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# Rotary-Blade Cooling Machine

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The characteristics of a rotary-bladed machine operating in an open-loop gas cycle for cold production are considered. Mathematical modeling of physical and technical processes has been carried out within the framework of numerical integration of the unsteady Reynolds-averaged system of Navier-Stokes equations (URANS). For this purpose, the ANSYS Fluent 2021 R2 software package has been used. The obtained results on the integral performance indicators of the rotary-blade machine allow us to count on its effective use.

**Keywords:** rotary-blade machine, cooler, non-stationary heat exchange, numerical simulation.

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## Introduction

A number of technical devices are known, with varying degrees of efficiency, to solve the problem of generating cold or thermostating a limited area of space below ambient temperature. Various phenomena are used for this purpose that lead to a decrease in temperature. The most common of the cooling effects are phase transformations, expansion of compressed gas to produce external work (Siemens cycle), throttling (Joule-Thomson process), the vortex effect (Rank tube), thermoelectric cooling (Peltier effect), etc. The final choice of cooling medium is influenced not only by the possibility of heat dissipation at a given temperature, but also a combination of factors such as the availability of thermal, mechanical, or electrical energy necessary to solve the problem, the high efficiency of the implemented process, environmental considerations when using refrigerants, and the resource and reliability of autonomous operation.

An open-loop gas cycle design has certain advantages, all other things being equal. In this case, the environment itself is a working fluid, and the absence of an additional coolant greatly simplifies the design of the cooling generator.

The general design that implements such a cooling process is based on the use of a compressor, in which, as a result of gas compression, the temperature increases above the temperature of the walls of the heat exchanger to remove heat into the environment beyond the boundary of the cooled circuit and an expander that partially compensates for the work expended during gas compression. Both piston and vane machines can be used as a compressor and expander. Their advantages and disadvantages are well known.

The advantages of vane machines include high productivity (gas consumption) and a relatively low level of aerodynamic losses due to the fact that the gas does not make the oscillatory motion typical of piston machines, there is no valve system regulating the intake and exhaust of the working fluid. The disadvantage of vane machines is the need for high rotor speeds and the technological complexity of manufacturing the flow part. In particular, the expansion turbine of P.L. Kapitza uses this design [1], combining compressor and expander stages on one shaft. Piston machines, on the contrary, can offer low costs and their manufacture is less labor-intensive [2].

In addition to the two types of technical devices mentioned above, which perform gas compression and expansion, some types of rotary machines allow combining the two approaches. In a classic Wankel engine, each of the three working chambers sequentially performs filling, compression, expansion, and exhaust cycles per rotation of the rotor [3]. Another example is rotary-plate devices widely used as compressors, pumps, and hydraulic drives [4,5]. In double-acting rotary-plate machines (two-chamber elliptical design), it is possible to combine the functions of a compressor and an expander. Such designs have not yet been developed and are practically not discussed outside of the patent literature, where they are mainly considered as internal combustion engines [6]. Let us note the papers devoted to the study of the Wankel engine [7] and the rotary-plate machine [8], which solve problems of numerical modeling related to this study related to the definition of kinematic laws of displacement of computational domains, the use of a dynamic computational grid, and the choice of a turbulence model.

Let's pay attention to the rotary-vane machine [9, 10], which in the generally accepted classification is also called rotary-piston. It combines both the capabilities of the piston group (low and moderate shaft rotation speeds, high compression ratio) and the direct-flow movement of the working fluid, characteristic of vane and plate machines, and the absence of intake and exhaust valves. From the point of view of the compressor and expander layout, compared, for example, with a turbo expander, where two impellers are required for each of the compression or expansion processes, in a rotary vane machine, the same working chamber is used both as a compressor and as an expander per shaft revolution in different phases of the cycle. The working chamber itself has a simple design form, which gives the rotary-vane machine an indisputable advantage. The authors do not know of any studies that would analyze the use of a rotary-vane machine in the refrigeration cycle. This study focuses on the application of a rotary vane machine as a cooled medium generator.

The operation of a rotary-vane machine in compressor mode is studied in Ref. [11–14]. The authors pay special attention to the analysis of leaks in the non-contact seals of the engine. The experimental tests of the rotary-vane compressor proved its operability [12]. The possibility of using a rotary-vane compressor as part of a cooling system is indicated in Ref. [14]. However, in the schemes of cryogenic machines under consideration, Stirling piston machines act as expander cavities. It should be noted that the mentioned studies consider two ways of organizing the movement of work chambers. In the first variant, the vanes of the rotary vane machine „oscillate" (make a reciprocating motion), in the second variant, the vanes of the rotary vane machine „rotate" (reciprocating motion is combined with unidirectional rotational motion). In both cases, the motion conversion device is based on a mechanism with non-circular gears. In this paper, the cam mechanism of motion transformation is used to define the kinematic laws of motion of „rotating" vanes [15].

The simplest estimates of the performance of a rotary-vane machine in the cold generator mode are possible according to the indicator diagram [16]. In particular, it is possible to accurately determine the behavior of thermodynamic functions in the process of gas compression and expansion, however, the definition of gas exchange between machine elements and the description of the mode of unsteady heat exchange is extremely difficult within the framework of this model. The mathematical model in nodal values provides broader possibilities. The successful experience of using this model to calculate the characteristics of a rotary-vane engine with external heat supply is given in Ref. [17]. A mathematical model for calculating the operating mode of a rotary-vane engine in a two-dimensional formulation is considered in Ref. [18]. In these papers, the modeling of friction and heat transfer was carried out in a quasi-stationary formulation and was based on an integral methodology for evaluating ongoing processes. However, the unsteady process of gas flow and heat exchange in the pipe requires a more complete formulation of the problem. In this paper, the issues related to the regime of unsteady gas flow and heat exchange are considered in a three-dimensional approximation. The performance characteristics of the remaining gas cooler units were modeled in a two-dimensional approximation in a layer of variable thickness.

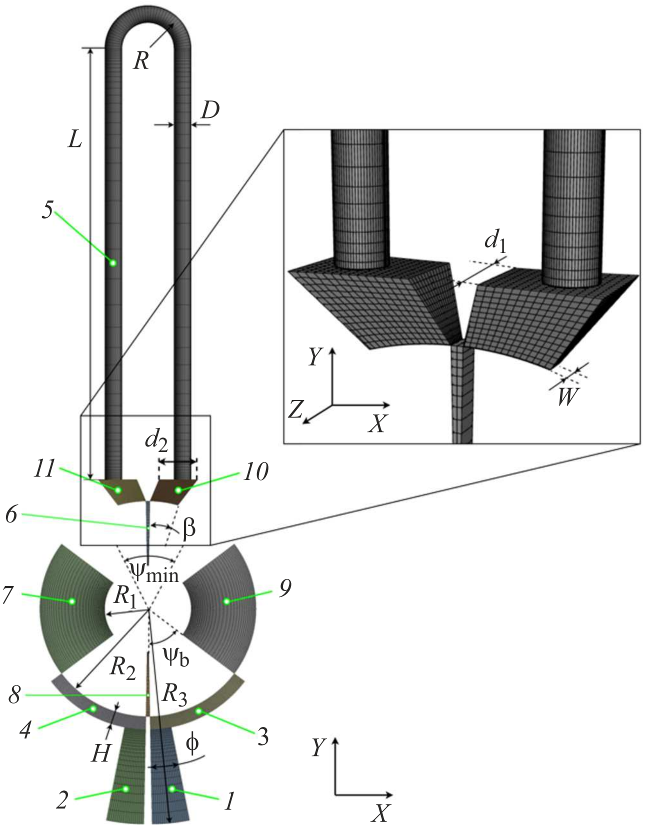


**Figure 1.** The design area of a refrigerating machine based on a rotary-vane engine.

## 1. Problem statement

Let us consider a design based on a four-vane machine. The calculated area for determining the characteristics of a rotary-vane gas cooler is shown in Fig. 1. It consists of cavities that serve to supply *1* and drain *2* the cooled gas. The inlet *3* and outlet *4* collectors are gas distribution devices and provide communication between the working chambers and the gas supply and discharge devices. The heat exchanger *5* for heat dissipation is a group of similar elements in the form of cylindrical U-shaped pipes connected to the inlet and outlet manifolds. The picture shows one of the pipes. In the simplest case, the engine is described using four working chambers *6–9*. The processes in the engine are cyclic, determined by the frequency of rotation of the shaft. During one period, the working chambers make one counterclockwise rotation, while the volume of each chamber changes with twice the frequency. At the selected time, as shown in the figure, the chambers *6* and *8* have a minimum volume, and the chambers *7* and *9* have a maximum volume. If we trace the movement of, for example, the chamber *8* in one period in space, then when turning counterclockwise, it will consistently switch with the input cavity, after which it will occupy the position of the chamber *9*. Next, the chamber will be switched with the inlet collector *10* of the heat exchanger. After half a period, the chamber will have a minimum volume and occupy the chamber position *6*. At subsequent points in time, the chamber will be connected to the outlet manifold *11* of the heat exchanger. The process of gas expansion in the chamber will end when it is in the chamber position *7*. The end of the cycle will be accompanied by its switching with the output cavity and returning to the state at the beginning of the cycle.

To simplify the determination of the characteristics of the refrigerating machine, a simulation was performed in a hybrid $2D - 3D$ setting. The essence of this setting of the problem is as follows. The state of the gas in the U-shaped tube of the heat exchanger is determined within the framework of solving a three-dimensional problem with the natural formulation of correct boundary conditions at the inlet and outlet of the pipe, reflecting the essence of the non-stationary process. The remaining machine objects are described without taking into account some „edge effects“. This approach is possible due to the fact that the heat exchanger consists of the same type of elements in the form of cylindrical pipes, therefore, it is sufficient to evaluate the processes occurring for one pipe. Accordingly, it is possible to trace the „current tube“ passing through the selected tube of the heat exchanger in other parts of the structure under consideration: the collectors of the heat exchanger, the working chambers of the engine, the inlet and outlet cavities. These parts of the structure are represented as a layer of variable thickness (the layer thickness changes in the collector area, in the rest of the listed elements it is constant) with symmetry conditions that exclude friction, mass and heat transfer through the lateral boundaries of the layer. The number of layers corresponds to the number of pipes in the heat exchanger, and the characteristics of the refrigerating machine are determined in specific quantities per pipe of the heat exchanger. Assuming that $50 - 100$ heat exchange tubes will be used in the refrigerating machine, the friction and heat exchange losses from the interaction of gas with the walls of the working chambers, which will occur in the first and last layers, will lead to a correction of the overall assessment of the processes occurring by no more than 1 %–2 %.

The U-shaped heat exchanger tube consists of two rectilinear sections with a length of $L = 0.5$ m, connected by a elbow with a 180 degree rotation of the heat exchanger°. Radius of curvature of the midline of the elbow $R = 0.04$ m. Pipe diameter $D = 0.02$ m.

The considered sections of the inlet and outlet manifolds per one tube of the heat exchanger have a prismatic hexagonal shape. The face in contact with the heat exchanger tube is flat, its dimensions are $d_1 = 0.04$ m, $d_2 = 0.05$ m. The opposite face is a section of the lateral surface of a cylinder with radius $R_2 = 0.125$ m. The length of the surface in the angular direction $\beta = 16^\circ$, in the direction of the axis $z$ of the cylinder $w_b = 0.01$ m. The other four faces are flat.

The engine has four vanes that divide the space between the body and the rotor into four chambers. The opposite vanes are connected in pairs and move according to the same law. The angular size of the vane is $\psi_b = 52^\circ$. The movement mechanism of the vanes ensures convergence of their midlines at an angle of $\psi_{\min} = 54.1^\circ$, with the maximum angle being $\psi_{\max} = 125.9^\circ$. The length of the vane in the direction $z$ per tube of the heat exchanger, $w_b$; radius of the rim of the housing $R_2 = 0.125$ m; radius of the rotor $R_1 = 0.05$ m.

The vanes move kinematically in pairs, governed by two angular velocities

$$\omega_b = \omega(1 \mp (\frac{\pi}{2} - \psi_{\min}) \sin(2\varphi)),$$

where $\omega$ is the angular velocity of the rotor,$\varphi$ is the angular coordinate of the position of the midline of the vane [15]. The position of the vane faces $\varphi_{bj}$ can be found from the solution of an ordinary differential equation

$$\frac{d\varphi_{bj}}{dt} = \omega_{bj}.$$

The inlet and outlet cavities have the shape of a ring segment with an attached nozzle. The length of the cavity at a radius of $R_2$ with an angular coordinate of 51.95° (the size of the intake and exhaust windows). The ring size is $H = 0.015$ m in the radial direction. The gas inlet and outlet sections of the rotary vane machine are located at a radial distance of $R_3 = 0.25$ m with an extension along the angular coordinate of $\phi = 10^\circ$. The size of the cavities according to „layer thickness“ $w_b$ coincides with the corresponding size of the working chambers in this coordinate direction.

Air is the working gas. The state of the gas is determined by the thermodynamic Mendeleev-Clapeyron equation and the caloric equation for a perfect gas. Gas inlet and outlet pressure is 0.1 MPa, gas inlet temperature is 300 K. The temperature at the output boundary, in case of backflow, is set to 231 K. This value corresponds to the average outlet gas temperature and is determined as a result of a preliminary calculation. The wall temperature of the U-shaped pipe is 400 K. At the remaining boundaries, the thermal insulation condition is met. The angular velocity of rotation of the rotor $\omega = 10\pi$ rad/s.

The regime of gas flow and heat exchange in a rotary-vane gas cooler is determined within the framework of the Reynolds-averaged system of Navier-Stokes equations. A Realizable $k - \varepsilon$-model of turbulence was used to determine the characteristics of turbulence. The authors of Ref. [19] performed numerical simulation of a Stirling engine with tubular heat exchangers. Different turbulence models were used in these calculations. A comparison of the integral characteristics obtained in the calculations with the experimental results showed that the most appropriate option is to use a Realizable $k - \varepsilon$-model of turbulence.

## 2. Algorithm for solving the problem

The solution of the problem is obtained as a result of numerical integration of a system of unsteady Reynols averaged Navier Stokes equations (URANS). The ANSYS Fluent 2021 R2 software package was used for this purpose, which made it possible to consider a computational domain with a complex topology and its shape changing over time, as well as with the possibility of creating and destroying switching connections between its elements and taking into account the balances of mass, energy and amount of motion. The discretization of the computational domain was carried out using the finite volume method using hexagonal elements. In the main pipe, collectors, and inlet/outlet cavities, the grid was non-tunable. A grid with a variable number of elements used was used to describe the behavior of function values in working chambers. The dynamic layering algorithm was used when adding or removing sampling elements is associated with moving the boundary of the computational domain. The method consists in combining the cells on the border of the area with the cells of the next row if the controlled cell size becomes smaller than the set value. If this parameter exceeds the set value, the cell is divided in the selected ratio.

The interface procedure is used to describe the interaction of the working chambers with the collectors or gas inlet and outlet cavities. When various elements of the design area came into contact, the interface section in the contact zone was permeable to the working fluid. In all other situations, the interface implements the boundary condition of an adiabatic solid wall.

The Coupled scheme (implicit approximation of all thermodynamic functions) and counterflow approximations of flows at the boundaries of control volumes with the second order of accuracy are used for numerical integration. The number of sampling elements at the beginning of each cycle was about $10^5$, which ensured the determination of the total value of the heat flow on the wall of the tube of the heat exchanger with an error not exceeding 1.5%. In connection with the reconstruction of the computational grid, the time integration step was chosen to be $10^{-4}$ s.

## 3. Analysis of the results

The intermittent nature of the mass supply to the heat exchanger of the rotary-vane machine leads to an unstable gas flow regime. Let's consider some key points.

The unsteady mode of gas flow in a rotary-vane machine is caused by the discreteness of its piston group. In the cyclic process, the gas flow rate in the inlet cavity is determined by the sequential switching of all four working chambers with it and is intermittent. An increase in the volume of the working chamber creates a reduced pressure in it, which leads to its filling with gas from the inlet cavity. Leakage occurs through a variable-size switching window. In the initial moments of the filling phase of the working

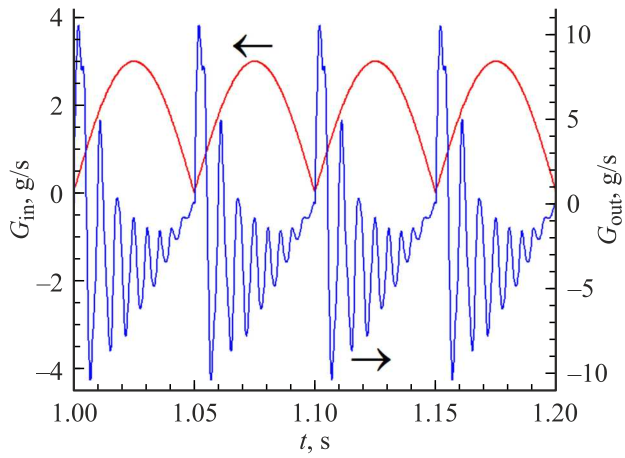


**Figure 2.** Dependence of gas flow rate on time in the inlet and outlet cavities in one cycle.

chamber, the length of the gas intake boundary corresponds to the size of the working chamber in angular coordinate, but in subsequent moments, when the working chamber „comes out of the shadow" of the inlet cavity, the size of the overlap decreases to zero. The combination of these factors (the size of the switching window and the rate of increase in the volume of the working chamber) leads to the fact that the dependence of gas consumption on time when gas enters the rotary vane machine has the form of a smooth periodic function.

When the gas flows out of the working chamber into the outlet cavity, high-frequency oscillations can be observed, which are superimposed on the main periodic signal. This is due to the fact that when the gas flows out of the working chamber, the pressure in it is lower than the pressure in the outlet cavity. When switching the working chamber and the outlet cavity, a gas flow occurs with pressure equalization in the working chamber and the outlet cavity. The process is characterized by damping pressure fluctuations and a change in the direction of gas flow between objects. The time dependence of the flow rate in the inlet and outlet cavities is shown in Fig. 2. Positive flow values correspond to the flow of gas into the cooler, negative values correspond to its expiration.

At a gas inlet temperature of 300 K, the average outlet temperature from the outlet cavity is 231 K. The dependence of the average mass temperature of the gas in the control section of the outlet cavity on time is shown in Fig. 3.

An idea of the gas flow regime in the heat exchanger tube is given in Fig. 4, where the flow rate in the inlet and outlet sections of the pipe is shown. The outflow of the working fluid from the tube corresponds to a negative flow rate, the inflow corresponds to a positive value. The connection of the working chambers to the heat exchanger is accompanied by a sharp change in flow rate over time. The most intense „upward " surges are caused by the unsteady supply of gas

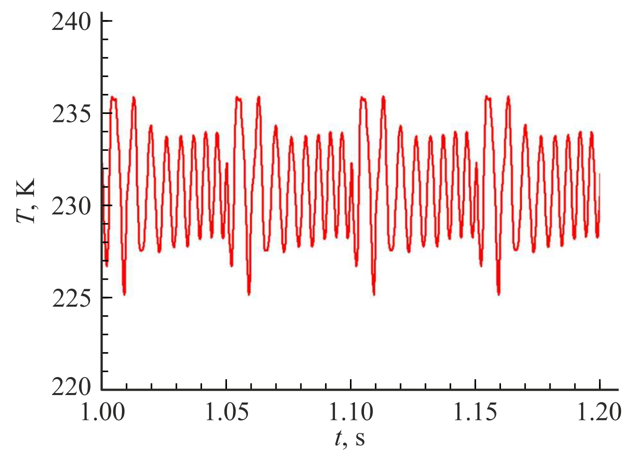


**Figure 3.** Change in the average mass temperature in the outlet cavity in one cycle.

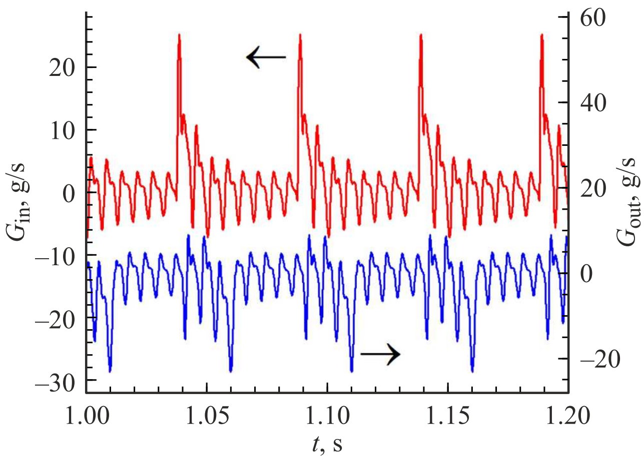


**Figure 4.** Gas flow through the end sections of the heat exchanger tube per cycle.

from the working chamber through the inlet manifold. At the same time, a compression wave is formed in the pipe, which, with a delay time of about 3 ms, is visible in the dependence of the gas flow rate at the other end of the pipe. The most intense „downward" surges are caused by unsteady outflow of gas from the heat exchanger through the outlet manifold with filling of the working chamber. In this case, the formation of gas flow in the outlet section of the pipe occurs under the influence of a dilution wave caused by an increase in the volume of the working chamber connected to the outlet collector of the heat exchanger. At all other points in time, the system of collectors and pipes of the heat exchanger is closed. In the heat exchanger tube, attenuation of gas fluctuations is observed with equalization of the pressure level between the inlet and outlet manifolds. The oscillation frequency is related to the geometry of the pipe, the size of the collectors and the mode of heat exchange on the pipe wall. A study of wave processes in a heat exchanger was conducted in proceedings of the conference [20].

The dependence of the average mass temperature value on time for the same pipe sections is shown in Fig. 5.

The average temperature of the incoming gas into the heat exchanger tube is 547 K. As can be judged by the dependence of the temperature value at the inlet to the heat exchanger, the manifestation of the non-stationary nature of the gas flow is observed not only in the flow component of the velocity component, but also in this flow characteristic. The average temperature of the gas flowing out of the pipe is 400 K. The fact that the temperature of the gas flowing out of the U-shaped pipe coincides with the temperature value, which is set as a boundary condition on the pipe wall, is more evidence of a suboptimal heat removal mode in the heat exchanger. At least, the outlet section of the pipe is not involved in the heat exchange of gas with the pipe wall.

Fig. 6 shows the change in the average mass temperature in the chamber over the period. At the beginning of the

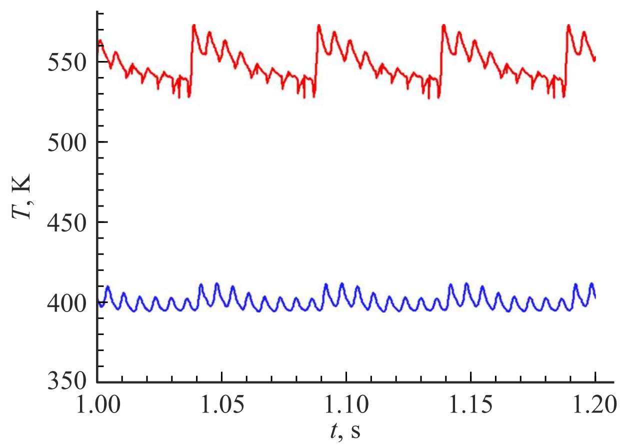


**Figure 5.** Values of the average mass temperature of the gas at the ends of the tube of the heat exchanger per cycle.

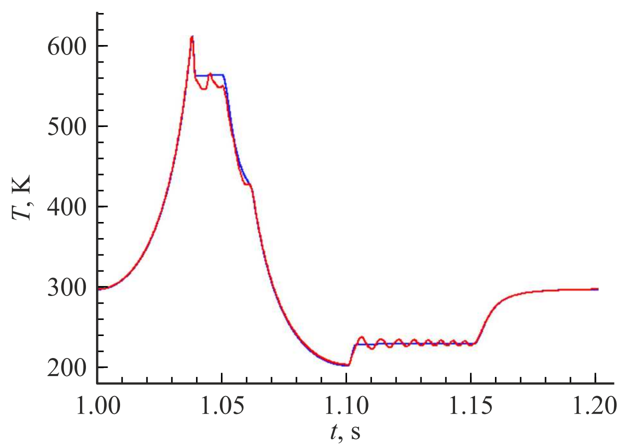


**Figure 6.** Change in the average mass temperature in the working chamber per cycle. The red curve is the result of numerical simulation in the Fluent package, the blue curve is the result of nodal analysis modeling.

cycle under consideration, the working chamber is in the chamber position *9* (Fig. 1); the average mass temperature of the gas in the chamber is equal to the temperature of the gas at the inlet to the cooler. The temperature increase from 300 to 600 K reflects the adiabatic process of gas compression (1–1.04 s). The process of pumping gas from the working chamber into the inlet manifold of the heat exchanger (1.04–1.05 s) is accompanied by stirring and occurs with a decrease in temperature. At the end of the process, the working chamber occupies the position of the chamber *6* and the temperature level in the chamber reflects the state of the gas at the „upper dead center". The subsequent decrease in gas temperature is caused by an increase in the size of the working chamber and its filling with gas from the outlet manifold of the heat exchanger (1.05–1.06 s). The final temperature in the chamber exceeds the temperature of the gas at the outlet of the pipe. This is due to the presence of a dead volume in the working chamber, in which gas remained that did not participate in the heat exchange [18].

In the process of adiabatic gas expansion (1.06–1.1 s), the average mass temperature of the gas in the working chamber decreases to 204 K. The working chamber occupies the position *7* („lower dead center"). The switching of the working chamber with the outlet manifold is accompanied by a rapid increase in temperature to the temperature at the outlet of the cooler. The temperature increase is associated with the pressure equalization between the working chamber and the outlet manifold. The injection of gas from the working chamber into the outlet manifold (1.1–1.15 s) is accompanied by a decaying wave process. The considered cycle of temperature change in the working chamber ends with its filling with gas from the inlet manifold (1.15–1.2 s).

The change in the temperature of the gas in the heat exchanger is due to the non-stationary heat exchange on the pipe wall. The dependence of the integral value of the heat flow on the wall of a U-shaped pipe on time is shown in Fig. 7 (red curve). With an average heat flow value

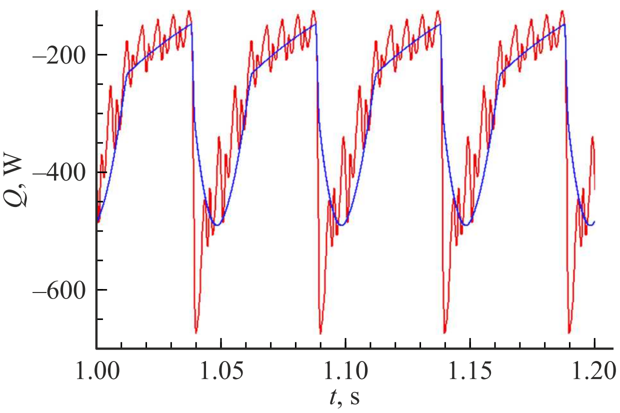


**Figure 7.** Values of the heat flow on the pipe wall in one cycle, obtained as a result of numerical modeling in the Fluent package (red curve); heat dissipation in the heat exchanger in one cycle, calculated by nodal analysis (blue curve).

of 288 W (modulo), the amplitudes of deviation from the average value are 160 and 386 W in a smaller and larger direction. Attention should be paid to the correlation of the heat flow with the gas flow rate in the inlet section of the heat exchanger tube (Fig. 4). The maximum (modulo) value of the heat flow is localized by the initial section of the tube of the heat exchanger and is caused by an intensive change in the flow rate at the inlet to the pipe when flowing out of the collector of the heat exchanger.

Considering the heat balance in the heat exchanger, the average value of the heat flow through the pipe wall per cycle can be correlated with a change in the internal enthalpy of the gas mass, which, in turn, is associated with a change in gas temperature. Based on the value of the average gas flow rate in the pipe over the period of 1.92 g/s, the decrease in gas temperature in the pipe is estimated at 147 K, and the specific heat capacity of the gas at constant pressure is 1006 J/(kg·K), then the change in internal enthalpy will be 284 W, which indicates a satisfactory energy balance in the heat exchanger at the level of cycle average values of thermodynamic functions.

According to the First Law of Thermodynamics, the work performed on the gas per cycle in the combined compressor and expander operation is determined by the difference between the heat removed and the change in internal energy at the device's inlet and outlet. If these estimates are performed, the work performed will be about 192 W.

To verify the results obtained in this study, a refrigeration machine was calculated using the nodal analysis method [17]. This method does not take into account the shape of the heat exchanger, the heat dissipation in the heat exchanger is determined based on the volumetric heat transfer coefficient, the value of which is assigned. Therefore, to compare the results, the volumetric heat transfer coefficient was set so that the average heat dissipation capacity in the heat exchanger per cycle, calculated using the nodal function method, was equal to the average heat flow per cycle on the pipe wall. The volume of the heat exchanger in the nodal analysis method was set equal to the volume of the pipe together with the adjacent collectors. The average heat flow (288 W) obtained in numerical simulation corresponds to the volumetric heat transfer coefficient equal to 100 KW/($m^3$·K).

Modeling of the heat exchanger by various methods leads to close dependences of total heat generation (Fig. 7). The values of the average mass temperature (Fig. 6) and pressure (Fig. 8) in the working chamber, obtained by different approaches, are almost identical for most of the working cycle due to the scale of the change in values. Discrepancies are observed at the moments of contact of the chamber with the collectors of the heat exchanger and with the outlet collector, when the manifestation of the elastic properties of the gas has a significant effect on the state of the working fluid in the chamber. The wave processes that cause characteristic fluctuations in numerical modeling cannot be reproduced in principle using the method of nodal function values.

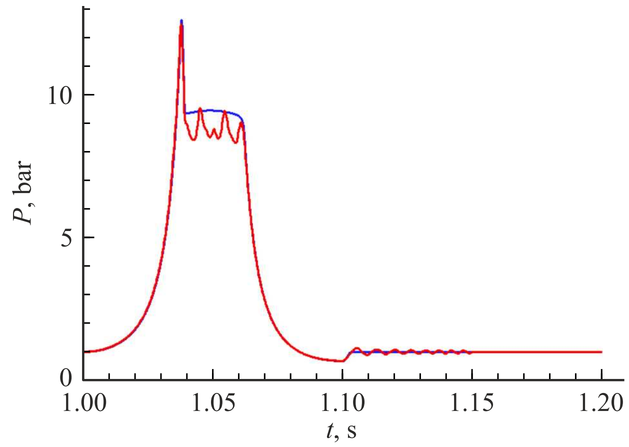


**Figure 8.** Change in the average mass pressure in the working chamber per cycle. The result of numerical simulation in the Fluent package is shown by the red curve, the result of nodal analysis modeling is shown by the blue curve.

## Conclusion

A fundamentally new method of using a rotary-vane machine has been proposed in this paper, which has not previously been described in the scientific literature. The possibility of creating a gas cooler based on a rotary-vane machine has been established by mathematical modeling. The values of the specific values (based on a typical element of the heat exchanger) of the gas dynamic functions in the working chambers and the heat exchanger are determined. In particular, for the considered design and operating mode, 192 W of the energy expended on the shaft of the rotary-vane machine accounts for 94 W of the decrease in the internal energy of the working fluid in the form of air. Under normal conditions and a gas flow rate of 1.92 g/s, the air temperature change will be 69 °C. The results were verified by comparison with the nodal analysis method. A qualitative correspondence has been obtained between the changes in thermodynamic functions in the working chambers, which confirms the correctness of the numerical simulation. It should be noted that the non-stationary mode of gas flow in the heat exchanger, which is not associated with the compression-expansion process, opens up wide opportunities for intensifying the heat removal process. The results obtained expand the scope of rotary vane machines and open up prospects for their use in refrigeration technology.

### Conflict of interest

The authors declare that they have no conflict of interest.

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