https://doi.org/10.15407/scine22.01.094

KOZIN, V. M. [1] (https://orcid.org/0000-0001-9821-7774),
SHARAPOV, S. O. [2] (https://orcid.org/0000-0002-8433-8580),
and BARSUKOVA, H. V. [1] (https://orcid.org/0000-0002-4261-2182)

[1] Sumy National Agrarian University,
160, Herasyma Kondratieva St., Sumy, 40021, Ukraine,
+380 542 70 1158, admin@snau.edu.ua
[2] Sumy State University,
59, Petropavlivska St., Sumy, 40014, Ukraine,
+380 542 66 5124, info@sumdu.edu.ua

INFLUENCE OF A LOW-TEMPERATURE REFRIGERATOR SCHEME SOLUTION BASED ON ENVIRONMENTALLY FRIENDLY REFRIGERANTS ON THE ENERGY EFFICIENCY OF THE CYCLE

Introduction. *Artificial cold at temperatures of –40 °C and below is widely used in various electrotechnological processes. At the same time, the choice of refrigerant and the configuration of the refrigeration cycle determine not only energy efficiency, but also the environmental safety of the system.*

Problem Statement. *For deep-cold conditions, multi-stage or cascade circuits are typically applied, with a rational selection of refrigerant pairs to ensure high efficiency.*

Purpose. *To assess the influence of the circuit design — given the environmental characteristics of refrigerants — on the energy efficiency of a low-temperature refrigeration cycle using the coefficient of performance (COP) and exergy efficiency (ε), and to provide practical recommendations for selecting both the circuit type and the refrigerants.*

Materials and Methods. *A thermodynamic and exergetic comparative analysis of theoretical low-temperature refrigeration cycles has been performed. Numerical modeling has been carried out in REFPROP for specified temperatures (t_c = 45 °C; t_b = –45 °C). The analyzed schemes include: a two-stage vapor-compression refrigeration machine (VCRM) with an intermediate tank and coil; cascade systems for the refrigerant pairs R717/R13, R717/R23, R717/R290, R717/R744, and R717/R32; a modified cascade cycle; and a three-stage VCRM.*

Results. *The highest energy efficiency (COP = 2.22; ε = 0.372) has been achieved using a three-stage R717 VCRM. Among conventional cascade refrigeration cycles, R717/R290 has been identified as the most*

Citation: Kozin, V. M., Sharapov, S. O., and Barsukova, H. V. (2026). Influence of a Low-Temperature Refrigerator Scheme Solution Based on Environmentally Friendly Refrigerants on the Energy Efficiency of the Cycle. *Sci. innov.*, 22(1), 94—106. https://doi.org/10.15407/scine22.01.094

energy-efficient (COP ≈ 2.00) and environmentally acceptable pair. The modified cascade cycle has demonstrated higher efficiency (COP ≈ 2.07) than conventional cascade schemes, and its implementation has enabled a notable reduction in thermomechanical parameters, positively affecting compressor service life.

Conclusions. *For low-temperature applications, multi-stage R717 VCRM systems should be preferred, as they have proven more energy-efficient than cascade circuits. When installation reliability is of primary importance, a modified cascade cycle with R717/R290 is advisable. For such a cycle, a rule for selecting the intermediate temperature has been proposed.*

Keywords: energy efficiency, exergy efficiency, energy saving, electrical technology, environmental friendliness, refrigeration machine, cascade refrigeration cycle, multi-stage refrigeration cycle.

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 characterised by the widest use of cold in all sectors of the economy. In fact, there are no industries where cold is not used [1]. Many technological processes require the use of low temperatures (no higher than –40 °C), which can be created and maintained with the help of low-temperature refrigeration machines as a type of electrical process. At the same time, they must have high energy efficiency, which is not achieved with conventional single-stage vapour compression refrigeration machines (VCRM). Therefore, it is necessary to apply other scheme solutions where multi-stage and cascade refrigeration machines are used.

However, starting with the Vienna Convention for the Preservation 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, the programme for the preservation of ozone in the Earth's stratosphere was launched, aimed primarily at creating refrigerants alternative to ozone-depleting ones, new types of refrigeration equipment, polymers, aerosols, etc.

Therefore, in addition to energy efficiency, modern refrigeration machines also meet stringent environmental safety requirements, which are assessed by such indicators as LCCP, ODP and GWP. At present, the use of natural refrigerants (ammonia, carbon dioxide) and refrigerants that do not contain chlorine atoms in their molecules is promising. Hydrocarbons, which are part of natural gas, have become particularly popular. These include propane, butane, pentane and their isomers, etc.

SCHEMATIC SOLUTIONS FOR LOW-TEMPERATURE REFRIGERATORS AND EVALUATION OF THEIR EFFICIENCY

Achieving low temperatures (below –25 °C) requires the use of special refrigeration machines based on two- and multi-stage, as well as cascade refrigeration cycles [2—4]. This is due to a significant reduction in the compressor's delivery ratio and a significant increase in its size, an increase in the temperature of the refrigerant and lubricant, which can cause the formation of carbon deposits on the inner surface of the compressor cylinder, as well as spontaneous combustion and self-decay of the refrigeration oil. There may be operating modes when moisture can form in the refrigerant during compression, which can lead to a hydraulic shock. To prevent this, a significant overheating of the refrigerant is created at the compressor suction, which leads to additional irreversible energy losses when it is supplied at low temperatures. Another negative circumstance of a low-temperature cycle using a single-stage refrigeration machine or parallel throttling schemes is the relatively high degree of dryness of the vapour after throttling to the boiling point. This additionally requires a higher mass flow rate of refrigerant, both in the first stage and in the subsequent stages. All of these factors have a negative impact on the energy efficiency of the cycle, which is usually assessed by the COP, or thermo-economic analysis methods can be used to determine the exergy efficiency of the cycle.

Energy-efficient design solutions should include full use of two- and multi-stage refrigeration schemes with full interstage refrigerant cooling in intermediate vessels, maximum use of ambient cold, minimum overheating of the refrigerant in the suction at the first stage of the compressor, maximum subcooling of the refrigerant after the condenser, use of sequential throttling, etc. Most of these requirements are met by a two-stage cooling cycle with full intermediate cooling and sequential throttling. However, the implementation of such a scheme solution is complicated by the problem of separating and removing refrigeration oil from the intermediate vessel, which enters it after the first stage compressor and through the throttle valve to the evaporator, significantly reducing the refrigeration capacity of the refrigeration machine. The solution to this problem was to use automatic monitoring and control devices or another less energy-efficient but more reliable scheme solution using parallel throttling and a coil in the intermediate vessel. The schematic diagram and theoretical cycle of a two-stage VCRM with an intermediate vessel and coil can be found in [5].

Studies have shown that this chiller has one of the best energy efficiency ratings compared to other two-stage schemes, even when using ammonia as a refrigerant. This is due to the complete intermediate cooling of the refrigerant at the suction side of the second stage of the compressor, as well as the more complete subcooling of the condensate in the intermediate vessel in combination with parallel throttling.

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

When CFCs are used in one of the cascades, a regenerative heat exchanger is used, and when ammonia is used, a supercooler is used.

The schematic diagram and theoretical cycle of a cascade refrigeration machine can be found in [5].

For the calculation of the lower stage cycle, we assume that it is formed by the more energy-efficient VCRM scheme with a regenerative heat exchanger. When calculating the upper stage cycle, we assume that it is formed by a VCRM cycle with a condensate supercooler, since the refrigerant used is ammonia (R717).

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

We compare the energy efficiency of cycles by determining the coefficient of thermal transformation COP. This value is determined regardless of the type of cycle according to a well-known relationship:

$$\mathrm{COP} = \frac{\dot{Q}_0}{\sum N_c}, \tag{1}$$

where $\dot{Q}_0$ is the cooling capacity of the refrigeration cycle; $\sum N_c$ is the total capacity of the compressors that ensure the cycle.

The schematic solution of a three-stage VCRM is also known. The use of an additional third stage compared to a two-stage scheme allows reducing the degree of pressure increase in one stage, which has a positive effect on the compressor's flow rate, its service life, final temperatures of the refrigerant after compression in the compressor, etc. The schematic diagram and theoretical cycle of a three-stage VCRM can be found in [5]. Compared to the scheme of a two-stage VCRM with an intermediate vessel and a coil, this scheme also allows for complete interstage cooling of the refrigerant, and, provided that the mechanism for removing lubricant and refrigerant from intermediate vessels is properly implemented, eliminates additional energy losses due to under-recovery of the flow in the coil.

REFRIGERANTS AND THEIR PROPERTIES

In addition to energy efficiency, modern refrigeration machines also meet stringent environmental safety requirements for the refrigerants used in them. They are assessed by such indicators as LCCP, ODP and GWP. At present, the use of natural refrigerants (ammonia, carbon dioxide) and refrigerants that do not contain chlorine atoms in their molecules is promising. Hydrocarbons, which are part of natural gas, have become particularly popular. These include propane, butane, pentane and their isomers, etc.

The studied refrigerants, which can be used in low-temperature refrigeration machines, have the following main indicators, including environmental ones, which are given in Table 1.

As can be seen from Table 1, all refrigerants except R13 have little or no impact on the ozone layer. R13 is included only for the purpose of comparing energy efficiency as a refrigerant that was previously widely used in the lower stages of refrigeration machines.

Modern approaches of scientists [2—4] to the choice of refrigerant are to use and further analyse the energy efficiency of refrigeration cycles based on natural substances that have the lowest LCCP, ODP and GWP values. Among the refrigerants listed in Table 1, special attention is paid to natural refrigerants, namely ammonia (R717), propane (R290) and carbon dioxide (R744) as the most environmentally friendly. However, among the refrigerants presented in Table 1, the main attention should be paid to ammonia (R717), as this study involves analysing the energy efficiency of an industrial model of a refrigerator with an average cooling capacity (at least 15 kW). Ammonia also has a number of good environmental characteristics and benefits:

- low cost, which ensures relatively low capital expenditure on the purchase and maintenance of refrigeration equipment;
- high intensity of heat exchange in the devices, which ensures relatively small dimensions;
- high density, which enables the use of smaller pipelines;
- high specific heat of condensation (evaporation), which ensures the lowest refrigerant consumption in the cycle;
- low evaporation threshold, which allows even the smallest refrigerant leaks to be detected even without the use of special equipment.

The disadvantages of using ammonia as a refrigerant include its toxicity and flammability. However, with proper health and safety precautions and the automation of the chiller, their harmful effects can be significantly reduced or even eliminated.

As we can see, the advantages of using R717 as a refrigerant are much greater than the disadvantages. Nevertheless, analysing numerous modern studies [2—4, 8—11], it is almost impossible to find this substance in at least one of the refrigeration schemes of cascade refrigeration cycles. That is why the authors decided to take R717 as a basis and select the best possible vapour for it in terms of energy efficiency and environmental friendliness.

Table 1. **Characteristics of Refrigerants [5—7]**

No.	Refrigerant	Chemical formula	Critical parameters		Normal boiling point t_b, °C	ODP	GWP	LCCP	Flammability	Toxicity
			pressure p_{cr}, Pa	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	4.240	96.7	−44.0	0	3	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	+	+

The aim of the study is to analyse existing schematic solutions for low-temperature refrigeration machines and the impact of refrigerant properties on environmental safety and energy efficiency of the refrigeration cycle.

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

- analysis of existing scheme solutions for low-temperature refrigeration machines;
- analysis of the impact of refrigerant properties on environmental safety and energy efficiency of the refrigeration cycle;
- developing recommendations for calculating the cascade refrigeration cycle.

The objects of research are the scheme solutions of low-temperature refrigeration machine cycles and the properties of refrigerants used in such cycles.

The subject of the study is the energy efficiency of a low-temperature refrigeration machine and the influence of its scheme diagram and the thermodynamic and environmental properties of the refrigerant.

The following methods were used to solve the tasks: analysis of scientific and technical information, numerical modelling using specialised software products (Refprop [6], Coolpack [12]).

INPUT PARAMETERS AND ASSUMPTIONS

To investigate the influence of the scheme solution of a low-temperature VCRM with medium cooling capacity and the choice of refrigerant on the energy efficiency of the cycle (coefficient of thermal transformation COP).

For all cases, the following conditions are set:

- boiling point $t_b = -45$ °C;
- condensing temperature (upper cascade for cascade chillers) $t_c = 45$ °C;
- theoretical cycle: adiabatic compressor efficiency $\eta_s = 1$, no pressure losses in the refrigeration machine elements;
- there are no heat losses to the environment from the devices and other elements of the refrigeration machine.

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

Consider the cycles and calculate the COP according to the procedure below:

a) two-stage VCRM with an intermediate vessel and coil; R717 refrigerant; overheating at the suction of the first stage is $\Delta t_{0h} = 5$ °C;

b) cascade refrigeration machine: for all cases — upper stage VCRM with condensate intercooler; R717 refrigerant; of the refrigerant in the intercooler $\Delta t_{cu} = 5$ °C; overheating during suction $\Delta t_{0h} = 5$ °C; under-recovery in the evaporator-condenser $\Delta t_{E-C}^{ur} = 3$ °C;

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

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

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

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

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

c) three-stage VCRM; R717 refrigerant; overheating on the first stage suction $\Delta t_{0h} = 5$ °C.

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

RESULTS OF CALCULATIONS

a) Two-stage VCRM with intermediate vessel and coil. A summary of the calculations is presented in Table 2.

b) Cascade refrigeration machine. The summary of the results of the calculations for items b.1—b. 5 is presented in Table 3.

Thus, based on the results of the calculations (see Table 3), the following conclusion can be made: the best value of the thermal transformation coefficient was obtained for the ratio of refrigerants: upper stage — R717, lower stage — R290, which was 2.00. Carbon dioxide did not show sufficient efficiency as a lower-cycle refrigerant, although the R717/R744 pair is the most well-known and studied and, according to [13—16], can be used in cascade refrigeration machines of high cooling capacity at temperatures in the cooled objects not exceeding –30 °C. In [17, 18], which deals with cascade cycles of low-temperature refrigeration machines, R744 is recommended to be used in the upper cascade with a transcritical cycle. This will indeed ensure environmental safety, in contrast to the use of R23 in the lower cascade recommended by the same authors, which can cause global warming (see Table 1). It is also difficult to agree with the authors' opinion that the result will be a refrigeration machine with minimal weight and dimensions, because at least the ultra-high pressures of R744 in the upper stage will require the use of thick-walled massive pipes, the isobaric process of heat removal in the R744 cooler will lead to a decrease in the temperature head and heat transfer coefficient of this apparatus, and therefore to an increase in its surface area, and, accordingly, in its weight and dimensions.

Nevertheless, the value obtained is lower than the thermal transformation coefficient obtained in the calculation of a two-stage refrigeration machine with an intermediate vessel and coil, which was 2.07. Therefore, in order to identify possible ways to increase the COP of the cascade cycle, the effect of the intermediate temperature of the cycle will be investigated.

c) Modified cascade refrigeration cycle. Calculations were performed for a modified cascade cycle consisting of a combination of a two-stage refrigeration machine with full intermediate cooling, parallel throttling and a coil in the intermediate vessel for the upper stage and a single-stage vapour compression refrigeration machine with a regenerative heat exchanger in the lower stage. The upper stage refrigerant is ammonia, and the lower stage refrigerant is the refrigerant that has the highest COP among the refrigerants studied (R13, R23,

Table 2. **Summary Results of Calculation of a Two-Stage VCRM with an Intermediate Vessel and a Coil**

Parameter	π_{st}	t_2	t_5	x_9	COP
Unit measurement	—	°C	°C	—	—
Value	5.75	76.6	121	0.134	2.07

Note: $\pi_{st} = p_c / p_i = p_i / p_b$ is the degree of pressure increase in the compressor stage; t_2, t_5 are refrigerant temperature after theoretical compression in the first and second stages of the compressor, respectively; x_9 is the degree of dryness of the refrigerant before the evaporator

Table 3. **Summary Results of the Calculation of a Cascade Refrigeration Machine**

Item of the VCRM scheme	π_{st}^U	π_{st}^L	t_{2U}, °C	t_{2L}, °C	x_{5U}	x_{5L}	COP
b.1	4.653	3.895	117	22.6	0.160	0.328	1.80
b.2	4.653	4.272	117	41.0	0.160	0.288	1.80
b.3	4.653	5.326	117	23.8	0.160	0.290	2.00
b.4	4.653	4.194	117	63.4	0.160	0.260	1.79
b.5	4.653	5.807	117	66.0	0.160	0.176	1.94

Note: $\pi_{st}^U = p_{cU} / p_{bU}$, $\pi_{st}^L = p_{cL} / p_{bL}$ are the degree of pressure increase in the upper and lower cascades, respectively; t_{2U}, t_{2L} are refrigerant temperature after theoretical compression in the upper and lower stages, respectively; x_{5U}, x_{5L} are the degree of dryness of the refrigerant before the evaporator in the upper and lower stages, respectively

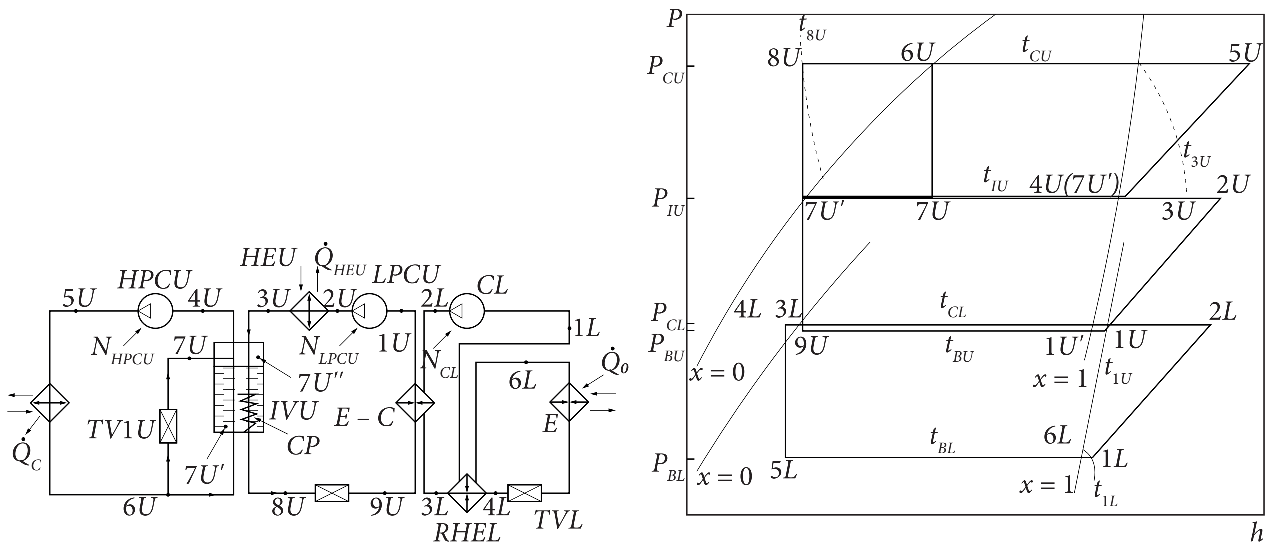

Fig. 1. Schematic diagram (*a*) and theoretical cycle (*b*) in the p-h diagram of the modified cascade cycle: upper stage — two-stage VCRM with an intermediate vessel and coil, lower stage — single-stage VCRM with RHE: *LPCU* — low pressure compressor (first stage of the upper stage) of the upper stage; *HPCU* — high pressure compressor (second stage of the upper stage) of the upper stage; *HEU* — heat exchanger of the upper stage; *IVU* — intermediate vessel of the upper stage; *C* — condenser; *TV1U, TV2U* — throttle valves of the upper stage; *CP* — coil; *TVL* — throttle valve of lower stage; *RHEL* — regenerative heat exchanger of the lower stage; *CL* — compressor of the lower stage; *E* — evaporator; *E – C* — evaporator-condenser

R290, R744, R32) for a conventional cascade cycle, namely propane R290.

Figure 1 shows the schematic diagram of the modified cascade refrigeration machine and its cycle in p-h diagram.

The thermal transformation coefficient of the modified cascade cycle is determined by the formula:

$$\mathrm{COP} = \frac{q_{0L}}{\dfrac{q_{cL}}{q_{0U}} \cdot \left(l_{LPCU} + \dfrac{h_{3U} - h_{8U}}{h_{4U} - h_{6U}} \cdot l_{LPCU} \right) + l_{CL}}, \quad (2)$$

where $l_{LPCU} = h_{2U} - h_{1U}$ is the specific work of the first stage compressor of the upper stage, kJ/kg; $l_{HPCU} = h_{5U} - h_{4U}$ is the specific work of the second stage compressor of the upper stage, kJ/kg; $l_{CL} = h_{2L} - h_{1L}$ is the specific work of the downstream compressor, kJ/kg; $q_{0U} = h_{1U} - h_{5U}$ is the specific cooling capacity of the upper stage, kJ/kg; $q_{0L} = h_{6L} - h_{5L}$ is the specific cooling capacity of the lower stage, kJ/kg; $q_{cL} = h_{2L} - h_{3L}$ is the specific heat load of the lower stage condenser, kJ/kg.

The summary results of the calculations are shown in Table 4.

As can be seen from the calculation results (see Table 4), the use of a two-stage cycle in the upper stage did not lead to an increase in the energy efficiency of the refrigeration cycle as a whole and cannot be unequivocally recommended for implementation due to its significant complexity. On the other hand, the obvious advantages of this cycle include a significant reduction in refrigerant temperatures after compression and pressure boost stages in each stage of the upper stage compressor. This increases the compressor's capacity factor, reduces its weight and size, and extends its service life.

Three-stage VCRM. The coefficient of thermal transformation of the three-stage cycle of VCRM is determined by expression (1). The summary results of the calculations are given in Table 5.

As can be seen from the calculation results (see Table 9), the use of a three-stage VCRM allows increasing the COP by 7.2 % compared to a two-stage refrigeration machine or a modified cascade refrigeration machine. However, after compression at the third stage, the refrigerant temperature rises to 92 °C compared to 75 °C in the modified cascade cycle, which increases the requirements for the properties of refrigeration oil and also leads to an increase in irreversible losses due to non-isothermal heat removal in the condenser.

d) Investigation of the influence of the lower stage refrigerant condensation temperature on the energy efficiency of the cascade cycle. For a cascade refrigeration machine with maximum COP, we investigate the choice of the optimal condensation temperature of the lower stage refrigerant. To perform the numerical study, we set the values of the condensation temperatures of the lower stage refrigerant $t_{cL} = -20; -10; 10; 20$ °C. The lower stage refrigerant is R290, the upper stage refrigerant is R717.

The results of the calculations are summarised in Table 6.

As can be seen from the calculation results (see Table 6), the optimal range of condensation temperatures of the lower stage is $-10 \le t_{cL} \le 10$ °C. Given the condensation temperature of the upper stage $t_{cU} = 45$ °C and the boiling point of the lower stage $t_{bL} = -45$ °C, it can be concluded that when calculating a cascade refrigeration machine using ammonia in the upper stage and propane in the lower stage, the condensation temperature of the lower stage can be set as an arithmetic mean

$$t_{cL} = 0.5(t_{bL} + t_{cU}). \quad (3)$$

This will help to maintain maximum energy efficiency in the cascade cycle.

e) Thermo-economic analysis of refrigeration cycles. As a rule, to assess the energy efficiency of the refrigeration cycle, the coefficient of thermal transformation COP is calculated [5] or entropy analysis methods are used [19, 20]. However, it is more correct to define another energy efficiency indicator — exergy efficiency. It allows estimating net energy losses that are called exergy.

In this article, the thermo-economic analysis is carried out using the postulates first proposed by J. Tsatsaronis [21]. Such an approach to assessing the energy efficiency of low-temperature refrigeration machines, including cascade ones, is nowadays increasingly widespread and is modern and more correct, as it allows assesings the real efficiency of any power plant regardless of its complexity. For example, paper [8] deals with the exergy analysis of a cascade refrigeration cycle for the ratio of R404A/R744 refrigerants, while paper [9] deals

Table 4. **Summary Results of the Calculation of the Modified Cascade Cycle (lower cascade R290, upper cascade R717)**

Parameter	π_{st}^U	π_{st}^L	t_{2U}	t_{5U}	t_{2L}	x_{5U}	x_{5L}	COP
Unit of measurement	—	—	°C	°C	°C	—	—	—
Value	2.157	5.326	55	75	24	0.10	0.213	2.07

Table 5. **Summary Results of the Calculation of a Three-Stage VCRM (R717)**

Parameter	π_{st}	t_2	t_4	t_7	x_{13}	COP
Unit of measurement	—	°C	°C	°C	—	—
Value	3.198	33	56	92	0.073	2.22

Note: $\pi_{st} = p_c / p_{i2} = p_{i2} / p_{i1} = p_{i1} / p_b$ is the degree of pressure increase in the compressor stage; t_2, t_5, t_7 are refrigerant temperature after theoretical compression in the first, second, and third stages of the compressor, respectively; x_{13} is the degree of dryness of the refrigerant before the evaporator

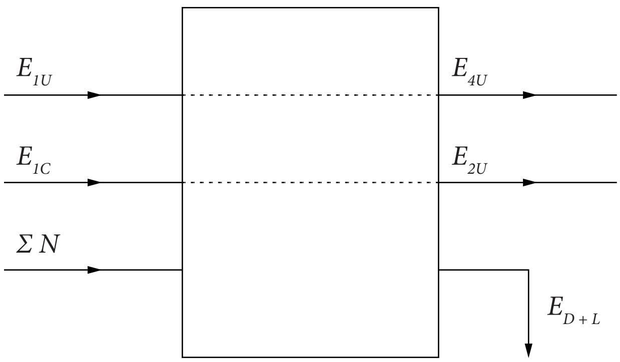

Fig. 2. Exergy flows in a refrigeration machine

with R1132a/R1234yf and 48 other refrigerant groups. In [22], the authors analyse the exergy efficiency of the R404A/R744 vapour cascade cycle. This method of analysis is also available in [10] and others. The increasing use of this method makes it possible to compare research results and draw unambiguous conclusions and recommendations that will contribute to the further development of refrigeration technology in the world.

The exergy efficiency of a refrigeration machine, regardless of the scheme, is determined by the formula:

$$\varepsilon = \frac{\sum E_p}{\sum E_f}, \tag{4}$$

where $\sum E_p$ is the sum of the product's energy flows; $\sum E_f$ is the sum of fuel exergy flows.

The refrigeration machine is considered by the "black box" method (see Fig. 2). In general, the refrigeration machine scheme is supplied with the following: the exergy flow of the recycled medium (index "u") to the condenser, the exergy flow of the cooled medium (index "c") to the evaporator, and the total electrical power of the compressor motor drives. The flows with indices "u" and "c" pass through the refrigeration scheme and correspond to the purpose of the refrigeration machine. At the same time, their state changes from "1u" to "4u" and from "1c" to "2c". Part of the exergy flow in the amount of E_{D+L} is lost.

Table 6. **Effect of the Upper Stage Condensation Temperature on the Energy Efficiency of the Cascade Cycle**

t_{cL}, °C	–20	–10	0	10	20
COP	1.94	2.01	2.00	2.00	1.91

The general expression of the fuel flows in a refrigeration machine will look like this

$$\sum E_f = \sum N \tag{5}$$

and for the product flow we get

$$\sum E_p = E_{2c} - E_{1c}. \tag{6}$$

The flow of product $\sum E_p$ is determined by expression (12) regardless of the refrigeration machine's scheme design.

The calculation of fuel consumption $\sum E_f$ depends on the chosen scheme solution:

- single-stage VCRM:

$$\sum E_f = N_C^{el} + \sum N_P^{el} + \sum N_F^{el}; \tag{7}$$

- two-stage VCRM:

$$\sum E_f = N_{HPC}^{el} + N_{LPC}^{el} + \sum N_P^{el} + \sum N_F^{el}; \tag{8}$$

- cascade VCRM:

$$\sum E_f = N_{CL}^{el} + N_{CU}^{el} + \sum N_P^{el} + \sum N_F^{el}; \tag{9}$$

- modified cascade VCRM:

$$\sum E_f = N_{HPCU}^{el} + N_{LPCU}^{el} + N_{CL}^{el} + \sum N_P^{el} + \sum N_F^{el}; \tag{10}$$

- three-stage VCRM:

$$\sum E_f = N_{1SC}^{el} + N_{2SC}^{el} + N_{3SC}^{el} + \sum N_P^{el} + \sum N_F^{el}. \tag{11}$$

Here N_C^{el} is the electric power of the compressor of the single-stage VCRM; N_{HPC}^{el}, N_{LPC}^{el} are the electric power of the first and second stages of the compressor of the two-stage VCRM, respectively; N_{CL}^{el}, N_{CU}^{el} are the electrical power of the compressors of the lower and upper stages of a cascade refrigeration machine, respectively; N_{HPCU}^{el}, N_{LPCU}^{el}, N_{CL}^{el} are the electrical capacities of the high and low pressure compressors of the upper stage and the lower stage compressor of the modified cascade cycle, respectively; N_{1SC}^{el}, N_{2SC}^{el}, N_{3SC}^{el} are the electrical capacities of 1—3 stages of the three-stage VCRM compressor; $\sum N_P^{el}$ is the total electrical power of the drives of pumps that pump liquid coolants, such as water through a condenser; $\sum N_F^{el}$ is the total electrical power of fans pumping gaseous coolants, such as air through an air cooler.

The numerical values of the powers $\sum N_p^{el}$ and $\sum N_f^{el}$ are close to zero compared to other values of electric power, so these values are neglected when performing calculations.

Using the above expressions (4)—(11), the exergy efficiency and its components are calculated depending on the scheme solution. The results of the calculations are presented in Table 7.

As can be seen from the results presented in Table 7, the choice of a scheme solution affects the energy efficiency of the cycle. As is well known, for low-temperature refrigeration cycles, it is advisable to use multi-stage or cascade refrigeration cycles. However, the increase in energy efficiency of a refrigeration machine is faster for multi-stage machines compared to cascade machines. For example, a one-stage increase when switching from a one- to a two-stage refrigeration cycle results in a 21 % increase in both the thermal conversion factor and exergy efficiency, and a subsequent switch from two-stage to three-stage compression results in a 7 % increase in energy efficiency. It should be borne in mind that such a transition also complicates the scheme, potentially reducing its reliability and increasing the cost of installation.

The use of cascade, including modified, refrigeration cycles, even with the use of environmentally friendly and most energy-efficient refrigerants, such as R290 (propane) as a lower stage refrigerant, results in lower efficiency indicators compared to multi-stage refrigeration cycles using R717. This can be explained by the lower thermodynamic efficiency of other refrigerants compared to ammonia, as well as the presence of additional thermodynamic losses during heat transfer from the lower to the upper stage in the evaporator-condenser. At the same time, the use of a modified cascade refrigeration cycle made it possible to obtain the lowest maximum refrigerant temperatures and the lowest degree of pressure increase in the upper stage. This is also confirmed in [3], where a thorough similar numerical study was performed. However, it does not analyse the ratio of R717/R290 as refrigerants in the upper and lower stages.

The article has examined approaches to improving the energy efficiency of a low-temperature refrigeration machine while considering the environmental safety of refrigerants. Two principal strategies have been analyzed: the selection of refrigerants or their combinations, and the choice of the circuit configuration. Both traditional thermodynamic methods and modern exergy-based

Table 7. **Refrigeration Cycle Calculation Results (condensation temperature 45 °C, boiling point –45 °C; cooling capacity 100 kW)**

Schematic solution	$N_{(cycle)}$, kW [1]	COP [1]	$\sum E_f$, kW [2]	$\sum E_p$, kW	ε [2]
Single-stage VCRM with condensate subcooler (R717)	58.48	1.71	85.254	24.454	0.287
Two-stage VCRM with intermediate vessel and coil (R717)	48.31	2.07	70.384	24.454	0.347
Cascade refrigerating machine (upper stage refrigerant R717; lower stage refrigerant R290)	50.00	2.00	73.334	24.454	0.333
Modified cascade chiller (upper stage refrigerant R717; lower stage refrigerant R290)	48.31	2.07	74.024	24.454	0.330
Three-stage VCRM with full interstage cooling and sequential throttling (R717)	45.05	2.22	65.786	24.454	0.372

Note: 1 — internal capacity of the theoretical cycle; 2 — electrical power of the actual cycle (h_{trm} = 0.99 — compressor transmission efficiency; h_{mec} = 0.92 — compressor mechanical efficiency; h_s = 0.8 — compressor adiabatic efficiency; h_{el} = 0.94 — compressor electric drive efficiency; environmental parameters: pressure p_{env} = 100 kPa; T_{env} = 293 K approaches have been applied to evaluate the energy efficiency of the refrigeration cycle.

R717 has been used as the basis of the study because it satisfies environmental and energy-efficiency requirements and, as the literature review has shown, has been largely — and undeservedly — overlooked due to its flammability and toxicity. Nevertheless, with the advancement of automation systems and adherence to safety standards, the use of R717 in industrial refrigeration systems is feasible and justified.

The numerical analysis of various schematic solutions for refrigeration machines, summarized in Table 7, has shown that the highest coefficient of performance (COP = 2.22) and exergy efficiency (ε = 0.372) have been obtained for a three-stage refrigeration machine with full interstage cooling and sequential throttling (refrigerant R717). Therefore, for low-temperature refrigeration applications, the use of multi-stage refrigeration cycles has proven to be more effective than cascade refrigeration circuits.

When a two-stage VCRM with a coil-type intermediate vessel and a modified cascade refrigeration machine — a new design option considered as a promising low-temperature solution — have been evaluated, the COP and ε have been at least by 7% lower as compared with the three-stage cycle. However, an assessment of intermediate operating parameters has indicated that the modified cascade refrigeration machine deserves preference, as it provides more favorable compressor operating conditions — namely, lower pressure ratios and lower final refrigerant temperatures after compression. These features reduce compressor size and weight and have contributed to greater reliability and longer service intervals.

The study of lower-stage refrigerant options in terms of cycle energy efficiency has demonstrated for the first time that the combination of R717/R290 has achieved the highest exergy efficiency. These refrigerants are environmentally friendly and fully satisfy application requirements. By contrast, commonly used lower-stage refrigerants such as R13 and R23 have been found to be 11.1% less energy-efficient, with carbon dioxide showing an even lower efficiency (11.7% decrease).

The paper has also shown that when selecting the intermediate temperature of phase transition in a cascade cycle, the condensation temperature of the lower stage can be determined using expression (3).

REFERENCES

1. Kozin, V. M., Vertepov, Y. M. (2014). *Refrigeration technologies: textbook*. Sumy [in Ukrainian].
2. Oginni Olarewaju Thomas, Bukola Olalekan Bolaji, Olatunde Ajani Oyelaran, Kazeem Aderemi Bello, Ezekiel Adegoke Fadiji, and Francis Olusesi Borokini. (2023). An Overview of a Two-Stage Vapour Compression Cascade Refrigeration System for Low-Temperature Applications. *Current Journal of Applied Science and Technology*, 42(42), 10—18. https://doi.org/10.9734/cjast/2023/v42i424269
3. Saeed, M. Z., Contiero, L., Blust, S., Allouche, Y., Hafner, A., Eikevik, T. M. (2023). Ultra-Low-Temperature Refrigeration Systems: A Review and Performance Comparison of Refrigerants and Configurations. *Energies*, 16(21), 7274. https://doi.org/10.3390/en16217274
4. Rangel, V., Almeida, A. (2021). Cascade refrigeration system for low temperatures using natural fluids. *Revista de Engenharia Térmica*, 20(2), 20—26. http://dx.doi.org/10.5380/reterm.v20i2.81783
5. Morozyuk, T. V. (2006). *Theory of refrigeration machines and heat pumps*. Odesa [in Russian].
6. REFPROP. URL: https://trc.nist.gov/refprop/MINIREF/MINIREF.HTM (Last accessed: 21.02.2025).
7. Restrepo, G., Weckert, M., Bruggemann, B., Brüggemann, R., Gerstmann, S., Frank, H. (2008). Ranking of Refrigerants. *Environment Science & Technology*, 56—61. URL: https://www.unipamplona.edu.co/unipamplona/portalIG/home_10/recursos/general/pag_contenido/articulos/06082010/art_guill_ranking_patterns.pdf (Last accessed: 21.02.2025).
8. Jeon, M-J. (2022). Experimental Analysis of the R744/R404A Cascade Refrigeration System with Internal Heat Exchanger. Part 2: Exergy Characteristics. *Energies*, 15(3), 1251. https://doi.org/10.3390/en15031251

9. Sannan, Salabat Butt, Uthpala, A. Perera, Takahiko, Miyazaki, Kyaw, Thu, Yukihiro Higashi. (2024). Energy, exergy and environmental (3E) analysis of low GWP refrigerants in cascade refrigeration system for low temperature applications. *International Journal of Refrigeration*, 160, 373—389. https://doi.org/10.1016/j.ijrefrig.2023.12.020
10. Kumar, Vipin, Karimi, Munawar, Kamboj, Sandeep. (2020). Comparative analysis of cascade refrigeration system based on energy and exergy using different refrigerant pairs. *Journal of Thermal Engineering*, 6(1), 106—116. https://doi.org/10.18186/thermal.671652
11. Zhili, Sun, Jianhua, Quan, Yi'an, Wang, Ruishen, Li, Jie, Peng, Haoran, Zhang, Jiapei, Xu, Huan, Sun, Shen, Tian, Shengming, Dong, Kaiyong, Hu, Wenxiang, Chen. (2022). Performance comparison of the single-refrigerant cascade refrigerating system. *Energy Reports*, 8, 8259—8270. https://doi.org/10.1016/j.egyr.2022.06.055
12. COOLPACK. URL: http://www.holodonline.com/programme/coolpack-1-5. (Last accessed: 21.02.2025).
13. Bingming, W., Huagen, W., Jianfeng, L., Ziwen, X. (2009). Experimental Investigation on the Performance of NH_3/CO_2 Cascade Refrigeration System with Twin-Screw Compressor. *International Journal of Refrigeration*, 32(6), 1358—1365. https://doi.org/10.1016/j.ijrefrig.2009.03.008
14. Lee, T., Liu, C., Chen, T. (2006). Thermodynamic analysis of optimal condensing temperature of cascade-condenser in CO_2/NH_3 cascade refrigeration systems. *International Journal of Refrigeration*, 29(7), 1100—1108. https://doi.org/10.1016/j.ijrefrig.2006.03.003
15. Sachdeva, G., Jain, V., Kachhwaha, S. S. (2014). Performance study of cascade refrigeration system using alternative refrigerants. *World Academy of Science, Engineering and Technology International Journal of Mechanical and Mechatronics Engineering*, 8(3), 522—528. htpps://doi.org/10.5281/zenodo.1091148
16. Yamaguchi, H., Niu, X.-D., Sekimoto K., Neksa, P. (2011). Investigation of Dry Ice Blockage in an Ultra-Low Temperature Cascade Refrigeration System Using CO_2 as a Working Fluid. *International Journal of Refrigeration*, 34(2), 466—475. https://doi.org/10.1016/j.ijrefrig.2010.11.001
17. Sokolovska-Efimenko, V. V., Gaiduk, S., Moshkatyuk, A., Yerema, V. (2018). Thermodynamic analysis of the cascade refrigeration machine of a marine refrigeration container. *Refrigeration Engineering and Technology*, 54(6), 28—38 [in Ukrainian].
18. Dopazo, A., Femandez-Seara, J. (2011). Experimental Evaluation of a Cascade Refrigeration System Prototype with CO_2 and NH_3 for Freezing Process Application. *International Journal of Refrigeration*, 1, 257—267. https://doi.org/10.1016/j.ijrefrig.2010.07.010
19. Morosuk, T., Nikulshin, R., Morosuk, L. (2006). Entropy-cycle method for analysis of refrigeration machine and heat pump cycles. *Thermal Science*, 10(1), 111—124.
20. Nikulshin, R. K., Morozyuk, L. I. (2012). Entropic method of modeling and analysis of two-stage cycles of refrigeration machines and heat pumps. *Sustainable development and artificial cold: materials of the VIII International scientific and technical conference, devoted to the 90th anniversary of the Odesa State Academy of Refrigeration. 90th anniversary of Odesa State Academy of Cold (8—10 October, 2012, Odesa, Ukraine),* 141—146 [in Russian].
21. Tsatsaronis, J. (2002). *Interaction of thermodynamics and economics to minimize the cost of energy conversion system.* Odesa [in Russian].
22. Xiaonan, Chen, Qichao, Yang, Weikai, Chi, Yuanyang, Zhao, Guangbin, Liu, Liansheng, Li. (2022). Energy and exergy analysis of NH_3/CO_2 cascade refrigeration system with subcooling in the low-temperature cycle based on an auxiliary loop of NH_3 refrigerants. *Energy Reports*, 8, 1757—1767. https://doi.org/10.1016/j.egyr.2022.01.004

Received 11.03.2025
Revised 22.06.2025
Accepted 07.08.2025

В.М. Козін [1] (https://orcid.org/0000-0001-9821-7774),
С.О. Шарапов [2] (https://orcid.org/0000-0002-8433-8580),
Г.В. Барсукова [1] (https://orcid.org/0000-0002-4261-2182)

[1] Сумський національний аграрний університет,
вул. Герасима Кондратьєва, 160, Суми, 40021, Україна,
+380 542 70 1158, admin@snau.edu.ua
[2] Сумський державний університет
вул. Петропавлівська, 59, Суми, 40014, Україна,
+380 542 66 5124, info@sumdu.edu.ua

ВПЛИВ СХЕМНОГО РІШЕННЯ НИЗЬКОТЕМПЕРАТУРНОГО ХОЛОДИЛЬНОГО КОНТУРУ НА ОСНОВІ ЕКОЛОГІЧНО БЕЗПЕЧНИХ ХОЛОДОАГЕНТІВ НА ЕНЕРГОЕФЕКТИВНІСТЬ ЦИКЛУ

Вступ. Штучний холод від –40 °C і нижче широко використовують для різних електротехнологічних процесів. Водночас від вибору холодильного агенту та схеми циклу залежить не лише енергетична ефективність, а й екологічна безпека холодильної установки.

Проблематика. Для умов глибокого холоду використовують багатоступеневі або каскадні схеми з раціональним підбором пар холодоагентів задля максимальної ефективності.

Мета. Оцінити вплив схемного рішення з урахуванням екологічних властивостей холодоагентів на енергоефективність низькотемпературного холодильного циклу через коефіцієнт термотрансформації (COP) та ексергетичний ККД (ε) й надати практичні рекомендації щодо вибору схеми і холодоагентів.

Матеріали й методи. Застосовано термодинамічний та ексергетичний порівняльний аналіз теоретичних низькотемпературних холодильних циклів, виконано їх числове моделювання у *Refprop* відповідно до заданих температур ($t_{конд}$ = 45 °C, $t_{кип}$ = –45 °C). Як об'єкт використано двоступеневу парокомпресійну холодильну машину (ПКХМ) з проміжною ємністю та змійовиком, каскадну холодильну машину для пар холодильних агентів R717/R13, R717/R23, R717/R290, R717/R744, R717/R32, модифікований каскадний цикл, триступеневу ПКХМ.

Результати. Найбільшу енергоефективність (COP = 2,22; ε = 0,372) досягнуто для триступеневої ПКХМ на R717. Серед традиційних каскадних холодильних циклів найбільш енергоефективною (COP ≈ 2,00) та екологічною парою є R717/R290. Модифікований каскадний цикл перевершив за ефективністю традиційні каскадні схеми (COP ≈ 2,07), а його застосування дозволяє суттєво зменшити термомеханічні параметри, що позитивно впливає на ресурс компресорів.

Висновки. Для низькотемпературних застосувань доцільно віддавати перевагу багатоступеневим ПКХМ на R717, які за ефективністю перевершують каскадні схеми. Якщо ключовим є підвищення надійності установки, доцільно розглянути використання модифікованого каскадного циклу на R717/R290. Для такого циклу запропоновано правило вибору проміжної температури.

Ключові слова: енергетична ефективність, ексергетична ефективність, електротехнологія, енергозбереження, холодильна машина, екологічність, каскадний холодильний цикл, багатоступеневий холодильний цикл.