Ministry of Education and Science of Ukraine
Black Sea Universities Network

ODESA NATIONAL UNIVERSITY OF TECHNOLOGY

International Competition of Student Scientific Works

BLACK SEA SCIENCE 2022

PROCEEDINGS

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Odesa National University of Technology

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Proceedings of International Competition of Student Scientific Works «Black Sea Science 2022» contain the works of winners of the competition.

The author of the work is responsible for the accuracy of the information.

Odesa National University of Technology, 2022.

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INTRODUCTION

International Competition of Student Scientific Works “Black Sea Science” has been held annually since 2018 at the initiative of Odesa National University of Technology (formerly Odesa National Academy of Food Technologies) with the support of the Ministry of Education and Science of Ukraine. It has been supported by Black Sea Universities Network (the Association of 110 higher education institutions from 12 countries of the Black Sea Region) since 2019, and by Iseki-FOOD Association (European Integrating Food Science and Engineering Knowledge into the Food Chain Association) since 2020.

The goal of the competition is to expand international relations and attract students to research activities. It is held in the following fields:

- Food science and technologies
- Economics and administration
- Information technologies, automation and robotics
- Power engineering and energy efficiency
- Ecology and environmental protection

The jury includes both Ukrainian and foreign scientists. In the 4 years that the competition has been held, the jury included scientists from universities of 24 countries: Angola, Azerbaijan, Benin, Bulgaria, China, Czech Republic, France, Georgia, Germany, Greece, Israel, Italy, Kazakhstan, Latvia, Lithuania, Moldova, Pakistan, Poland, Romania, Serbia, Slovakia, Switzerland, Turkey, USA.

At the same time, every year the geography has expanded and the number of foreign jury members has increased: from 46 jury members representing 25 universities from 12 countries in 2018, to 73 jury members of the 46 universities from 19 countries in 2022.

More than a thousand student research papers have been submitted to the competition from both Ukrainian and foreign institutions from 25 countries: China, Poland, Mexico, USA, France, Greece, Germany, Canada, Costa Rica, Brazil, India, Pakistan, Israel, Macedonia, Lithuania, Latvia, Slovakia, Romania, Kyrgyzstan, Kazakhstan, Bulgaria, Moldova, Georgia, Turkey, Serbia.

The interest of foreign students in the competition grew every year. In 2018, the students representing 15 institutions from 7 countries have submitted 33 works. In 2021 the number of submitted works increased to 73, authored by the students of 40 institutions from 18 countries.

The competition is held in two stages. In the first stage, student research papers are reviewed by members of the jury who are experts in the relevant fields. In the second stage of the competition, the winners of the first stage have the opportunity to present their work to a wide audience in person or online.

All participants of the competition and their scientific supervisors are awarded appropriate certificates, and the scientific works of the winners are included in the electronic proceedings of the competition. Every year the competition receives a large number of positive responses from Ukrainian and foreign colleagues with the desire to participate in the coming years.

4. POWER ENGINEERING AND ENERGY EFFICIENCY

ENERGY EFFICIENT CIRCUIT SOLUTIONS FOR LOW-TEMPERATURE REFRIGERATION MACHINES BASED ON ENVIRONMENTALLY FRIENDLY REFRIGERANTS

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Abstract. *Modern refrigeration equipment and technology has two trends: energy efficiency and strict environmental requirements for substances used as refrigerants. Energy efficiency requires the use of refrigeration cycles that have minimal irreversible energy losses and are as close as possible to the Carnot cycle. This is especially true for low-temperature refrigeration machines, where to implement the cycle it is necessary to overcome a significant pressure drop. In the field of low-temperature refrigeration technology, energy-efficient solutions based on two- and multi-stage, as well as cascade refrigeration machines are used. A comparison of their effectiveness in terms of COP is presented in this paper.*

The environmental aspect of the problem today is solved through the use of natural and some artificial refrigerants, and is assessed primarily by LCCP, GWP and ODP. The paper considers refrigeration cycles based on R717, as well as its combinations with R23, R290, R744 and R32, used in the lower stage. For comparison, the result of the COP calculation using the base ozone-depleting refrigerant for the lower stage, which is R13, is presented.

The paper also analyzes the influence of the choice of the intermediate temperature of the phase transition in the evaporator-condenser on the energy efficiency of the cycle for the ratio of refrigerants of the upper and lower stages, which has the highest energy efficiency.

The practical significance of the work is to provide guidelines for further experimental studies of energy efficient solutions of low-temperature refrigeration machines.

Keywords: *thermal calculation, refrigeration machine, COP, cascade cycle.*

I. INTRODUCTION

Artificial cold has become one of the main indicators of the level of technical and cultural development of any country. The current stage of development of the refrigeration industry is characterized by the widest use of cold in all sectors of the economy. In fact, there are no industries where the cold is not used. [1] Many technological processes require the use of low temperatures (not higher than –40 °C), which can be created and maintained by low-temperature refrigeration machines. At the same time, they must have high energy efficiency, which is not achieved by using conventional single-stage steam compression refrigeration machines (SCRM). Therefore, it is necessary to apply other circuit solutions, among which are multistage and cascade refrigeration machines. It is necessary to address the issue of choice as the scheme itself.

However, since the Vienna Convention for the Conservation of the Ozone Layer in 1985, the Montreal Protocol on Substances that Deplete the Earth's Ozone Layer

(1987), and including subsequent agreements in London, Copenhagen, Vienna and Montreal, a large Earth's stratosphere ozone conservation program, aimed primarily at creating refrigerants, alternatives to ozone-hazardous, new types of refrigeration equipment, polymers, aerosols, fire extinguishers, etc.

Therefore, in addition to energy efficiency, modern refrigeration machines are also subject to strict environmental safety requirements, which are assessed by indicators such as ODP and GWP. Currently, the use of natural refrigerants (ammonia, carbon dioxide) and refrigerants that do not contain chlorine atoms in their molecule is promising. Hydrocarbons, components of natural gas, have become especially popular nowadays. These are propane, butane, pentane and their isomers. Research work is devoted to these topical issues.

II. LITERATURE ANALYSIS

2.1. Circuit solutions for low-temperature refrigerators and evaluation of their efficiency

Obtaining low temperatures (below –25 °C) requires the use of special refrigeration machines based on two- and multi-stage, as well as cascade cycles. This is due to a significant reduction in the feed rate of the compressor and a significant increase in its size, rising temperature of the refrigerant and oil, which can cause the formation of a deposit on the inner surface of the compressor cylinder, as well as spontaneous combustion, self-decomposition of oil. Possible modes of operation when the refrigerant compression can get wet steam and hydraulic shock, to prevent this create significant overheating of the refrigerant on suction into the compressor, which leads to additional irreversible energy loss during its supply at low temperatures. Another negative circumstance of the low-temperature cycle using a single-stage refrigeration machine or circuit solutions with parallel throttling is to obtain a relatively high degree of dryness of steam after throttling to boiling pressure, which further requires more mass consumption of refrigerant in the first and subsequent stages. All these factors negatively affect the energy efficiency of the cycle, which is estimated by the coefficient of thermal transformation COP.

Energy-efficient circuit solutions must include full use of two- and multi-stage refrigeration circuits with full interstage refrigerant cooling, which is realized in intermediate vessels, maximum use of ambient cold, minimal overheating of the refrigerant on suction in the first stage of the compressor, creation of the maximum supercooling of the refrigerant after the condenser, use of serial throttling, etc. Most of these requirements are met by a two-stage refrigeration cycle with complete intermediate cooling and sequential throttling. However, the implementation of such a circuit solution is complicated by the problem of separating the oil from the intermediate vessel, which gets there after the first stage compressor and through the throttle valve to the evaporator, significantly reducing the refrigeration capacity of the refrigerator. The solution to this problem was the use of automatic control and management devices or other less energy efficient but more reliable circuit solution using parallel throttle and coil in the intermediate vessel, shown in Fig. 1 and 2.

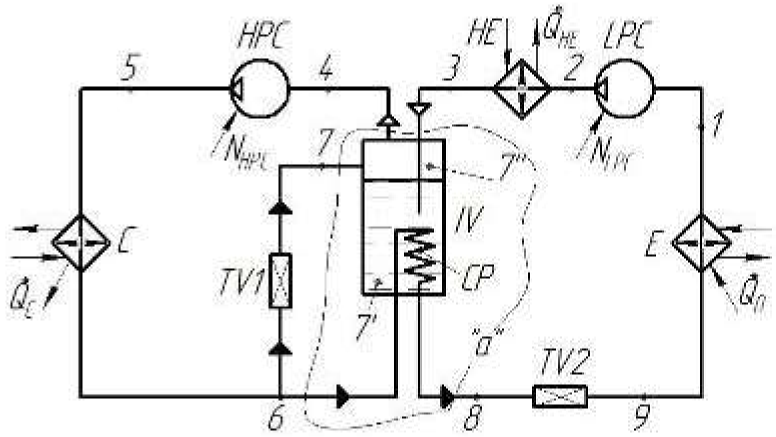
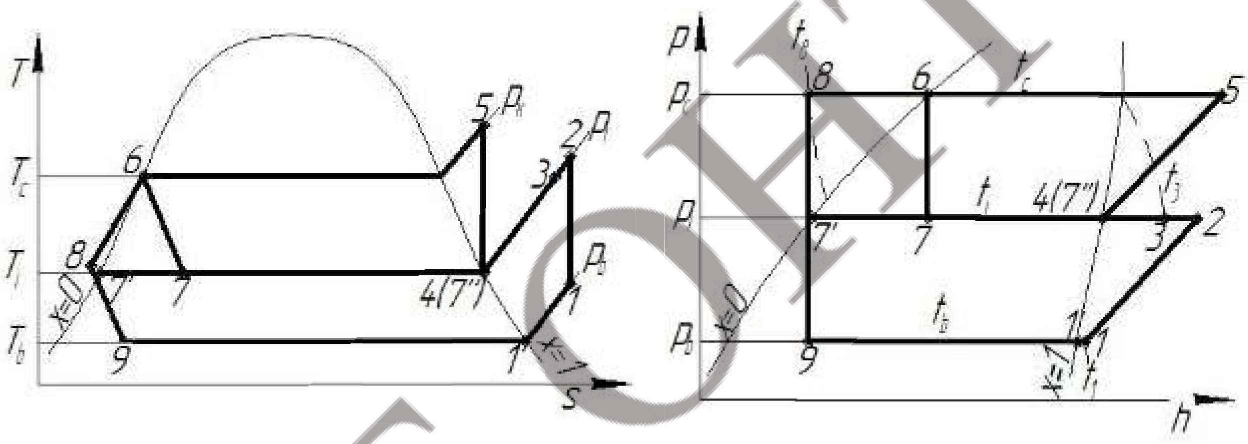


Fig. 1. Schematic diagram of a two-stage SCRM with a coil intermediate vessel: *LPC* – low pressure compressor (first stage); *HPC* – high pressure compressor (second stage); *HE* – heat exchanger; *IV* – intermediate vessel; *C* – condenser; *TV1, TV2* – throttle valves; *E* – evaporator; *CP* – coil pipe



a) *b)*

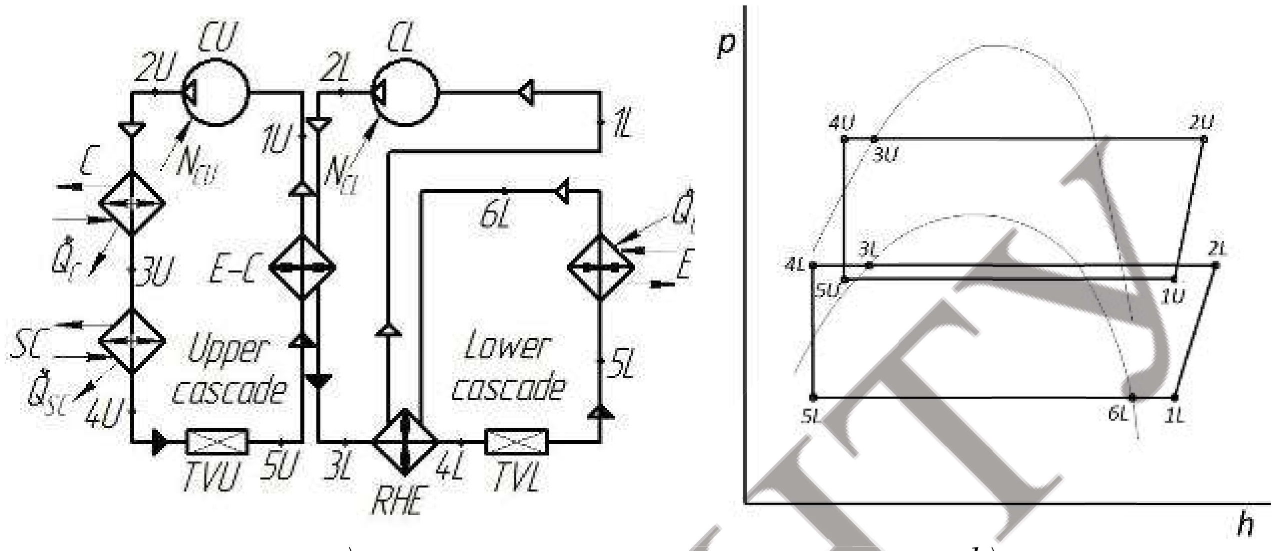
Fig. 2. Theoretical cycle of a two-stage SCRM with a coil intermediate vessel in *T*–*s* and *p*–*h* diagrams: p_b, T_b (t_b) – boiling pressure and temperature; p_c, T_c (t_c) – condensing pressure and temperature; p_i, T_i (t_i) – pressure and temperature in the intermediate vessel

Studies show that such a refrigeration machine has one of the best energy efficiency compared to other two-stage schemes and when ammonia is used as a refrigerant. This is ensured by the implementation of complete intermediate cooling of the refrigerant on the suction in the second stage of the compressor, as well as more complete supercooling of condensate in the intermediate vessel in combination with parallel throttling.

Another way to increase the energy efficiency of the low-temperature refrigeration cycle is the use of cascade circuits. The most widespread schemes consisting of two cascades: upper and lower. In fact, the cascade cycle combines two single-stage steam compression refrigeration machines. The lower and upper cascades interact through a special heat exchanger called an evaporator-condenser.

In the case of the use of CFCs in one of the cascades, a circuit solution with a regenerative heat exchanger is used, and in the case of the use of ammonia, a circuit with a subcooler is used.

Theoretical cycle of a cascade SCRM in *p–h* diagram of submissions on Fig. 3.



a) b)

F
i
g

When calculating the cycle of the lower cascade, we assume that it is formed by a more energy-efficient scheme SCRM with RHE. When calculating the cycle of the upper cascade, we assume that it is formed by the cycle SCRM with subcooler, because the refrigerant in it is ammonia (R717).

When using a cascade cycle, two problems arise, which are investigated in this work, namely: the choice of refrigerant for the lower and upper cascade, as well as the choice of intermediate phase transition temperature in the condenser-evaporator.

Comparison of energy efficiency of cycles was performed by calculating the coefficient of thermal transformation COP. This value was determined by the following dependences, expressed in terms of specific values:

– two-stage steam-compression refrigeration machine with a coil intermediate vessel

$$COP = \frac{\dot{Q}_0}{N_{LPC}+N_{HPC}}, \tag{1}$$

$$COP = \frac{q_0}{l_{LPC}+\frac{\dot{m}_{HPC}}{\dot{m}_{LPC}} \cdot l_{HPC}}, \tag{2}$$

$$COP = \frac{q_0}{l_{LPC}+\frac{h_3-h_8}{h_4-h_6} \cdot l_{HPC}}, \tag{2a}$$

where $\dot{Q}_0$ – refrigeration capacity of the cycle, W;

N_{HPC} – high pressure compressor power, W;

N_{LPC} – low pressure compressor power, W;

$q_0 = h_1 - h_9$ – specific mass refrigeration capacity, kJ/kg;

$l_{LPC} = h_2 - h_1$ – specific mass work of the low pressure compressor, kJ/kg;

$l_{HPC} = h_5 - h_4$ – specific mass work of the high pressure compressor, kJ/kg;

$\dot{m}_{HPC}$ – mass consumption of refrigerant in the high pressure compressor, kJ/kg;

$\dot{m}_{LPC}$ – mass consumption of refrigerant in the low pressure compressor, kJ/kg.

– cascade refrigeration machine

$$COP = \frac{\dot{Q}_0}{N_{CL} + N_{CU}}, \tag{3}$$

$$COP = \frac{q_{0L}}{l_{CL} + \frac{q_{cL}}{q_{0U}} \cdot l_{CU}}, \tag{4}$$

where N_{CL} – lower cascade compressor power, W;

N_{CU} – upper cascade compressor power, W;

$l_{CU} = h_{2U} - h_{1U}$ – specific mass work of the upper cascade compressor, kJ/kg;

$l_{CL} = h_{2L} - h_{1L}$ – specific mass work of the lower cascade compressor, kJ/kg;

$q_{0U} = h_{1U} - h_{5U}$ – specific mass refrigeration capacity of the upper cascade, kJ/kg;

$q_{0L} = h_{6L} - h_{5L}$ – specific mass refrigeration capacity of the lower cascade, kJ/kg;

$q_{cL} = h_{2L} - h_{3L}$ – specific mass heat load of the lower cascade condenser, kJ/kg. [2]

2.2. Refrigerators and their properties

In addition to energy efficiency, modern refrigeration machines are also subject to strict environmental safety requirements, which are assessed by indicators such as LCCP, ODP and GWP. Currently, the use of natural refrigerants (ammonia, carbon dioxide) and refrigerants that do not contain chlorine atoms in their molecule is promising. Hydrocarbons, components of natural gas, have become especially popular nowadays. These are propane, butane, pentane and their isomers. Research work is devoted to these topical issues.

In the work the refrigerants have basic indicators, including ecological, which are given in Tab. 1.

Tab. 1. Characteristics of refrigerants considered in [2–4]

№	Symbol	Chemical formula	Critical parameters		Normal boiling point t_N, °C	ODP	GWP	LCCP	Combustibility	Toxicity
			pressure p_{cr}, МПа	temperature t_{cr}, °C						
1	R717	NH_3	11,333	132.2	–33.3	0	0	0,25	+	+
2	R13	$CClF_3$	3,879	28.8	–81.5	1,0	14400	130	–	+
3	R23	CHF_3	4,832	26.1	–82.0	0,0004	14310	270	–	–
4	R290	C_3H_8	4,251	96.7	–42.1	0	20	0,041	+	–
5	R744	CO_2	7,377	31.0	–78.4	0	1	120	–	–
6	R32	CH_2F_2	5,782	78.1	–51.6	0	670	4,9	+	+

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. Among those listed in Tab. 1 refrigerants special attention is paid to natural refrigerants, namely: ammonia (R717), propane R290) and carbon dioxide (R744) as the most environmentally friendly.

Given the fact that the work involves the study of energy efficiency of an industrial design of a refrigeration machine, ie one with an average refrigeration capacity (at least 15 kW), as well as taking into account good environmental performance, based on refrigerant R717. In addition to environmental friendliness, ammonia has a number of advantages:

– low cost, which provides relatively low capital costs for the purchase and maintenance of refrigeration;

– high intensity of heat transfer in the devices, which provides a relatively small size;

– high density, which ensures the use of smaller diameters of pipelines;

– high specific heat of condensation (evaporation), which allows to ensure the lowest consumption of refrigerant in the cycle;

– low threshold of odor evaporation, which allows you to detect even the smallest leaks of refrigerant, even without the use of special equipment (current detectors).

The disadvantages of using ammonia as a refrigerant include its toxicity and flammability. However, with the observance of safety and labor protection measures, as well as the automation of the refrigeration machine, their harmful effects can be significantly reduced or even eliminated.

III. OBJECT, SUBJECT, AND METHODS OF RESEARCH

The aim of the research is to analyze the existing circuit solutions of low-temperature refrigeration machines and the impact of the properties of refrigerants on the environmental safety and energy efficiency of the refrigeration cycle.

To achieve this *goal*, the following tasks are formulated and solved:

– analysis of existing circuit solutions of low-temperature refrigeration machines;

– analysis of the impact of refrigerant properties on environmental safety and energy efficiency of the refrigeration cycle;

– development of recommendations for the calculation of the cascade refrigeration cycle.

The objects of research are circuit solutions of cycles of low-temperature refrigeration machines and properties of refrigerants used in such cycles.

The subject of research is the energy efficiency of low-temperature refrigeration machine and the impact on it of the schematic diagram and thermodynamic and environmental properties of the refrigerant.

In solving the tasks such *methods* were used as: analysis of scientific and technical information, mathematical modeling using specialized software products (Refprop, Coolpack).

IV. RESULTS

4.1. Initial parameters and assumptions

Investigate the influence of the circuit solution of low-temperature SCRM of medium refrigeration capacity and the choice of refrigerant on the energy efficiency of the cycle (coefficient of thermal transformation COP).

For all cases ask:

– boiling point $t_b = -45\ °C$;

– condensation temperature (upper circuit for cascade refrigeration machines) $t_c = 45\ °C$;

– theoretical cycle (relative internal efficiency of the compressor $\eta_{oi} = 1$; there are no pressure losses in the devices;

– heat loss to the environment from appliances and other elements of the refrigeration machine are absent).

For cascade refrigeration machine, the condensing temperature of the refrigerant of the lower circuit of the cascade refrigeration machine $t_{cL} = 0\ °C$.

Consider cycles and calculate the coefficient of thermal transformation COP :

a) two-stage SCRM with a coil intermediate vessel; refrigerant R717; overheating on the suction of the first degree $\Delta t_{0h} = 5\ °C$;

b) cascade RM: for all cases – the upper cascade of SCRM with subcooler; refrigerant R717; hypothermia in the subcooler $\Delta t_{cu} = 5\ °C$; overheating on suction $\Delta t_{0h} = 5\ °C$; underrecuperation in the condenser evaporator $\Delta t_{E-C}^{ur} = 3\ °C$

b.1) SCRM with regenerative heat exchanger (RHE); lower cascade: refrigerant R13; overheating on suction $\Delta t_{0h} = 10\ °C$;

b.2) SCRM with RHE; lower cascade: refrigerant R23; overheating on suction $\Delta t_{0h} = 10\ °C$;

b.3) SCRM with RHE; lower cascade: refrigerant R290; overheating on suction $\Delta t_{0h} = 10\ °C$;

b.4) SCRM with RHE; lower cascade: refrigerant R744; overheating on suction $\Delta t_{0h} = 10\ °C$;

b.5) SCRM with RHE; lower cascade: R32 refrigerant; overheating on suction $\Delta t_{0h} = 10\,°C$.

For a cascade refrigeration machine with a maximum COP, investigate the choice of the optimal condensation temperature of the lower circuit refrigerant t_{cL}.

4.2. Calculation results

a) Two-stage SCRM with a coil intermediate vessel

The calculation scheme and theoretical cycle are shown in Fig. 1 and 2.

The calculation was performed in accordance with the task (see more section 4.1, item a).

Summary results of calculations are given in Tab. 2.

Tab. 2. Summary calculation's results of two-stage SCRM with a coil intermediate vessel

Parameter	π_{st}	t_2	t_5	x_9	COP
Unit of measurement	–	$°C$	$°C$	–	–
Result	5.75	76.6	121	0.134	2.07

Symbols in Tab. 2:

$\pi_{st} = p_c / p_i = p_i / p_b$ – the degree of pressure increase in the compressor stage;

t_2, t_5 – the temperature of the refrigerant after theoretical compression in the first and second stages of the compressor;

x_9 – the degree of dryness of the refrigerant before the evaporator;

COP – cycle thermal transformation coefficient.

b) Cascade refrigeration machine

The calculation scheme and theoretical cycle are shown in Fig. 3.

The calculation was performed in accordance with the task (see more section 4.1, item b).

Summary results of calculations in accordance with the item b.1 are given in Tab. 3.

Tab. 3. Summary calculation's results of cascade refrigeration machine (lower cascade – R13, upper cascade – R717)

Parameter	π_{st}^{U}	π_{st}^{L}	t_{2U}	t_{2L}	x_{5U}	x_{5L}	COP
Unit of measurement	–	–	$°C$	$°C$	–	–	–
Result	4.653	3.895	117	22.6	0.160	0.328	1.80

Symbols in Tab. 3:

$\pi_{st}^{U} = p_{cU} / p_{bU}$, $\pi_{st}^{L} = p_{cL} / p_{bL}$ – the degree of pressure increase in the upper and lower cascades;

t_{2U}, t_{2L} – the temperature of the refrigerant after theoretical compression in the upper and lower cascades;

x_{5U}, x_{5L} – the degree of dryness of the refrigerant before the evaporator in the upper and lower cascades;

COP – cycle thermal transformation coefficient.

Summary results of calculations in accordance with the item b.2 are given in Tab. 4.

Tab. 4. Summary calculation's results of cascade refrigeration machine (lower cascade – R23, upper cascade – R717)

Parameter	π_{st}^{U}	π_{st}^{L}	t_{2U}	t_{2L}	x_{5U}	x_{5L}	COP
Unit of measurement	–	–	°C	°C	–	–	–
Result	4.653	4.272	117	41	0.160	0.288	1.80

Summary results of calculations in accordance with the item b.3 are given in Tab. 5.

Tab. 5. Summary calculation's results of cascade refrigeration machine (lower cascade – R290, upper cascade – R717)

Parameter	π_{st}^{U}	π_{st}^{L}	t_{2U}	t_{2L}	x_{5U}	x_{5L}	COP
Unit of measurement	–	–	°C	°C	–	–	–
Result	4.653	5.326	117	23.8	0.160	0.290	2.00

Summary results of calculations in accordance with the item b.4 are given in Tab. 6.

Tab. 6. Summary calculation's results of cascade refrigeration machine (lower cascade – R744, upper cascade – R717)

Parameter	π_{st}^{U}	π_{st}^{L}	t_{2U}	t_{2L}	x_{5U}	x_{5L}	COP
Unit of measurement	–	–	°C	°C	–	–	–
Result	4.653	4.194	117	63.4	0.160	0.260	1.79

Summary results of calculations in accordance with the item b.5 are given in Tab. 7.

Tab. 7. Summary calculation's results of cascade refrigeration machine (lower cascade – R32, upper cascade – R717)

Parameter	π_{st}^{U}	π_{st}^{L}	t_{2U}	t_{2L}	x_{5U}	x_{5L}	COP
Unit of measurement	–	–	°C	°C	–	–	–
Result	4.653	5.807	117	66	0.160	0.176	1.94

Symbols in Tabs. 4–7 are the same as Tab. 3.

Therefore, the results of the calculations include the following conclusion: the best value of the coefficient of thermal transformation was obtained for the ratio of refrigerants upper stage R717 lower R290, which was 2.00. However, the value obtained is lower than the coefficient of thermal transformation obtained in the calculation of a two-stage refrigeration machine with a coil intermediate vessel, which was 2.07. In order to improve the COP for the cascade cycle, we investigate the effect of intermediate temperature (condensing temperature of the refrigerant of the lower circuit on the value of this parameter).

c) Modified cascade refrigeration cycle

In this work, calculations of a modified cascade cycle consisting of a combination of a two-stage refrigeration machine with full intermediate cooling, parallel throttling and a coil in an intermediate vessel in the upper cascade and a single-stage steam-compression refrigeration machine with regenerative heat exchanger in the lower cascade are performed. Ammonia is used as the refrigerant of the upper stage, and for the lower stage we use the refrigerant that received the highest CPC among the investigated refrigerants (R13, R23, R290, R744, R32) for the normal cascade cycle (R290).

This circuit solution has not been described in the literature. However, its implementation can theoretically have a better indicator of energy efficiency.

In Fig. 4 and 5 show the schematic diagram of the combined cascade cycle and its cycle in the *p–h* diagram.

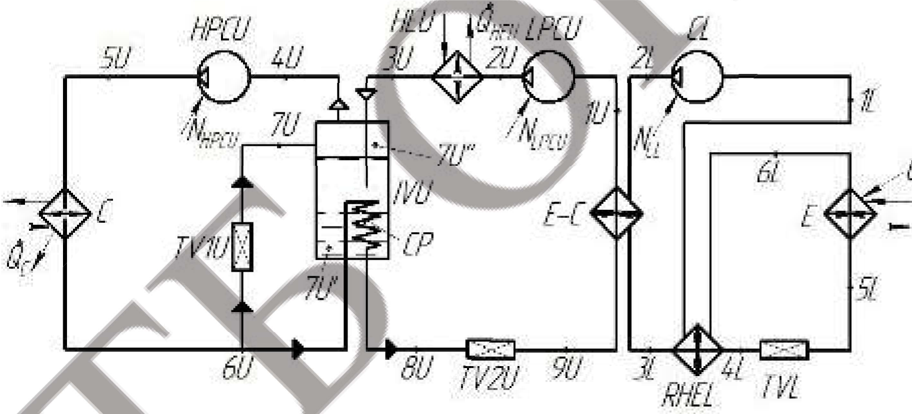

Figure 4 – Schematic diagram of the modified cascade cycle:
the upper cascade is a two-stage SCRM with a coil intermediate vessel, the lower cascade is a single-stage SCRM with RHE

Symbols for Figure 4: LPCU – low pressure compressor (first stage of the upper cascade) of the upper stage; *HPCU* – high pressure compressor (second stage of the upper cascade) of the upper stage; *HEU* – heat exchanger of the upper cascade; *IVU* – intermediate vessel of the upper cascade; *C* – condenser; *TV1U*, *TV2U* – throttle valves of the upper cascade; *CP* – coil pipe; *TVL* – throttle valve of the lower cascade; *RHEL* – regenerative heat exchanger of the lower cascade; *CL* – compressor of the lower cascade; *E* – evaporator; *E-C* – evaporator-condenser

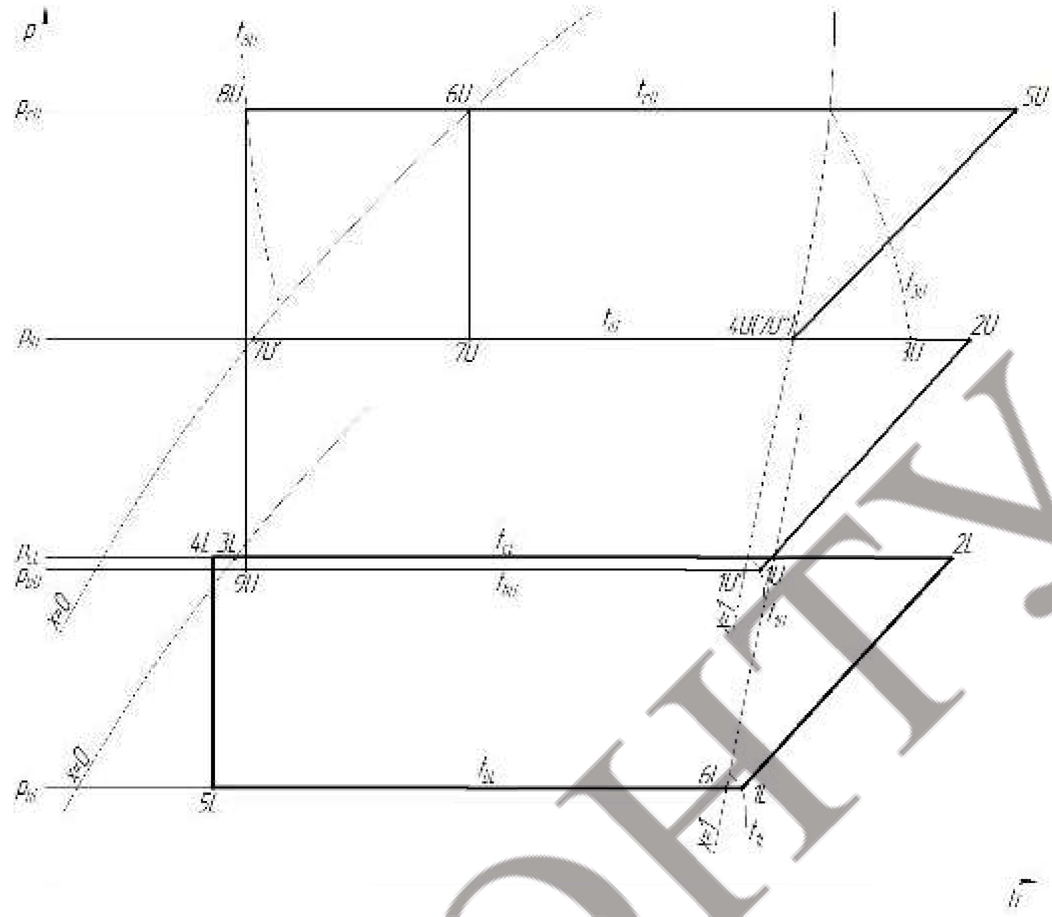

Figure 5 – Theoretically modified cascade cycle in *p–h* diagram:
p_{bU}, t_{bU} – boiling pressure and temperature of the upper cascade; p_{bL}, t_{bL} – boiling pressure and temperature of the lower cascade; p_{cU}, t_{cU} – condensing pressure and temperature of the upper cascade; p_{cL}, t_{cL} – condensing pressure and temperature of the lower cascade; p_{iU}, t_{iU} – pressure and temperature in the intermediate vessel of the upper cascade

The energy efficiency of the modified cascade cycle was determined by full or specific values in terms of:

$$COP = \frac{\dot{Q}_0}{N_{LPCU}+N_{HPCU}+N_{CL}}, \tag{5}$$

$$COP = \frac{q_{0L}}{\frac{q_{cL}}{q_{0U}}\cdot\left(l_{LPCU}+\frac{h_{3U}-h_{8U}}{h_{4U}-h_{6U}}\cdot l_{HPCU}\right)+l_{CL}}, \tag{6}$$

where $\dot{Q}_0$ – refrigeration capacity of the cycle, W;

N_{CL} – lower cascade compressor power, W;

N_{LPCU} – low pressure compressor power of the upper cascade, W;

N_{HPCU} – high pressure compressor power of the upper cascade, W;

$l_{LPCU} = h_{2U} - h_{1U}$ – specific mass work low pressure compressor of the upper cascade, kJ/kg;

$l_{HPCU} = h_{5U} - h_{4U}$ – specific mass work high pressure compressor of the upper cascade, kJ/kg;
$l_{CL} = h_{2L} - h_{1L}$ – specific mass work of the lower cascade compressor, kJ/kg;
$q_{0U} = h_{1U} - h_{5U}$ – specific mass refrigeration capacity of the upper cascade, kJ/kg;
$q_{0L} = h_{6L} - h_{5L}$ – specific mass refrigeration capacity of the lower cascade, kJ/kg;
$q_{cL} = h_{2L} - h_{3L}$ – specific mass heat load of the lower cascade condenser, kJ/kg.

Summary results of calculations are given in Tab. 8.

Tab. 8. Summary calculation's results of modified cascade cycle (lower cascade – R290, upper cascade – R717)

Parameter	π^{U}_{st}	π^{L}_{st}	t_{2U}	t_{5U}	t_{2L}	x_{5U}	x_{5L}	COP
Unit of measurement	–	–	$°C$	$°C$	$°C$	–	–	–
Result	2.157	5.326	55	75	24	0.10	0.213	2.07

As can be seen from the calculation results (see Tab. 8), the use of a two-stage cycle in the upper cascade did not increase the energy efficiency of the refrigeration cycle as a whole and can not be absolutely recommended for implementation due to significant complication. On the other hand, the obvious advantages of this cycle include a significant reduction in refrigerant temperatures after compression and degrees of pressure increase in each stage of the upper cascade compressor. This will increase the feed rate of the compressor, reduce its weight and size and increase its service life.

The generalized results of the coefficients of thermal transformation according to comparative schemes are summarized in Tab. 9.

Tab. 9. Comparison of COP coefficients for different circuit solutions of refrigeration machines; condensation temperature 45 °C, boiling point –45 °C

№ cf.	Schematic solution	COP	Note
1	2	3	4
1	One-stage SCRM with subcooler*	1,71	refrigerant R717
2	Two-stage SCRM with a coil intermediate vessel	2,07	refrigerant R717
3	Cascade refrigeration machine	2,00	refrigerant of the upper cascade R717; refrigerant of the lower cascade R290
4	Modified cascade refrigeration machine	2,07	refrigerant of the upper cascade R717; refrigerant of the lower cascade R290

* calculations were not performed in this work

d) Investigation of the influence of the condensation temperature of the lower curcuit refrigerant on the energy efficiency of the cascade cycle

For a cascade refrigeration machine with a maximum COP, we investigate the choice of the optimal condensation temperature of the refrigerant of the lower cascade. To do this, set the values $t_{cL} = -20; -10; 10; 20\ °C$. The refrigerant of the lower cascade is R290, the refrigerant of the upper cascade is R717.

Thermal calculations of the cycles of the upper and lower cascades are performed accordence with the scheme shown in Fig. 3. The coefficient of tremotransformation COP is determined by formula (4).

The results of the calculations are summarized in Tab. 10.

Tab. 10 The influence of the condensation temperature of the upper circuit on the coefficient of thermal transformation of the cascade cycle

t_{cL}, °C	-20	-10	0	10	20
COP	1,94	2,01	2,00	2,00	1,91

As can be seen from the calculation results, the optimal condensing temperature range of the lower stage is $-10 \le t_{cL} \le 10\ °C$. Taking into account the condensation temperatures of the upper cascade $t_{к} = 45\ °C$ and boiling of the lower cascade $t_0 = -45\ °C$ it can be concluded that when calculating the cascade refrigeration machine using ammonia in the upper cascade, and in the lower propane, you can set the condensation temperature of the lower cascade as the arithmetic mean

$$t_{cL} = 0{,}5\left(t_{bL} + t_{cU}\right). \tag{7}$$

This will maintain the maximum energy efficiency of the cascade cycle.

V. CONCLUSIONS

Ways to increase the energy efficiency of low-temperature refrigeration machine are considered in the research work. Two basic approaches were considered: the choice of refrigerant or their combinations and the choice of circuit design.

As shown by the results of variable calculation of different circuit solutions of refrigeration machines, given in Tab. 9, the highest value of the cycle thermal transformation coefficient (COP = 2.07) was obtained in two cases: when using a two-stage SCRM with a coil intermediate vessel and a modified cascade refrigeration machine – this is a new circuit solution not described in classic textbooks on refrigeration machines. In addition, comparing the intermediate operating parameters of these circuit solutions, it is recommended to prefer a modified cascade refrigeration machine to provide better operating conditions for its compressors, namely, maintaining lower cascades of pressure increase and final refrigerant temperatures after compression.

Studies of the choice of refrigerant of the lower cascade from the standpoint of energy saving have shown that the largest COP will occur in the following combination of refrigerants: refrigerant of the upper cascade R717, refrigerant of the lower cascade R290. These refrigerants are environmentally friendly, which meets the conditions of the task. Refrigerants previously used traditionally in the lower cascade, such as R13 and R23, were less effective. R744, which is currently often recommended for use in cascade cycles, has shown insufficient efficacy (for details, see Tab. 3–7).

The paper also shows (for details, see Tab. 10) that when selecting the intermediate temperature of the phase transition of the cascade cycle, you can set the condensation temperature of the lower cascade, using the formula (7).

VI. REFERENCES

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