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ANALYSIS OF THE EFFECT OF THERMOHYDRAULIC IRREVERSIBILITY OF PROCESSES IN THE CYCLE OF A REFRIGERATION MACHINE WITH A NON-AZEOTROPIC MIXTURE OF REFRIGERANTS

Об'єктом дослідження є вплив термогідравлічної незворотності енергетичних процесів в циклі холодильної машини, що використовує озонобезпечну неазеотропну суміш як холодоагент, на її енергетичну ефективність. Одним з найбільш проблемних місць в ході розробки і проектування таких холодильних машин є те, що в силу особливостей термодинамічних властивостей, що пов'язані з різним хімічним складом компонентів, неазеотропні суміші характеризуються відмінністю рівноважних концентрацій компонентів в рідкій і паровій фазах. Ця властивість неазеотропних сумішей представляє певну складність для їх ефективного застосування в холодильних машинах і теплових насосах. В ході дослідження використовувалися сучасні методи аналізу і синтезу термодинамічних систем, що засновані на застосуванні теоретичного апарату технічної термодинаміки, термоекономіки, теорії тепло- і масопереносу, а також елементів теорії системотехніки. Було розглянуто питання оцінки енергетичної ефективності циклів холодильних установок, що працюють на неазеотропній суміші холодоагентів, з урахуванням варіативності складу компонентів суміші. Розроблено методику формування складу багатокомпонентної суміші з урахуванням впливу неізобарності процесів в гідравлічному контурі циркуляції холодоагенту на енергетичну ефективність холодильної машини. На основі чисельного експерименту встановлено вплив зміни концентрацій компонентів суміші R32, R125, R134a на неізотермічність фазового переходу в випарнику і конденсаторі, а також на втрати тиску в елементах гідравлічного контуру автономного кондиціонера. Проаналізовано вплив тертя на температурному рівні нижче температури навколишнього середовища на ексергетичну ефективність холодильної машини. Проведено поглиблений ексергетичний аналіз холодильного циклу з неазеотропною сумішшю, в результаті якого визначені неминуча і переборна, а також ендогенна і екзогенна складові деструкції ексергії в елементах. Запропонована методика в силу своєї ілюстративності дозволяє істотно спростити визначення термодинамічних параметрів холодоагенту в вузлових точках циклу холодильної машини при чисельному моделюванні.

Ключові слова: ексергетичний аналіз, неазеотропна суміш, неізотермічність фазового переходу, гідравлічний контур холодоагенту.

1. Introduction

With the adoption of the Montreal and Kyoto protocols, the refrigeration industry entered a new stage of its development, characterized by a complete rejection of the use in refrigeration and heat pump units of refrigerants that deplete the ozone layer. Choosing a new alternative refrigerant is a difficult task, as it is always a compromise between conflicting requirements. Consideration should be given to aspects of global and local, direct and indirect environmental impact of the refrigerant. In addition, the refrigerant must have good thermodynamic and operational characteristics, as well as a low level of toxicity. Recently, environmental safety requirements have prevailed over other refrigerant requirements. Meanwhile, the issue of ensuring high energy efficiency of refrigeration equipment remains one of the main ones, since with a decrease in energy costs for the drive of the refrigeration machine, the indirect influence of the refrigerant on the environment decreases. As the efficiency of the refrigeration machine increases, energy consumption is reduced and thereby emissions of carbon dioxide into the atmosphere during its production in power plants are reduced. Thus, the environmental consequences are more dependent on the thermodynamically sound choice of an energy-efficient refrigerant than directly on the direct hazard indicators of the refrigerant.

Currently, the choice of one-component refrigerants is significantly limited, which in their properties could completely replace environmentally hazardous refrigerants. One of the directions for replacing ozone-hazardous

refrigerants of refrigerators is the use of non-azeotropic mixed refrigerants [1, 2].

Therefore, research aimed at increasing the energy efficiency of the cycles of refrigeration units operating on a non-azeotropic mixture of refrigerants, taking into account the variability of the composition of the components of the mixture, is relevant.

2. The object of research and its technological audit

The object of research is the effect of thermohydraulic irreversibility of energy processes in the cycle of a refrigeration machine using an ozone-safe non-azeotropic mixture as a refrigerant on its energy efficiency.

Non-azeotropic mixtures of refrigerants are characterized by a difference in the equilibrium concentrations of the components in the liquid and vapor phases. This makes it possible to form a refrigerant in such a way that for given temperature conditions [2, 3]:

– by changing the concentrations of the components in the mixture to increase the cooling capacity of the installation;

– without changing the cooling capacity, lower the boiling point, reduce the steam temperature at the discharge, reduce the irreversibility of the heat transfer process, and much more.

Due to the peculiarities of the thermodynamic properties associated with the different chemical composition of the components, non-azeotropic mixtures are of particular difficulty for use in refrigeration machines (RM) and heat pumps (HP). Compared with refrigerants, the properties of which obey the laws for pure substances, mixtures have a number of differences, the main of which is the non-isothermal phase transition during evaporation and condensation. In the literature, this property of the mixture is called temperature glide [3]. The non-isothermality of mixture arises as a result of the inhomogeneity of the dependence of the saturation pressure on temperature, due to differences in the boiling and condensation temperatures of its components.

As is known [2], the non-isothermality of mixture depends on two factors. The first factor is the concentration of the components of the mixture. The second is the level of hydraulic resistances along the mixture flow path. Moreover, the influence of the second factor is especially enhanced in the off-design operation mode of the installation and significantly affects the design dimensions of the evaporator and condenser during design.

Selective loss of one of the components of the mixture leads to a change in pressure in the elements of RM. Therefore, when developing a hydraulic circuit for the refrigerant circuit, it is important to be able to predict the level of pressure loss in it depending on changes in the concentrations of the components of the mixture. In this case, the search for the effective composition of the mixture must be carried out in conjunction with the development of structural elements of the piping of RM. This design approach will minimize the effect of thermo-hydraulic irreversibility in the cycle.

3. The aim and objectives of research

The aim of research is development of a method for the formation of the composition of a multicomponent mixture, taking into account the influence of the non-isobarity of processes in the hydraulic circuit of the refrigerant circulation on the energy efficiency of the refrigeration machine.

To achieve this aim, it is necessary to complete the following objectives:

1. Based on a numerical experiment, determine the effect of changes in the composition of a non-azeotropic mixture on pressure losses in the refrigerant circuit.

2. Evaluate the effect of friction at a temperature level below ambient temperature on the exergy efficiency of the refrigeration machine.

3. Conduct an advanced exergy analysis of the refrigeration cycle with a non-azeotropic mixture in order to determine the avoidable and unavoidable, as well as the endogenous and exogenous components of the destruction of exergy in the elements.

4. Research of existing solutions of the problem

Despite the rather long history of scientific and technical research and practical experience in operating refrigeration machines, the influence of the hydrodynamics of the processes in the main units of the circuit and its connecting elements on the energy efficiency of the refrigeration cycle has not been adequately reflected in the available scientific publications. At the same time, depending on the technological function of the refrigeration machine and the layout of the main equipment consuming cold, the influence of this factor can be quite significant.

In [4], the influence of the hydrodynamics of its piping path on energy efficiency is studied for a heat pump. For well-known reasons, a number of conclusions obtained in this work, from a qualitative point of view, can be extended to the assessment of the energy efficiency of a refrigeration machine. A computational experiment performed in [4] showed that pressure losses in the connecting pipelines have a significant impact on the efficiency of the heat pump. The paper gives two cycles for pipelines of different lengths with a diameter of 15 mm connecting the evaporator to the compressor-condensation unit. It is shown that the cycle parameters change with increasing length of the pipeline. The role of the pressure drop at the inlet to the compressor is especially highlighted, since this leads to a decrease in the density of the refrigerant, and, consequently, the mass flow rate in the circulation circuit. To quantify the effect of pressure losses, the authors of this work considered a heat pump with R22 refrigerant when using atmospheric air with an average temperature of 21 °C as a low-potential heat source. The temperature of the refrigerant at the inlet to the evaporator is assumed to be 10 °C, and the condensation temperature after cooling the steam is 50 °C. In the calculation study, the diameter of the connecting pipelines d_p is taken equal to 12, 16 and 20 mm. It is shown that with $d_p = 15$ mm, with an increase in the length of the pipeline L_p from 2 m to 15 m, the heating capacity of the installation decreases accordingly by 16.2 %. In addition, in the same work it is shown that with an increase in the length of the connecting pipeline due to an increase in pressure losses, the heating capacity of the heat pump decreases the more, the smaller the diameter of the pipeline. So, with a pipeline length of 15 m and a decrease in its diameter from 20 mm to 12 mm, the heating capacity

decreased by 17.1 %. The question remains of assessing the effect of thermohydraulic irreversibility in the cycle for mixed refrigerants.

A physical model and quantitative estimates of the effect of pressure losses in the suction and discharge pipelines of a refrigeration machine are given in [5]. It is shown that, if, for example, the pressure loss during the absorption of refrigerant into the compressor is 0.1 bar, then at a boiling point of 15 °C the loss of cooling capacity will be 9.2 %. At the same time, the power on the compressor shaft (for a given cooling capacity) should be increased by almost 7 %, which is mainly due to an increase in the compression ratio by 3.5 %. The calculation performed in [5] with the same initial data showed that a pressure loss in the discharge pipeline of about 0.2 bar (at a condensation temperature of 30 °C) leads to a 1.3 % loss in cooling capacity. A concomitant increase in the compression ratio by 1.66 % leads to an increase in the required power on the compressor shaft by 2.8 %. Regarding the pressure loss in the liquid pipeline in [5] there is no quantitative assessment. Only the negative effect of reducing the flow rate through the throttle valve due to a decrease in inlet pressure is noted. The indicated decrease in flow rate can be enhanced due to the possible spontaneous vaporization that occurs with a significant pressure loss.

A decrease in the cooling capacity of RM and an increase in the energy consumed by it with an increase in pressure losses in the connecting pipelines are also indicated in [6]. Here, as in [4], it is noted that pressure losses in the suction pipelines have a special effect on cooling capacity. Pressure losses in the liquid pipeline can be considered insignificant. The authors of [6] estimate the pressure losses by the equivalent temperature difference along the pipeline. At the same time, specific recommendations are given for designing: a decrease in the boiling point, equivalent to the corresponding pressure drop, should not exceed 1–2 K. Based on this, recommendations are given in [6] for optimal refrigerant movement velocity in refrigerant pipelines for various refrigerants: on lines suction (from 5 m/s to 40 m/s) and discharge (from 8 m/s to 30 m/s). The latter is somewhat contrary to the correct observation of the authors themselves about the special effect of pressure losses in the suction pipelines.

In [7], the results of calculating the non-isothermal index (temperature glide) for mixed non-azeotropic refrigerants R32/R134a and R32/R152a are presented. It is determined that for the R32/R134a mixture in the boiling point range 10–50 °C, the temperature glide decreases from 6.3 K to 5.1 K with increasing boiling point. In the same boiling point range for the R32/R152a mixture, the temperature glide decreases from 7.2 K to 6.2 K. Calculations show that when the aggregate state of the mixture changes, the temperature glides increases the latent heat of vaporization. The positive effect of the temperature glide on the energy efficiency of the heat pump installation is revealed. So, for example, the non-isothermality of mixture reduces the compression work in the compressor. When determining the effective composition of the mixture, the maximum allowable limit of the working discharge pressure of compressor for R32 refrigerant (up to 4.9 MPa) is taken into account. To achieve high heating capacity of the installation at permissible pressures (up to 2.5 MPa), the concentration of component R32 in the mixture R32/R134a

should be 30 %. The permissible discharge pressure for the mixture R32/R152a is 2.1 MPa. However, the author of [7] investigated the influence of only the temperature glide in the evaporator and condenser in the presence of a limitation of the permissible compressor discharge pressure. Meanwhile, when designing the hydraulic circuit of the heat pump, it is necessary to take into account the permissible restrictions on the loss of steam pressure at the suction in the compressor, as well as on the pressure loss in the liquid pipeline. Neglect of these factors can lead to an overestimated design energy efficiency of the installation, as well as to erroneous mass and size characteristics of its elements.

In [8], experimental studies are conducted to control the heating capacity of the heat pump by changing the composition of the mixture R32/R134a (40/60 %, 70/30 %, 50/50 %, respectively). During the experiments it is found that the heating capacity of the installation should be selected by regulating the mixed composition of the refrigerants. So, for example, to increase the heating capacity, the authors recommend enriching the composition of the mixture R32/R134a with component R32. With an increase in the percentage of R32 in the composition of the mixture R32/R134a, the capacity of the installation increases, both in heating and in cooling. At the same time, the authors of this work note that the refrigeration coefficient of an installation operating on a R32/R134a mixture increases only due to the effect of non-isothermality of mixture. This, in turn, indicates that the effective working composition of the mixture should be determined not from the condition of ensuring the maximum capacity of the installation, but from the condition of achieving maximum exergy efficiency with variability of the values of the minimum temperature head in the evaporator and condenser.

The papers [9, 10] describe the theoretical thermodynamic characteristics of a steam compression refrigeration system using a mixture as an alternative to replace the R22 refrigerant. In these studies, a thermodynamic analysis of the air conditioner is carried out with R431A, R410A, R419A, R134a, R1270, R290 and fifteen mixtures of refrigerants consisting of R134a, R1270 and R290. The authors constructed a model of the real thermodynamic cycle taking into account pressure losses in the processes of evaporation and condensation, as well as pressure losses in the connecting pipelines. The studied mixtures of refrigerants are evaluated at a condensation temperature of 54.4 °C and a boiling point of 7.2 °C. The results show that the thermodynamic efficiency for mixed refrigerants R134a/R1270/R290 (50/5/45 by weight in percent) is generally 2.10 % higher than for refrigerants R22, R431A, R410A, R419A, R134a, R1270 and R290. It should be noted that for a complete and final assessment of the thermodynamic efficiency of the mixed refrigerant cycle, it is necessary to take into account the exergy destructions in each element.

In [11, 12], thermodynamic models of a refrigeration unit for analyzing the exergy efficiency of a cycle with mixed refrigerants R502, R404A, R507 are presented. The models make it possible to evaluate the effect of exergy losses in the installation elements depending on the operating parameters of the installation, as well as the ambient temperature. The effect of pressure losses in the evaporator and condenser on the exergy efficiency of the installation is taken into account. A limitation of these models is the lack

of recalculation of the values of boiling point and condensation temperature with allowance for pressure losses in the cycle. In addition, the authors do not consider the effect of changes in the concentrations of the components of the mixture on the energy characteristics of the installation.

In [13], a mathematical model is given for predicting the energy characteristics of a small vapor-compression refrigeration machine when the heat engineering efficiency of the evaporator and condenser changes. As refrigerants are considered mixed R407C, R410A and single-component refrigerants R134a. When constructing the model, pressure losses in the real cycle are not taken into account. In addition, the isentropic efficiency of the compressor is set constant, which is permissible only as a first approximation in the calculation of the cycle.

Thus, the analysis results allow to conclude that hydraulic resistance in the pipelines of both refrigeration machines and heat pumps have a significant impact on their energy efficiency. Moreover, the information available in the literature on the problem under consideration is fragmentary and does not reflect the effect of thermo-hydraulic irreversibility in the cycle for mixed refrigerants. This determines the prospects of studies establishing the relationship between the energy efficiency of refrigeration machines and heat pumps and hydraulic resistance in pipelines connecting the main equipment, as well as in the pipe system of the condenser and evaporator.

5. Methods of research

In Fig. 1 in the P,i-diagram, the RM cycle, taking into account the influence of hydraulic losses along the mixture flow path, is designated as $1'-2'-2_d-3-4-5'-5^*-6'-7-8$. The cycle without losses is $1-2-3-4-5-6-7$. This diagram illustrates two stages of calculating the parameters of the cycle, which form the basis of the method. At the first stage of the calculation, the parameters are determined at the nodal points of the cycle without taking into account pressure losses. The second stage involves the introduction of refinements into an already formed cycle, due to the non-isobaric processes in the circulation circuit.

In Fig. 1 marked:
ΔP_c – pressure drop in the condenser;
ΔP_e – pressure drop in the evaporator;
ΔP_s – pressure loss in the suction pipeline;
ΔP_d – pressure loss in the discharge pipeline;
ΔP_l – pressure loss in the liquid pipeline;
ΔT_{oh} – overheating value;
ΔT_{sc} – subcooling value;
ΔT_e, ΔT_c – non-isothermality values in the evaporator and condenser without taking into account pressure losses;
$\Delta T''_e$, $\Delta T''_c$ – the same, only taking into account pressure losses.

The construction of a cycle taking into account the non-isobarity of the processes of evaporation and condensation requires setting as fixed parameters of the mixture at the beginning or at the end of the process. In relation to the design calculation, the states of the refrigerant on the right boundary curve are considered fixed (points 3 and 7 in Fig. 1). To analyze the effect of pressure losses in the cycle on the energy characteristics of a real RM, the thermodynamic parameters of the mixture at points 3 and 7 are set in a first approximation. For subsequent approximations, the method provides for a recalculation

procedure that takes into account the increase in the temperature of the refrigerant at the beginning of the condensation process (point 3) by a value equal to the change in the average thermodynamic temperature in this process as a result of pressure losses in the condenser ΔP_c. The recalculation procedure also provides for a decrease in temperature at the end of the evaporation process (point 7) by a value equal to a change in the average thermodynamic temperature due to a pressure drop in the evaporator ΔP_e.
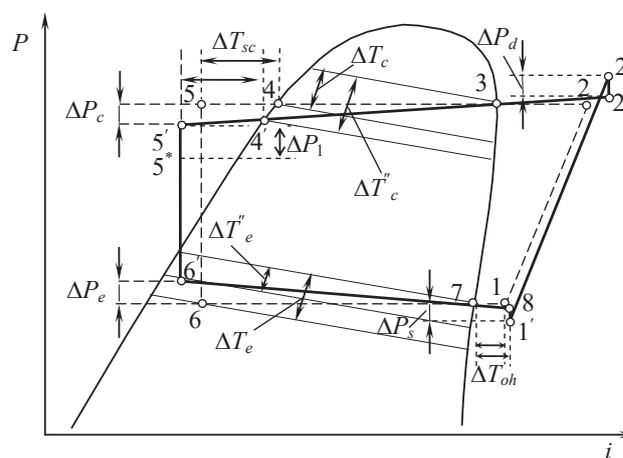
Fig. 1. The real cycle of the refrigeration machine for non-azeotropic mixture

The average thermodynamic (dissipative) temperatures of the refrigerant in non-isobaric processes of evaporation and condensation are found as:

$$T_{av}^p = \frac{\int\limits_{in}^{out} \mathrm{d}i}{\sum\limits_{j=1}^{n} \dfrac{\int \mathrm{d}i}{T_j}}, \tag{1}$$

where T_j – the average logarithmic temperature in the selected zone of the heat exchanger. For the evaporator, these are the zones of actual evaporation and overheating. In the condenser, zones for removing overheating, condensation and a zone of liquid subcooling (if any) are allocated.

As can be seen from Fig. 1, pressure losses along the path of the evaporator and condenser affect the non-isothermality of mixture. If for a condenser it increases with increasing pressure loss, then for the evaporator the opposite situation takes place – the non-isothermality of mixture decreases. So, for example, in the absence of pressure losses in the evaporator, the temperature of the mixture on the right boundary curve is always higher than the temperature of the two-phase flow at the inlet to the evaporator. In the non-isobaric process of evaporation in the cycle, the opposite situation occurs when the temperature of the mixture at the outlet of the evaporation zone is lower than the temperature at the inlet.

Based on the constructions of the non-isobaric cycle (Fig. 1), it is possible to carry out an integrated assessment of the hydraulic efficiency of the refrigerant circuit:

$$\eta_{hidr} = \frac{i_2 - i_1}{i_2 - i_4} \cdot \frac{i_{2d} - i_{4'}}{i_{2d} - i_{1'}}. \tag{2}$$

For the element-wise analysis of losses from irreversibility, a grapho-analytical method is proposed for taking into account the non-isobarity of evaporation and condensation processes in a cycle.

Grapho-analytical constructions on the T,s-diagram for a non-isobaric cycle are presented in Fig. 2.

Fig. 2. Grapho-analytical method for taking into account non-isobarity in the cycle of a refrigeration machine with a non-azeotropic mixture

To determine the degree of thermodynamic perfection of the non-isobaric cycle $1-2_d-3-4-5-6-7$, let's construct the theoretical cycle without taking into account the non-isobarity $1^T-2^T-3-4^T-5^T-6^T-7$ and, from it, the reversible sample cycle $m_1-m_2-m_3-m_4$.

A reversible sample cycle is constructed under the condition of maintaining a constant positive effect of RM, i. e., the constancy of its cooling capacity in all cycles. To ensure this equality, the following rebuilding should be carried out. Point m_1 should lie at the intersection of the line $i=$const and line $(8-9)$. The process of cooling the coolant in the evaporator $(8-9)$ is conditionally displayed on the T,s-diagram, and in order to simplify the analysis (as a first approximation), the nature of process $8-9$ should coincide with the nature of non-isothermality of the phase transition process in the evaporator. The point m_2 is defined as the end of the isentropic compression process m_1-m_2 at the intersection of the line $(10-11)$. The point m_3 is defined on the line $(10-11)$ as the beginning of the adiabatic expansion process m_3-m_4, taking into account the position of the point m_4. It should be noted that the process of heating the coolant in the condenser $(10-11)$, as well as the process $(8-9)$, is only conditionally displayed on the T,s-diagram and should coincide in nature (in a first approximation) with the nature of the non-isothermality of phase transition in the condenser. Point m_4 should lie at the intersection of the reconstructed line from point 6^T and the line of the coolant cooling process $(8-9)$. In this case, equality in cooling capacity is observed:

– reversible sample cycle $q_{0(m1-m2-m3-m4)}=i_e-i_g\equiv$ area of $(e-g-m_1-m_4)$;

– the theoretical cycle without taking into account the non-isobarity $q_{0(1^T-2^T-3-4^T-5^T-6^T-7)}=i_f-i_g\equiv$ area of $(f-7-6^T-g)$;

– non-isobaric cycle $q_{0(1-2d-3-4-5-6-7)}=i_e-i_d\equiv$ area of $(e-r-6-d)$.

Loss of work due to the irreversibility of heat transfer in the condenser and the presence of hydraulic resistance along the path of movement of the refrigerant is equivalent to the area of $(2-3-4^T-a-b-d-q-p)$. Moreover, the area of $(4^T-a-b-s)$ is equivalent to losses from irreversibility due to friction in the condenser. Loss of work due to the irreversibility of the expansion process in the throttle device can be represented by the area of $(d-q-5-z)$. Loss of work in the evaporator as a result of irreversibility of heat transfer and the presence of hydraulic resistance can be represented on the T,s-diagram in the form of several areas: $(6-m_4-m_3-q)$, (m_4-6^T-7-u). Moreover, the area of $(f-u-m_1-e)$ reflects losses due to friction during evaporation. Losses from the irreversibility of the compression process in the compressor can be shown as the area of $(n-11-p-y)$. The friction heat equal to the friction work will be only the area under the curve of the real process $1-2_d$, i. e. $(n-2_d-1-w)$. The area of $(1-2_d-2-1^T)$ represents an increase in the total work of compression caused by the supply of friction heat and an increase in the specific volume of gas.

Losses of exergy in an ordinary regenerative cycle caused by irreversibility of heat transfer in a regenerative heat exchanger (processes $(4-5)$ and $(1-7)$) are not separately distinguished in the analysis of the cycle. They are included as components of the losses of the remaining elements of RM. In this sense, losses in the regenerative heat exchanger are «latent internal losses of cycle». The diagram can only show losses due to friction during steam overheating. These losses are equivalent to the area of $(w-1-1^T-y)$.

6. Research results

6.1. The effect of friction on the losses of exergy in the cycle of the refrigeration machine. In the present work, an attempt is made, based on an exergy method, to consider determining the energy costs for compensating the irreversibility of the friction process at a temperature level below the ambient temperature T_a.

It is known that in addition to the generated cold in the refrigeration machine due to the presence of friction, heat is generated that must be removed from the system to the environment. Often, in calculations, the heat released during friction is equated to the work spent on overcoming the friction forces. This is permissible only as a first approximation. During friction at a temperature level significantly lower than the ambient temperature, one more important indicator must be taken into account – exergy of the friction heat [14]. The exergy of the heat of friction is the minimum work necessary to transfer this heat from the temperature level of cold generation to the environment. Thus, the RM compressor consumes energy not only to overcome friction in the hydraulic circuit of the refrigerant circulation, but also carries out a kind of «cleaning of the system» from the heat generated at the low temperature level, removing it to the environment.

To illustrate the friction process at a temperature level below the ambient temperature in terms of the exergy method, let's use the constructions in the T,s-diagram (Fig. 3).

The destruction of exergy due to friction can be represented on the T,s-diagram with the area of $(a-b-c-d)$ and is determined by the dependence:

$$E_{D,k}^{\Delta p} = T_a S_k^{\Delta p} = A_{Q_f}^{T_h} = T_a \frac{Q_f}{T_h} = T_a \frac{m_r v \Delta p}{T_h} = T_a \frac{N_f}{T_h}, \qquad (3)$$

where $S_k^{\Delta p}$ – generation of entropy due to friction; $A_{Q_f}^{T_h}$ – anergy of the friction heat; Q_f – friction heat; T_h – the average temperature during friction; m_r – the mass flow rate of the refrigerant; v – the specific volume of the refrigerant; Δp – pressure losses to overcome friction; N_f – energy consumption in the compressor to overcome friction in the process.
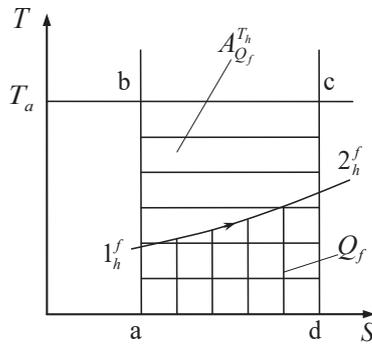
Fig. 3. A graphical representation of the friction process at a temperature level below ambient temperature

Since $T_h < T_a$, then:

$$E_{D,k}^{\Delta p} = A_{Q_f}^{T_h} > N_f = Q_f, \qquad (4)$$

$$Q_f = A_{Q_f}^{T_h} + E_{Q_f}^{T_h}, \qquad (5)$$

where $E_{Q_f}^{T_h}$ – exergy of the heat of friction, which is found from the expression:

$$E_{Q_f}^{T_h} = Q_f \left(1 - \frac{T_a}{T_h}\right). \qquad (6)$$

The friction heat Q_f on the T,s-diagram can be represented by the area of (a–1–2–d).

The energy consumption in the compressor to overcome friction in the process:

$$N_f = Q_f = A_{Q_f}^{T_h} - \left|E_{Q_f}^{T_h}\right|. \qquad (7)$$

Thus, the destruction of exergy at a temperature level below the ambient temperature is numerically greater than the energy consumed by the compressor, necessary to overcome friction:

$$E_{D,k}^{\Delta p} = A_{Q_f}^{T_h} = Q_f + \left|E_{Q_f}^{T_h}\right|. \qquad (8)$$

Let's evaluate the value of energy expenditures for overcoming friction in the RM evaporator, as well as the value of exergy destruction associated with hydraulic losses along the evaporator path.

RM features:
– a mixture of R407C;
– evaporator and air-cooled condenser;
– theoretical compressor volumetric flow rate $V_T=$ $=0.0821$ m³/s;
– pressure loss in the evaporator $\Delta p = 0.5$ bar;

– temperature at the end of the evaporation process $T_0^e = -21.4$ °C;
– average temperature in the friction zone $T_h = -20.81$ °C;
– ambient temperature $T_a = 20$ °C;
– air temperature at the inlet to the evaporator $T_e^{in} =$ $=-15$ °C;
– mass flow rate of the refrigerant $m_r = 0.720$ kg/s;
– refrigerant specific volume $v = 0.112$ m³/kg.
Calculation results:

$$E_{D,k}^{\Delta p} = T_a \frac{m_r v \Delta p}{T_h} = 293.15 \frac{0.720 \cdot 0.112 \cdot 50}{252.34} = 4.684 \text{ kW},$$

$$E_{Q_f}^{T_h} = Q_f \left(1 - \frac{T_a}{T_h}\right) = 4.03 \left(1 - \frac{293.15}{252.34}\right) = -0.649 \text{ kW},$$

$$N_f = A_{Q_f}^{T_h} - \left|E_{Q_f}^{T_h}\right| = 4.684 - 0.649 = 4.03 \text{ kW}.$$

As can be seen from the calculation, at $T_h < T_a$, exergy destruction $E_{D,k}^{\Delta p}$ is greater than N_f. This conclusion has important methodological significance in assessing the exergy cost of the generated cold, taking into account the thermodynamic perfection of the refrigeration unit.

A comparative analysis of the known thermoeconomic models of the formation of the exergy cost of cold, carried out in [15], showed significant differences in the cost of cold due to neglect of the effect of friction.

Table 1 presents the results of an advanced exergy analysis of the real cycle of RM with R407C refrigerant.

Table 1

The exergy analysis results of the real cycle of refrigeration machine with R407C

RM element	$E_{D,k}^{UN}$, kW	$E_{D,k}^{AV}$, kW	$E_{D,k}^{EN}$, kW	$E_{D,k}^{EX}$, kW
Compressor	1.069	10.901	10.405	1.565
Condenser	1.021	9.259	0.5356	9.694
Evaporator	4.141	2.186	6.325	0
Throttle	4.37	5.955	1.83	8.495
System	10.601	28.301	19.09	19.754

The calculation task is to identify the contribution of certain types of components of exergy losses in order to increase the thermodynamic efficiency of the elements and the system as a whole.

Table 1 shows the following indicators: unavoidable $E_{D,k}^{UN}$ and avoidable $E_{D,k}^{AV}$, as well as endogenous $E_{D,k}^{EN}$ and exogenous $E_{D,k}^{EX}$ components of exergy destruction.

To determine the unavoidable $E_{D,k}^{UN}$ and avoidable $E_{D,k}^{AV}$ destructions of exergy, let's use the method of constructing theoretical and quasi-theoretical cycles [16]. Grapho-analytical method and conditions for constructing cycles are given in subsection 6.3 of this work.

When constructing a quasi-theoretical cycle with unavoidable losses in the elements, the following conditions are accepted. The compression process is considered non-isentropic (indicator efficiency of compressor $\eta_{is}=0.95$). The expansion process is irreversible (throttling). The minimum temperature head in the evaporator and condenser is assumed to $\Delta T_{min}^{UN} = 0.5$ K. Such initial data for constructing a quasi-theoretical cycle are due to technical limitations

associated with the maximum achievable at the current level of development of production of heat engineering and technological characteristics of heat exchange and compressor equipment.

To find $E_{D,k}^{EN}$ and $E_{D,k}^{EX}$ let's use the method of constructing hybrid cycles [17]. When constructing a hybrid cycle, processes in all elements are idealized, except for the process in the element in question. Thus, a cycle with only one irreversibility is analyzed, in other processes irreversible losses are «mentally» eliminated.

The results of the calculation of the endogenous and exogenous parts of the destruction of exergy in the elements showed that those elements for which $E_{D,k}^{EN} > E_{D,k}^{EX}$ are primarily subject to improvement. As can be seen from the Table 1, such RM elements are a compressor and an evaporator. In the case of a condenser, the value of $E_{D,k}^{EN} < E_{D,k}^{EX}$ indicates the need to reduce irreversibility in other elements of the installation, which should ultimately affect the increase in the efficiency of the condenser.

The proportion of the components of the destruction of exergy in the elements of RM is given in Table 2.

Table 2

The shares of the unavoidable $E_{D,k}^{UN}$, avoidable $E_{D,k}^{AV}$, endogenous $E_{D,k}^{EN}$ and exogenous $E_{D,k}^{EX}$ components of the destruction of exergy in the elements of the refrigeration machine

RM element	$E_{D,k}^{UN}$, %	$E_{D,k}^{AV}$, %	$E_{D,k}^{EN}$, %	$E_{D,k}^{EX}$, %
Compressor	8.93	91.07	86.93	13.07
Condenser	9.93	90.07	5.24	94.76
Evaporator	65.45	34.55	100	0
Throttle	42.32	57.68	17.72	82.28
System	27.25	72.75	49.15	50.85

As can be seen from the Table 2, a rather high proportion of the unavoidable destruction of exergy takes place in the evaporator $E_{D,k}^{UN} = 65.45$ %. Therefore, for the above features of RM operation, the desirability of constructive improvement of the evaporator is secondary. The share of unavoidable destruction in it substantially depends on the temperature mode of the installation. First of all, it is necessary to consider the possibility of improving the elements with a high share of $E_{D,k}^{AV}$, which are the compressor and condenser.

6.2. The effect of the composition of the multicomponent mixture R32/R125/R134a on the energy efficiency of the refrigeration cycle of an autonomous air conditioner. Solving the problem of choosing the effective composition of a multicomponent mixture of refrigerants requires the use of experiment planning theory methods [18]. Since the sum of the fractions of all the components that make up the mixture is equal to 1, the factor space can be represented by the correct simplex, and for the three-component mixture by the right triangle. The properties of the composition of the mixture are studied at predetermined points of the simplex. Processing the results of a numerical experiment is carried out using a simplex centroid plan.

The thermodynamic properties of the mixture are calculated using the certified database REFPROP 7.0 (Benedict-Web-Rubin equation of state) [19]. The calculation of pressure losses during in-pipe boiling and condensation of

the mixture is carried out according to the recommendations of [20, 21]. Pressure losses in connecting pipelines during a single-phase medium flow are determined using the Darcy equation [5]. The calculation of the static heat engineering characteristics of the air conditioner is carried out using the method [22, 23].

Initial data includes the following values:
– molar fractions of the components of the mixture X;
– cooling capacity Q_0;
– boiling point T_7 and condensation temperature T_3 on the right boundary curve;
– overheating in the evaporator ΔT_{oh};
– subcooling in the condenser ΔT_{sc};
– fictitious lengths of pipelines of heat exchangers and connecting pipelines;
– pipe diameters.

The mode-geometric parameters of the refrigerant circuit are as follows. Air evaporator with a cooling capacity of 10 kW. The diameter of the evaporator tubes is 0.015 m, the number of strokes is 4, the length of one stroke is 7 m. An air condenser with the diameter of the pipes is 0.015 m, the number of strokes is 6, the length of one stroke is 7 m. Connecting pipelines: discharge – 4.51 m long and 0.012 m in diameter; suction – 3 m long, 0.015 m in diameter; liquid – 6.8 m long, 0.012 m in diameter.

The temperature boundaries of the thermodynamic cycle (Fig. 1) are taken as follows: $T_7 = 5$ °C; $T_3 = 40$ °C; $\Delta T_{oh} = 6$ °C; $\Delta T_{sc} = 5$ °C.

Below, Fig. 4–7 show the dependences of pressure losses in the evaporator and connecting pipelines ΔP (kPa) obtained for a given hydraulic circuit of the refrigerant circulation while simultaneously varying at different levels the mole fractions of the mixture components R32, R125, R134a.

ΔP_e, kPa
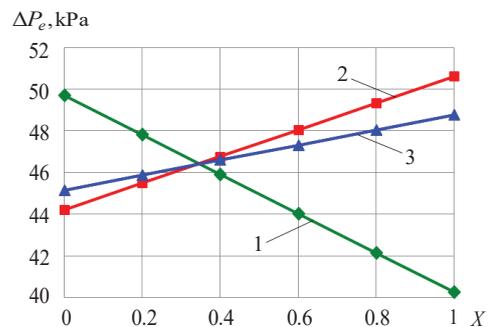
Fig. 4. The pressure losses in the evaporator, depending on the change in the molar fraction of the variable component of the mixture R32, R125, R134a: 1 – R32; 2 – R125; 3 – R134a

ΔP_l, kPa
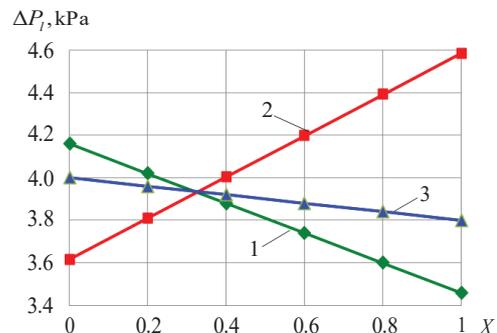
Fig. 5. The pressure losses in the liquid pipeline, depending on the change in the molar fraction of the variable component of the mixture R32, R125, R134a: 1 – R32; 2 – R125; 3 – R134a
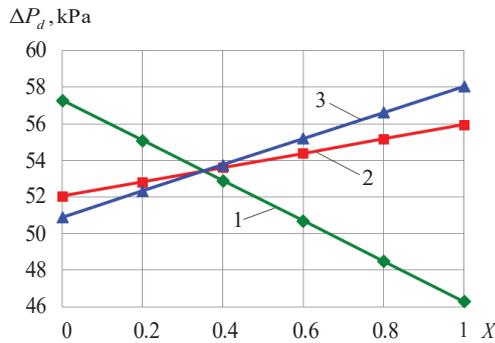
Fig. 6. Pressure losses in the discharge pipeline, depending on the change in the molar fraction of the variable component of the mixture R32, R125, R134a: 1 – R32; 2 – R125; 3 – R134a

Fig. 8. The temperature glide (non-isothermality) of the mixture of R32, R125, R134a in the evaporator, depending on the change in the molar fraction of the variable component: 1 – R32; 2 – R125; 3 – R134a

The nature of the dependence of the air conditioning coefficient of performance (COP) on the change in the molar fraction of the variable component of the mixture is shown in Fig. 9.

Fig. 7. Pressure losses in the suction pipeline, depending on the change in the molar fraction of the variable component of the mixture R32, R125, R134a: 1 – R32; 2 – R125; 3 – R134a

Fig. 9. The coefficient of performance of the air conditioner, depending on the change in the molar fraction of the variable component of the mixture R32, R125, R134a: 1 – R32; 2 – R125; 3 – R134a

The analysis of the dependences showed that an increase in the molar fraction of R32 refrigerant in the mixture always leads to a decrease in pressure losses in the evaporator and connecting pipelines. Meanwhile, the influence of the concentrations of the two other components of the mixture R125 and R134a is far from unambiguous and primarily depends on their state of aggregation in the elements of the hydraulic circuit. As can be seen from Fig. 5, an increase in the molar fraction of R134a leads to a decrease in pressure losses in the liquid pipeline. At the same time, for the evaporator (Fig. 4), as well as for the suction and discharge pipelines (Fig. 6, 7), an increase in the proportion of R134a, on the contrary, increases the pressure losses. With an increase in the molar fraction of R125, a sharp increase in pressure losses occurs in the evaporator, as well as in the liquid pipeline (Fig. 4, 5). On the other hand, an increase in the proportion of R134a has a greater influence on the increase in pressure losses in the suction and discharge pipelines (Fig. 6, 7).

The intersection point of the graphs (Fig. 4–7) corresponds to equal molar fractions of all components of the mixture. In this regard, in order to find the effective composition of the mixture, it is necessary to set the molar fractions of one of the selected components, and for the rest, recalculate their concentrations from the condition that the sum of all molar fractions should be equal to 1.

The effect of changes in the composition of the mixture on the non-isothermality of evaporation process is shown in Fig. 8. It is seen that the value of ΔT_e increases with increasing molar fraction of the component R134a. On the other hand, with an increase in the molar fraction of R125, the value of ΔT_e decreases.

As can be seen from Fig. 9, the coefficient of performance is sensitive to changes in the mole fractions of all components of the mixture. With an increase in the mole fraction of R134a, a decrease in COP is observed, which is due to an increase in hydraulic losses in the refrigerant circuit. At the same time, an increase in the molar fractions of the components R32 and R125 leads to an increase in COP due to a decrease in the compression ratio in the cycle, and, as a consequence, a reduction in the energy consumption for compressor drive.

6.3. Application. To determine the unavoidable and avoidable destructions of exergy, the method of constructing theoretical cycles is used [16].

Fig. 10 shows three cycles:
– real cycle (1_R–2_R–$3^{//}_R$–$3^{/}_R$–4_R);
– quasi-theoretical cycle with unavoidable losses (1_{RU}–2_{RU}–$3^{//}_{RU}$–$3^{/}_{RU}$–4_{RU});
– theoretical idealized cycle (1_T–2_T–$3^{//}_T$–$3^{/}_T$–4_T).

Conditions for constructing a theoretical idealized cycle (1_T–2_T–$3^{//}_T$–$3^{/}_T$–4_T). The processes of compression (1_T–2_T) and expansion ($3^{/}_T$–4_T) are assumed to be isentropic, i. e., $s_{1T}=s_{2T}$ and $s_{3T}=s_{4T}$. In a theoretical cycle, the throttling process is replaced by an ideal expansion process in a virtual expander. The condensation temperature on the right boundary curve (point $3^{//}_T$) should correspond to the temperature of the coolant at the outlet of

the condenser (point 7), and the temperature at the end of the evaporation process (point 1_T) should correspond to the temperature of the coolant at the inlet to the evaporator (point 8). The minimum temperature head in the condenser and evaporator in the theoretical cycle is taken equal to zero $\Delta T_{\min}=0$ K. The mass flow rate of the coolant in the condenser for the theoretical and real cycles are not $G_{6-7}^T \neq G_{6-7}^R$, while the mass flow rate of the coolant in the evaporator for the theoretical and real cycle are taken equal to $G_{8-9}^T = G_{8-9}^R$. This is due to the fact that the main condition for comparing cycles is the constancy of cooling capacity. Another condition for the analysis of the theoretical cycle is the requirement that the heating of the coolant in the condenser be greater than the non-isothermality of condensation process $T_3{}^{//}{}_T - T_3{}^{/}{}_T < T_7 - T_6$. The cooling of the coolant in the evaporator should also be less than the indicator of non-isothermality of the evaporation process $T_{1T} - T_4{}^{/}{}_T < T_8 - T_9$.
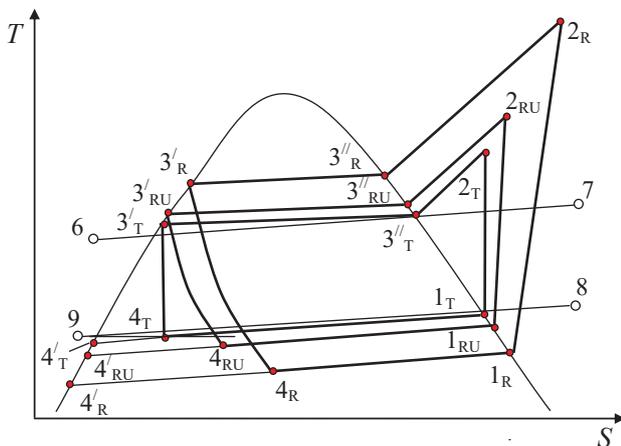
Fig. 10. Construction of real, quasi-theoretical and theoretical cycles for a non-azeotropic mixture of refrigerants

The theoretical cycle is the basis for constructing a quasi-theoretical cycle with unavoidable losses in the elements. The conditions for constructing such a cycle are as follows. The compression process ($1_{RU}-2_{RU}$) is non-isentropic ($\eta_{is}=0.95$). The minimum temperature head $\Delta T_{\min}^{UN}=0.5$ K is set in the evaporator and condenser. The expansion process takes place in a throttle device ($3'_{RU}-4_{RU}$).

6.4. Discussions of the results. The main results of the research can be formulated as follows:

1. Pressure losses along the refrigerant circulation path when using a non-azeotropic mixture have a significant effect on the non-isothermal phase transition in the evaporator and condenser. It is especially important to take into account the non-isobarity during the design of the evaporator, since pressure losses along the path cardinally affect the temperature change of the mixture during the phase transition.

2. The proposed method for taking into account non-isobarity in a cycle with a non-azeotropic mixture, by virtue of its visibility, allows to significantly simplify the numerical simulation of the thermodynamic parameters of the refrigerant at the nodal points of the cycle. Its grapho-analytical application in the T,s-diagram illustrates the nature of the redistribution of losses from irrever-

sibility in each RM element during the transition from a theoretical (isobaric) cycle to a non-isobaric one.

3. The effective composition of the non-azeotropic mixture must be selected by focusing on the energy characteristics of the hydraulic circuit of the refrigerant circulation and exergy efficiency. To calculate the RM parameters with variability of the composition of a non-azeotropic mixture, it is advisable to use the method proposed in this work. The effective composition of the mixture R32, R125, R134a, in which the considered autonomous air conditioner has a maximum exergy efficiency ($\eta_{ex}=0.20$) under the given operating conditions, is the composition R32 (0.329), R125 (0.249), R134a (0.422). From a thermodynamic point of view, an increase in the molar fractions of R32 and R125 in the mixture can increase the efficiency of the cycle, however, questions of reducing fire risk require additional research. For example, an excess of the percentage concentration of the component R32 by more than 30 % increases the risk of fire hazard [2].

4. The reliability of the results is confirmed using a certified database to determine the thermophysical properties of single-component and multi-component refrigerants REFPROP 7.0. As well as good agreement with the results of experimental studies of RM on the refrigerants R407C (R32 (23 %), R125 (25 %), R134a (52 %)) and R407A (R32 (20 %), R125 (40 %), R134a (40 %)) [24].

7. SWOT analysis of research results

Strengths. One of the advantages of the developed method is its versatility. Since it can be used both in the design and in the verification calculation of the refrigeration machine, as well as for a comprehensive assessment of irreversible losses in the cycle when conducting thermodynamic diagnostics of an existing installation.

Weaknesses. The weaknesses of the method include the lack of zone calculation of heat transfer in the evaporator and condenser. This, of course, reduces the accuracy of determining the heat transfer coefficient of heat exchangers. However, it should be noted that experimental studies to determine the heat transfer coefficient during boiling and condensation of mixed refrigerants are expensive and time-consuming, therefore, the publication of these results in the open literature is extremely limited.

Opportunities. Promising research should include thermoeconomic optimization. Since the proposed method is based on the block-modular principle of constructing a mathematical model, in the future it may include a block of thermoeconomic optimization of the parameters of the refrigeration machine.

Threats. The closest analogue to this development is the program for calculating the isobaric thermodynamic cycle of a vapor-compression refrigeration machine with mixed refrigerants KleaCalc.

8. Conclusions

1. As a result of a numerical experiment, based on the proposed method for taking into account non-isobarity in the cycle, the effect of changes in the concentration of components of a non-azeotropic mixture (R32, R125, R134a) on pressure losses in the refrigerant circuit is studied. It has been established that the most significant influence on pressure losses in the refrigeration circuit of

an autonomous air conditioner is exerted by a change in the concentrations of components R125 and R134a.

2. Based on the exergy methodology, the energy costs for compensating the irreversibility of the friction process at a temperature level below the ambient temperature T_a have been determined. It is established that the destruction of exergy due to friction in the evaporator is greater than the energy consumption in the compressor to compensate for friction in the evaporator.

3. The advanced exergy analysis of the refrigeration cycle with a non-azeotropic mixture shows that the evaporator has the highest share of unavoidable exergy destruction $E_{D,k}^{UN} = 65.45$ %. The largest share of avoidable destruction has a compressor $E_{D,k}^{AV} = 91.07$ % and a condenser $E_{D,k}^{AV} = 90.07$ %. A high share of exogenous destruction has a condenser $E_{D,k}^{EX} = 94.76$ %.

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