressure p_e to p_e^*

$$w_{com1}^{real} = \frac{h_1 - h_{1*}}{\eta_a}. \quad (19)$$

– for compression in the compressor from pressure p_e^* to p_c

$$w_{com2}^{real} = \left(h_2 - h_{1''}\right) \cdot \left(1 + y\right). \quad (20)$$

The total real specific work of compression is:

$$\sum w^{real} = w_{com1}^{real} + w_{com2}^{real}. \quad (21)$$

Coefficient of performance of the real refrigeration cycle is:

$$COP^{real} = \frac{q_0^{real}}{\sum w^{real}}. \quad (22)$$

The calculation results are presented in Table 4.

***Table 4** – System calculation results*

Parameter	Value
Real specific cooling capacity of the refrigeration system q_e^{real}, kJ·kg^{-1}	182
Minimum specific work for cold production w_{min}, kJ·kg^{-1}	55.29
Adiabatic specific work of compression from p_e to p_e^* w^a_{com1}, kJ·kg^{-1}	4.3
Adiabatic specific work of compression from p_e^* to p_c w^a_{com2}, kJ·kg^{-1}	109.88
Total adiabatic specific work of compression Σw^a, kJ·kg^{-1}	114.2
Minimum required specific work of compression to compensate for the entropy generation in the condenser when the refrigerant vapor is cooled $\Delta w^{min}_{cool.cond}$, kJ·kg^{-1}	30
Minimum required specific work of compression to compensate for the entropy generation in the condenser during the condensation of the refrigerant vapor Δw^{min}_{cond}, kJ·kg^{-1}	2.55
Minimum required specific work of compression to compensate for the entropy generation when throttling in expansion valve 1 Δw^{min}_{exp1}, kJ·kg^{-1}	9.57
Minimum required specific work of compression to compensate for the entropy generation when throttling in expansion valve 2 Δw^{min}_{exp2}, kJ·kg^{-1}	9.66
Minimum specific work of compression to compensate for the entropy generation in the evaporator Δw^{min}_{evap}, kJ·kg^{-1}	1.0054
Minimum specific work of compression to compensate for entropy generation in the economizer Δw^{min}_{econ}, kJ·kg^{-1}	9.1

Table 4 continuation

Parameter	Value
Total value of the minimum specific work to compensate for the entropy generation in all components of the refrigeration system $w^a{}_*$, kJ·kg^{-1}	117
Real specific work of compression Σw^{real}, kJ·kg^{-1}	98.57
Coefficient of performance of the real refrigeration cycle COP^{real}	1.59

The discrepancy between the calculated values of the adiabatic work of compression obtained by Eq. (11) and Eq. (18) does not exceed 2.5%. This result indicates a real overspend of work for each component of the refrigeration system.

The distribution of the design specific energy consumption by the components of the refrigeration system is shown in Figures 5 and 6. It has been found that the contribution to the total irreversibility is made by the processes in the evaporator (0.85%), in the condenser (27.82%), in the compressor (0%), in the economizer (7.7%), and in the expansion valves (16.4%) when comparing the results of the analysis in the components of the refrigeration system (Fig. 5). Such a distribution allows, under specific circumstances, to focus on the improvement of one or another component of the system.

In the presented real marine refrigeration cycle, the external irreversibility in the condenser has the greatest impact on energy efficiency; more precisely, in the overheat rejection zone. They accounted for 25.64% of the total energy consumption of the refrigeration system. The effect of irreversibility in the compressor is not fixed. This is due to the perfect design of the oil-flooded screw compressor.

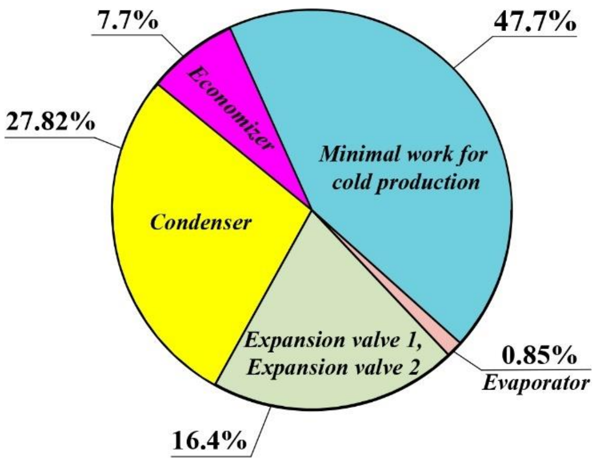


Figure 5 *– Graphical interpretation of the thermodynamic efficiency evaluation of the real marine refrigeration cycle*

In the processes of phase transformations occurring in the heat exchangers (condenser and evaporator), the minimal irreversibility of the total work in the refrigeration cycle has been recorded: 0.85% in the evaporator and 2.9% in the condenser. Such results are determined by the same nature of the change in the temperatures of non-azeotropic mixtures in the processes of phase transformations, coolant and water. In the graphical representation, the lines of the processes are equidistant (Fig. 4). It is known that the energy consumption of a refrigeration system largely depends on the temperature operation mode. The presented system operates in a low-temperature mode, so the minimum work for cold production is 47.7% of the total energy consumption of the refrigeration system.

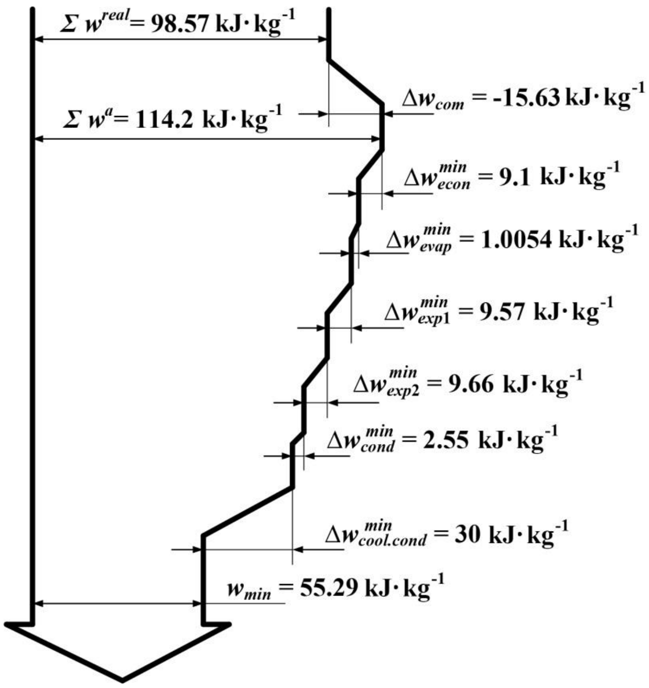


Figure 6 *– Diagram of the specific energy consumption distribution in the main components of the refrigeration system*

5. Discussion of the results and conclusions

The study uses two methods of thermodynamic analysis of real cycles: the method of cycles and entropy-statistical method. The method of cycles has been used as a tool to prepare for the analysis by the entropy-statistical method, which made it possible to determine the design parameters of the refrigeration system. This combination of methods simplified the analysis and ensured the reliability of the results. In the entropy-statistical method, the studied characteristics are entropy and compressor operation, which are associated with real processes in all components of the refrigeration system. These functions determine the timing of maintenance of the refrigeration system. The amount of work overrun evaluates the decrease in the energy efficiency of the cycle and indicates the components of the greatest influence on its reduction. According to the results of the study, the overheat rejection zone in the condenser has the greatest negative impact on energy efficiency. The overrun of work in this component is 27.7% of the total energy consumption in the refrigeration cycle. The reason for this phenomenon is the thermophysical properties of the refrigerant R407f (low heat capacity of superheated vapor).

At the same time, due to the fact that the processes of phase transformations are non-isothermic at changing temperatures of the brine and water, this leads to a decrease in heat transfer losses and brings the processes closer to reversible ones. The analysis of the refrigeration system by the entropy-statistical method makes it possible to extensively evaluate the influence of the thermophysical properties of the working fluids in each component of the system. The analytical solution of a scientific problem is accompanied by a presentation of the results in a graphical form. This contributes to a clear and precise understanding of the thermodynamic patterns of energy conversion in the refrigeration system, shows the progress of the analysis carried out in the form of a geometric image of the distribution of energy costs among the components of the system, and allows outlining ways to improve their performance.

The thermodynamic analysis shows that the presented real marine refrigeration system under real operating conditions has a high energy efficiency for low-temperature machines (COP_{real} = 1.59). Thus, this refrigeration system meets the mandatory IMO standards for the energy efficiency of ships. The improvement of refrigeration systems of this type is within the competence of shipbuilding design organizations and requires the replacement of the design of heat exchange equipment, in particular the condenser.

CRediT author statement

Larisa Morosuk: Conceptualization, Formal analysis, Writing – Original Draft. **Victoriia Sokolovska-Yefymenko**: Methodology, Writing – Review & Editing, Project administration, Supervision. **Volodymyr Ierin**: Investigation, Data Curation, Writing – Review & Editing. **Oleksandr Yefymenko**: Funding acquisition, Resources, Validation. **Andrii Moshkatiuk**: Visualization, Software.

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Термодинамічний аналіз циклу дійсної суднової холодильної установки круїзного лайнера «Scarlet Lady»

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У статті наведено енергетичний аналіз холодильної установки як елемента системи машинного обладнання круїзного лайнера «Scalet Lady», який є одним із найскладніших типів морських суден з енергетичної точки зору, що пов'язане з великою різноманітністю споживачів енергії. У дослідженні використовуються два методи термодинамічного аналізу реальних циклів: метод циклів та ентропійно-статистичний метод. Для аналізу використані експериментальні дані режимів роботи суднової холодильної установки. За результатами обробки експериментальних даних сформовано термодинамічний цикл і, виконано енергетичний розрахунок циклу з використанням класичних методів для двоступеневих холодильних машин. Для оцінки незворотних втрат у бортовій холодильній установці у відриві від усієї енергетичної системи судна використані ентропійні методи термодинамічного аналізу. За допомогою циклової моделі визначено зовнішні незворотності шляхом послідовного зменшення енергетичної ефективності ідеального циклу. Ідеальним зразком обраний цикл Лоренца, що враховує змінні температури підведення та відведення тепла при теплообміні «джерело тепла – фазові перетворення холодоагенту R407f». Поряд з оцінкою ефективності отримано розподіл втрат за елементами холодильної установки, визначено енергетично слабкі місця, що потребують удосконалення, намічені найефективніші шляхи зменшення енерговитрат. Аналітичні висновки мають графічну інтерпретацію оцінки термодинамічної ефективності циклу дійсної холодильної установки. Встановлено, що у розглянутому дійсному циклі суднової холодильної установки найбільший вплив на енергетичну ефективність мають зовнішні незворотності в конденсаторі, зокрема, у зоні зняття перегріву (25,64% від загального енергоспоживання установки). Впливу незворотності на роботу компресора не зафіксовано.
Ключові слова: *Дійсна холодильна система; Метод циклів; Ентропійно-статистичний метод; Енергоефективність*

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