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A NEW APPROACH TO OPTIMIZATION OF MIXED REFRIGERANT COMPOSITION

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Abstract

Joule-Thomson cryo-refrigerators operating on mixed working fluids have significant advantages over chillers using pure refrigerants. When optimizing the composition of zeotropic refrigerant mixtures, it is necessary to take into account the peculiarities of the operation of volumetric compressors. It is known that the flow rate of a reciprocating compressor significantly depends on the compression ratio and the compressor suction pressure. Therefore, it is impractical to optimize the composition of zeotropic refrigerant mixtures at a fixed molar flow rate, as is done in many studies. This paper describes a method for optimizing the operation of a refrigeration machine operating on a five-component zeotropic mixture of refrigerants. The maximum cooling capacity of the unit at the temperature of 120 K, which is based on a hermetic compressor TAG 2513Z, was chosen as the objective function. The following parameters were varied during the optimization: compressor discharge and suction pressures, the composition of the five-component working mixture, as well as the temperature upstream of the throttle valve, and the temperature at the inlet to the phase separator. As a result of processing the results of the numerical experiment, an analytical expression was obtained that approximates the operation of the refrigeration unit depending on the eight varied parameters. This made it possible to find the optimal operating mode of the refrigeration machine, which achieves maximum cooling capacity. At the optimum operating mode of the refrigeration unit, the suction pressure is 2.35 bars, and the discharge pressure is 16.0 bars. With the optimal composition of the working substance, the maximum cooling capacity of 147.7 W with energy consumption by the compressor of 2.36 kW is achieved.
Keywords: Klimenko cycle; Mixed refrigerant; Compressor; Volumetric flow rate; Optimization; Cryorefrigerator.

НОВИЙ ПІДХІД ДО ОПТИМІЗАЦІЇ СКЛАДУ СУМІШЕВОГО ХОЛОДОАГЕНТУ

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Анотація

Кріорефрижератори Джоуля-Томсона, що працюють на сумішевих робочих тілах, мають значні переваги перед холодильними машинами, які використовують чисті холодоагенти. В процесі оптимізації складу зеотропних сумішей холодоагентів необхідно враховувати особливості роботи компресорів об'ємної дії. Відомо, що подача поршневого компресора суттєво залежить від ступеня стиснення і тиску нагнітання компресора. Тому оптимізувати склад зеотропних сумішей холодоагентів за фіксованої молярної витрати, як це робиться в багатьох дослідженнях, недоцільно. У цій статті описано метод оптимізації роботи холодильної машини, що працює на п'ятикомпонентній зеотропній суміші холодоагентів. В якості цільової функції було обрано максимальну холодопродуктивність установки, побудованої на базі герметичного компресора TAG 2513Z, за температури об'єкта охолодження 120 K. Під час оптимізації варіювалися такі параметри: тиск нагнітання і всмоктування компресора, склад п'ятикомпонентної робочої суміші, а також температура перед дросельним вентилем і температура на вході в фазовий сепаратор. В результаті обробки результатів чисельного експерименту було отримано аналітичний вираз, який апроксимує роботу холодильної установки в залежності від восьми параметрів, що варіювалися. Це дозволило знайти оптимальний режим роботи холодильної машини, за якого досягається максимальна холодопродуктивність. За оптимального режиму роботи холодильної машини тиск всмоктування становить 2.35 бар, а тиск нагнітання – 16.0 бар. За оптимального складу робочої речовини досягається максимальна холодопродуктивність 147.7 Вт за витрати енергії компресором 2.36 кВт.
Ключові слова: цикл Клименка; змішаний холодоагент; компресор; об'ємна витрата; оптимізація; кріорефрижератор.

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Introduction

Low-capacity refrigerators (compressor flow rate under 25 m^3/h) for sample temperature control in the temperature range −90...−160 °C play an important role in many areas of science and technology.

In medicine, for example, they are used for the cryopreservation of medical and biological samples and for local cryogenic treatment of various organs and tissues. In engineering, they are used to cool critical elements in electronic and optoelectronic devices and high-temperature superconductor devices. In vacuum technology, they are used to purify gases. In metallurgy, they are used for cryogenic treatment to improve material properties, and so on.

Joule-Thomson (JT) refrigerators are the most common and simplest devices for solving the above tasks. Compared to pure refrigerants, Joule-Thomson coolers using zeotropic mixtures of refrigerants have great technical and economic advantages. Much research has been done in this area at many institutes around the world [1–9].

According to the published literature, the hydrocarbon zeotropic mixture is used to further improve the performance of coolers by temperature smoothing in the cooler heat exchanger [10–12]. These properties can be used to achieve a better temperature match between the cold and hot working fluid streams to reduce heat transfer irreversibility and improve system performance.

Up to now, most mathematical models of refrigeration cycles operating on zeotropic mixtures have been developed based on the assumption that a given molar mass of the working fluid circulates in the unit [13–18]. However, this approach does not reflect the reality of real-world conditions.

It is known that the delivery of a reciprocating compressor significantly depends on the compression ratio and the compressor discharge pressure. Therefore, it is impractical to optimize the composition of zeotropic refrigerant mixtures at a fixed molar flow rate, as is done in many studies.

When optimizing the composition of zeotropic refrigerant mixtures, it is necessary to take into account the peculiarities of the operation of volumetric compressors. Therefore, when optimizing the cycles of Joule-Thomson cryocoolers, it is advisable to choose the maximum cooling capacity at a given temperature level for a given compressor, or the minimum temperature reached at a given cooling capacity, again for a given compressor, as the objective function.

In this case, the result of the optimization is not only the composition of the multi-component mixture but also the suction and discharge pressures of the compressor and the temperatures at some key points. Therefore, in the following, we will consider not just optimizing the composition of the working mixture, but also optimizing the operation of the refrigeration unit, which, in addition to the optimal composition of the working substance, also provides optimal suction and discharge pressures of the compressor and temperatures at some key points in the cycle.

Below, we will consider the optimization of the operation of a refrigeration machine built on the basis of a hermetic compressor TAG 2513Z manufactured by Tecumseh, operating on a zeotropic mixture of refrigerants.

This facility is designed to store biological objects, so it needs to provide cooling at a temperature level of 120 K. At this temperature, all chemical reactions in the cells completely stop, which is a condition for the almost infinite storage of these cells.

As is well known, cryogenic technology is a branch of engineering that deals with obtaining and using temperatures below 120 K. Therefore, the refrigeration unit discussed in this article can be classified as a cryo-refrigerator.

The use of a refrigeration unit instead of liquid nitrogen has a number of advantages, the most important of which is that there is no need to constantly maintain the level of liquid nitrogen, which must be imported from a long distance. The second advantage of such installations is that the total energy consumption for storing biological objects when cooled by a refrigeration unit is much lower than when using liquid nitrogen. The economic consequences of replacing liquid nitrogen with a refrigeration unit are even more significant, as there is no need for transportation costs for the supply of liquid nitrogen.

Modelling the operation of the TAG 2513Z compressor

The purpose of studying the operation of the TAG 2513Z compressor manufactured by Tecumseh is to create a relatively simple mathematical model of such a compressor, which should relate the suction and discharge pressures of this compressor to the compressor's volumetric flow rate.

In the future, this mathematical model will be used to optimize the composition of the working mixture and the operating mode of the plant designed to produce temperatures below 120 K.

In refrigeration systems operating with pure refrigerants, the boiling and condensing temperatures automatically determine the compressor suction and discharge pressures. When refrigeration systems operate with zeotropic refrigerant mixtures, the compressor suction and discharge pressures, along with the refrigerant mixture composition, are additional optimization parameters. Given that a compressor's delivery ratio is highly dependent on the suction and discharge pressures of that compressor, the task of optimizing the composition of the working mixture that ensures maximum cooling capacity when using a particular compressor becomes difficult to solve.

Therefore, in order to determine the optimal composition of the refrigerant mixture that ensures maximum cooling capacity at a given temperature level when using the TAG 2513Z compressor, it is necessary to give a mathematical description of the dependence of the flow rate of this compressor on the compression ratio and discharge pressure.

The mathematical model of the compressor was built using the passport data on the cooling capacity and power consumption of the TAG 2513Z compressor when it operates on R404a refrigerant provided by the manufacturer [19].

Knowing the condensation and boiling points of the refrigerant, you can determine the unit's volumetric cooling capacity and the compressor's volumetric flow rate at different compressor suction and discharge pressures.

The manufacturer's materials provide the volume described by each of the three pistons of the TAG 2513Z compressor per revolution of the shaft – 100.7 cm^3. This data can be used to determine the compressor's delivery ratio in each of the seven operating modes described in the manufacturer's materials.

Figure 1 shows the calculated dependence of the TAG 2513Z compressor's flow rate on the refrigerant's boiling point at different condensing temperatures.

The graphs above show that as the refrigerant condensation temperature increases, the compressor delivery ratio drops significantly. The general view of the obtained graphs indicates a complex and nonlinear nature of the dependence of the compressor delivery ratio on the boiling and condensation temperatures of the refrigerant.

Knowing the boiling and condensing temperatures of the refrigerant, you can find the inlet and outlet pressures of the compressor and the compression ratio for that compressor. Knowing this, you can plot the compressor's capacity factor against the refrigerant's compression ratio for different condensing temperatures.

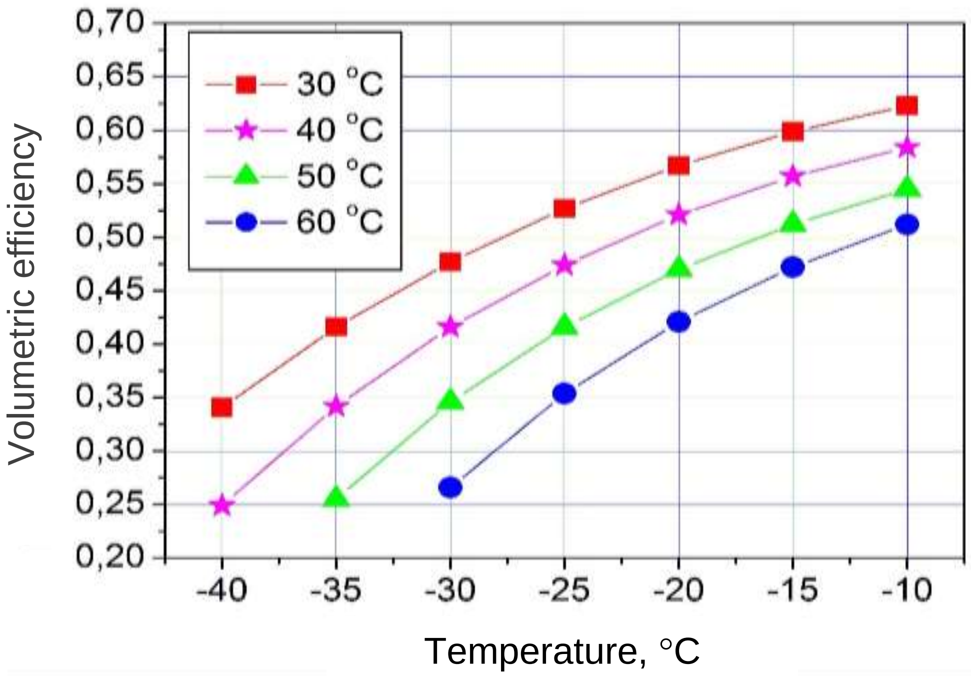

Fig. 1. Graphs of the TAG 2513Z compressor's volumetric efficiency versus refrigerant boiling point at different refrigerant condensing temperatures

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Figure 2 shows the graphs of the TAG 2513Z compressor's volumetric efficiency versus compression ratio and refrigerant condensing temperature (discharge pressure).

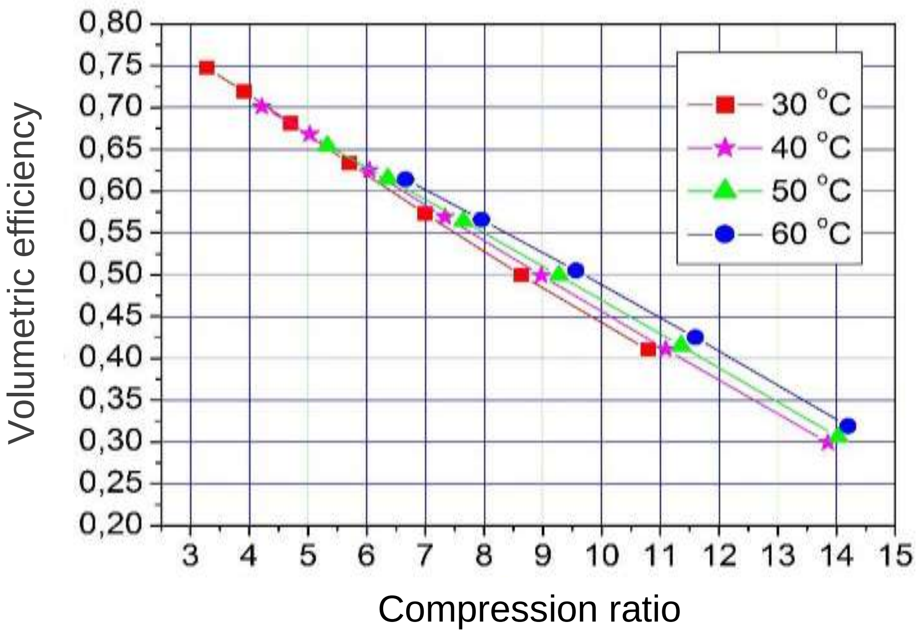

Fig. 2. Graphs of compressor volumetric efficiency versus compression ratio and refrigerant condensing temperature (discharge pressure) for the TAG 2513Z compressor

The dependence of the compressor feed ratio on the compression ratio was almost linear, and the dependence of the compressor feed ratio on the discharge pressure (condensation temperature) was relatively weak. Therefore, as a first approximation, the dependence of the delivery ratio on the discharge pressure can be neglected.

Thus, the study of the dependence of the TAG 2513Z refrigeration compressor's delivery ratio on various factors has resulted in an almost linear dependence of the compressor's delivery ratio on the gas compression ratio. This makes it possible to obtain a simple empirical dependence of the compressor's delivery ratio on the compression ratio, which is suitable for technical calculations.

As a result of approximating the data obtained by a linear relationship using the least square method, the following formula was obtained:

$$\eta = \mathrm{a} \cdot \xi + \mathrm{b}, \qquad (1)$$

where, η– is the compressor flow rate; ξ – is the compression ratio.

Based on this relationship, we obtain the following coefficients in Formula 1: a = –0.0453; b = 0.894.

Conducting and processing the results of a numerical experiment

The design scheme of the system designed to obtain temperatures below 120 K is shown in Figure 3. The system consists of a TAG 2513Z compressor (Compr_1), an air heat exchanger for overheating relief (Cooler_1), a heat recovery exchanger (Cooler_2- Heater_1), a phase separator (Flesh_4), a first throttle valve (Velve_1), a mixer (Mixer_1), a main heat exchanger (Cooler_3-Heater_13), a main throttle valve (Velve_2) and an evaporator (Heater_0).

As an objective function for optimizing the operation of the plant, the maximum cooling capacity at a temperature level of 120 K can be achieved using the TAG 2513Z compressor manufactured by Tecumseh. The following parameters were selected as the variables: concentrations of nitrogen, methane, ethane, and propane in the mixture on which the plant operates, compressor discharge pressure, compressor suction pressure, temperature at the inlet to the phase separator and temperature of the mixture before the main throttle.

The content of the fifth component of the mixture, isobutane, was determined from the material balance of the mixture.

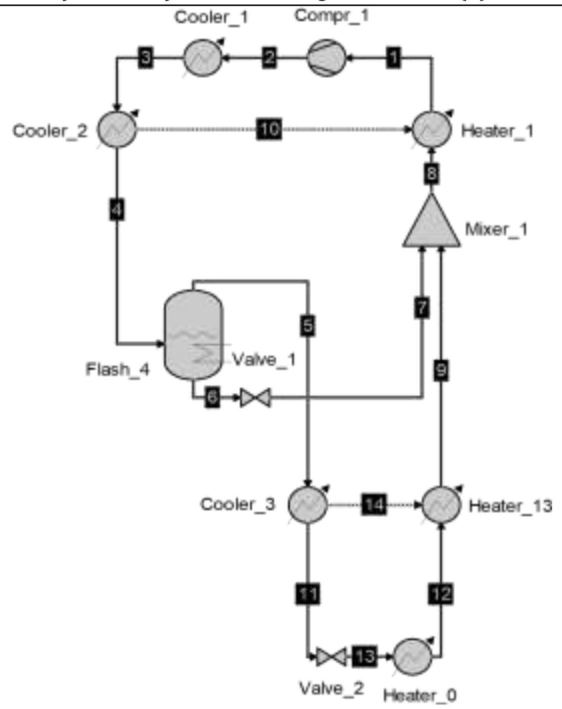

Fig. 3. Design scheme of the refrigeration unit

To optimize such a refrigeration machine, a three-level, eight-factor numerical experiment plan was drawn up [20–24]. The ultimate goal of the numerical experiment is to obtain an analytical dependence of the refrigeration capacity of the unit on eight parameters (factors). The list of natural and coded levels of the factorial experiment is given in Table 1. The coded values of the parameters to be varied are: –1, 0, 1.

Table 1

Natural and coded levels of the factorial experiment

№	Name of the factor	Natural value	Coded value
1	Compressor discharge pressure, bar	14	–1
		16	0
		18	1
2	Nitrogen content (molar fraction)	9	–1
		10	0
		11	1
3	Methane content (molar fraction)	47	–1
		48	0
		49	1
№	Name of the factor	Natural value	Coded value
4	Ethane content (molar fraction)	10	–1
		12	0
		14	1
5	Propane content (molar fraction)	3	–1
		4	0
		5	1
6	Compressor suction pressure, bar	2.0	–1
		2.2	0
		2.4	1
7	Temperature at the inlet to the separator, K	288	–1
		289	0
		290	1
8	Temperature before the main throttle, K	134	–1
		135	0
		136	1

The cooling capacity of the plant at a temperature level of 120 K was calculated in the COCO ChemSep program at the same mass flow rate of the mixture – 100 g/s.

Knowing the dependence of the TAG 2513Z compressor's volumetric flow rate on the compression ratio and the gas density at the compressor suction, it is easy to calculate the cooling capacity of the system when working with the TAG 2513Z compressor.

The plan of the multivariate numerical experiment is presented in Table 2.

Table 2

Numerical experiment planning matrix

№	Conditional level of the factor								
	1	2	3	4	5	6	7	8	R,W
1	–1	–1	–1	–1	–1	–1	–1	–1	105.0
2	1	–1	–1	1	1	1	1	1	108.3
3	0	0	–1	1	0	0	0	–1	132.7
4	1	0	–1	–1	–1	1	1	1	145.9
5	–1	1	–1	1	0	–1	–1	1	113.1
6	0	1	–1	–1	1	0	0	1	145.1
7	0	–1	0	0	0	1	–1	–1	142.5
8	1	–1	0	1	1	–1	0	0	122.0
9	–1	0	0	0	1	0	1	1	132.1
10	0	0	0	1	–1	1	–1	1	142.8
11	–1	1	0	1	0	0	–1	–1	128.9
12	1	1	0	0	–1	–1	0	–1	115.1
13	–1	–1	1	–1	–1	1	0	0	132.5
14	0	–1	1	0	0	–1	1	0	119.1
15	–1	0	1	0	1	1	0	0	143.3
16	1	0	1	–1	0	0	–1	–1	133.8
17	0	1	1	–1	1	–1	1	0	120.2
18	1	1	1	0	–1	0	–1	0	132.3

In total, eighteen calculations of the plant layout were performed with different values of the eight factors varied. The cooling capacity of the unit recalculated taking into account the actual compressor volume flow rate, is shown in the last column of Table 2.

This is the result of the numerical experiment.

The results of the cooling capacity calculations on a mixture of 5 components were processed to obtain an analytical dependence of the cooling capacity of the unit on the eight varied factors.

We will look for this analytical function of eight independent parameters as a sum of functions, each of which depends on only one factor:

$$R = \sum_{i=1}^{8} f_i(x), \qquad (2)$$

where

$$f_i(x) = a_i x^2 + b_i x + c_i \qquad (3)$$

When processing the results of the numerical experiment, the average value of the results of the numerical experiments in which this variable took the same value was found for each factor:

$$f_i\left(x^{-1}\right) = \frac{\sum_{6} R_j^{-1}}{6}; \qquad f_i\left(x^{0}\right) = \frac{\sum_{6} R_j^{0}}{6};$$

$$f_i\left(x^{+1}\right) = \frac{\sum_{6} R_j^{+1}}{6}; \qquad (4)$$

Then, using the three points obtained for each variable, a second–degree polynomial was built. For this purpose, the Lagrange interpolation formula was used:

$$f_i(x) = \left[\frac{f_i\left(x^{-1}\right)}{2} - f_i\left(x^{0}\right) + \frac{f_i\left(x^{+1}\right)}{2}\right]x^2 + \left[\frac{f_i\left(x^{+1}\right)}{2} - \frac{f_i\left(x^{-1}\right)}{2}\right]x + f_0. \qquad (5)$$

To simplify the calculations, we introduce new variables:

$$a_i = \left[\frac{f_i\left(x^{-1}\right)}{2} - f_i\left(x^{0}\right) + \frac{f_i\left(x^{+1}\right)}{2}\right]; \qquad b_i = \left[\frac{f_i\left(x^{+1}\right)}{2} - \frac{f_i\left(x^{-1}\right)}{2}\right]. \qquad (6)$$

Thus, as a result of processing the results of the numerical experiment, an analytical expression was obtained that approximates the behavior of the refrigeration unit depending on the eight

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varying parameters. This made it possible to find the optimal mode of operation of the refrigeration machine, which achieves maximum cooling capacity.

Figure 4 shows a histogram that gives a visual representation of the error of the analytical expression for estimating the cooling capacity of the unit.

As you can see from the histogram above, the agreement between the predicted values of the function and its true value is satisfactory.

The maximum cooling capacity of the system that can be achieved with the TAG 2513Z compressor is calculated using the formula:

$$R_{\max} = \sum_{8} \left\{ a_i \left(\frac{b_i}{2a_i} \right)^2 + b_i \left(\frac{b_i}{2a_i} \right) \right\}, \qquad (7)$$

where $\left(-\frac{b_i}{2a_i} \right)$ is the optimal value of the i–th coded parameter at which the maximum cooling capacity is achieved.

The predicted cooling capacity of the unit was 150.6 W. Substituting the optimal operating parameters of the refrigeration unit into the COCO ChemSep® program yields a cooling capacity of 147.7 W. This is higher than any of the cooling capacity values obtained in the variant calculations (see Table 2).

Table 3 shows the results of the refrigeration unit optimization.

Table 3

Optimization results of a refrigeration system based on the TAG 2513Z compressor

№	Name of the factor	Natural value	Coded value
1	Compressor discharge pressure, bar	16.03 bar	0.01
2	Nitrogen content (molar fraction)	10.1 %	0.07
3	Methane content (molar fraction)	48.5 %	0.48
4	Ethane content (molar fraction)	11.1 %	–0.45
5	Propane content (molar fraction)	4.0 %	0.00
6	Compressor suction pressure, bar	2.35 bar	0.74
7	Temperature at the inlet to the separator, K	288.9 К	–0.13
8	Temperature before the main throttle, K	136.8 К	1.85

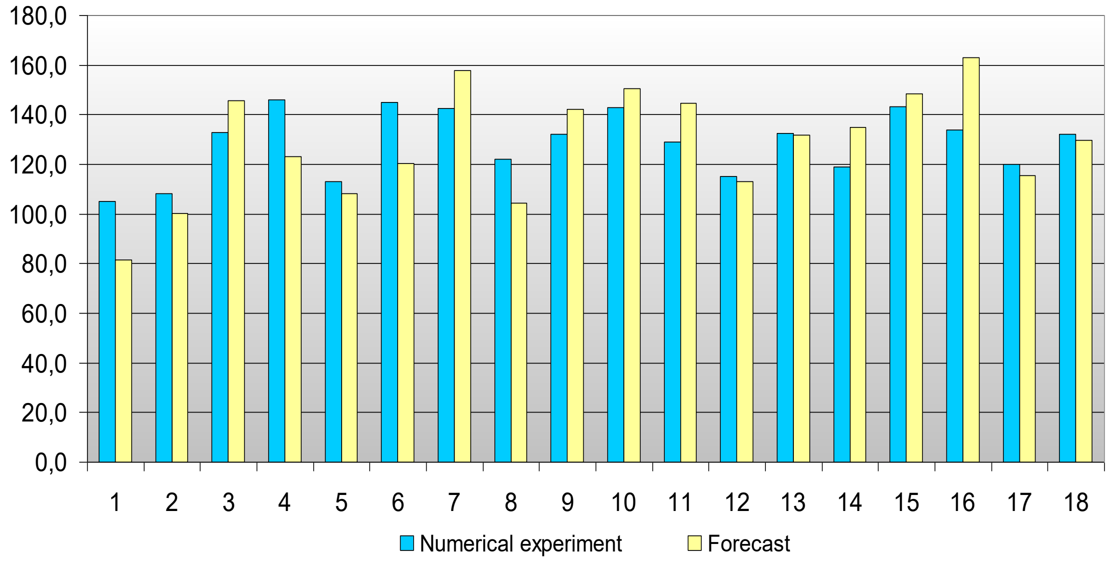

Fig. 4. Comparison of the cooling capacity of the unit calculated by the analytical expression and the results of the numerical experiment.

By processing the results of numerical experiments, the optimal operating parameters of the refrigeration unit based on the TAG 2513Z compressor were found. The optimal operating mode of the unit was as follows: suction pressure – 2.35 bar, discharge pressure – 16.03 bar, working fluid concentration: nitrogen – 10.1 %, methane – 48.5 %, ethane – 11.1 %, propane – 4.0 %, isobutene – 26.3 %; the temperature at the inlet to the phase separator – 288.9 K, temperature before the main throttle – 136.8 K. In this operating mode,

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the maximum cooling capacity of the unit is achieved, which is 147.7 W.

Conclusions

During the multivariate numerical experiment, the values of the optimization parameters were selected. At the optimal operating mode of the refrigeration unit, the suction pressure is 2.35 bars and the discharge pressure is 16.03 bars. With the optimal composition of the working substance, the maximum cooling capacity of 147.7 W with energy consumption by the compressor of 2.36 kW is achieved.

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