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MODELING A STAGE OF A MULTISTAGE CENTRIFUGAL COMPRESSOR: THE BLADES' THICKNESS EFFECT OF AN IMPELLER AND A DIFFUSER

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Abstract: *The article analyzes the design features and general modeling of the centrifugal compressor stages. Based on the analysis, the main tasks for modeling a centrifugal compressor designed to operate in the subsonic range of gas velocities are formulated. The aim of the work is to develop a model of a centrifugal compressor based on the proposed method of thermogasdynamic calculation with an extended determination of the number of the thermogasdynamic and design parameters of the centrifugal compressor in each of its stages, as well as to determine the optimal thickness of the impeller and diffuser blades.*

Key words: *simulation, stage, thermogasdynamics, centrifugal compressor, impeller, diffuser, methodology, design parameters.*

1. INTRODUCTION

Centrifugal compressors are one of the varieties of the group of vane compressors, and are energy machines in which the medium is compressed using centrifugal forces.

Turbine internal components are increasingly being upgraded to improve efficiency, increase the pushing mass ratio of the working medium, increase operational reliability and extend service life. Mass dimensions of modern turbines are reduced due to the design of new working wheels and diffusers, and new turbines have a smaller volume and lighter weight compared to the previous ones.

Compared to other types of power equipment, the design of centrifugal compressors is complicated by the presence of impellers and diffusers with blades having a curved profile. The manufacture of high-performance impellers and diffusers as a whole is a complex of high-precision technical operations.

The main element of this equipment is a rotor equipped by a shaft with symmetrical impellers. The inertial force acts on the gas particles, which arises due to the presence of a rotational motion made by the wheel blades during the operation of the equipment. The gas moves from the compressor's center to the impeller's edge. It is known from flow thermodynamics that when the flow accelerates, the gas pressure decreases.

Fig. 1 shows a diagram of a blade outlet with an annular cavity. The impeller blades have an expanding shape, and this leads to an increase in pressure at rotation with angular speed ω. The combined effect of these two influences leads to an increase in pressure, so that at the outlet of the impeller the absolute gas velocity becomes maximum, and its pressure increases. Further, the gas enters a stationary vaned diffuser, where it is decelerated with a corresponding increase in pressure. Then the flow in the reverse guide vanes passes through the inter-blade channels with an increasing cross-section and enters the next stage.

In this case, the width of the channels decreases according to design constraints, but at the same time the cross section of the channel increases. After being compressed in the previous stage, the gas is directed to the suction chamber of the next compressor stage [1]. At the last stage, immediately behind the blade diffuser, a spiral chamber (a prefabricated volute) is

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installed, from where the gas is supplied to the final cooler and then to the consumer [2].

The design of different blade shaped impellers for different productivity is presented in the literature [3-6]. The most significant and critical part of the centrifugal compressor is the impeller. A great attention is paid to the design of the impellers: the upgrading and optimization of their operating and geometric parameters, CFD-modeling and gas-dynamic calculation are carried out in works [7-10].

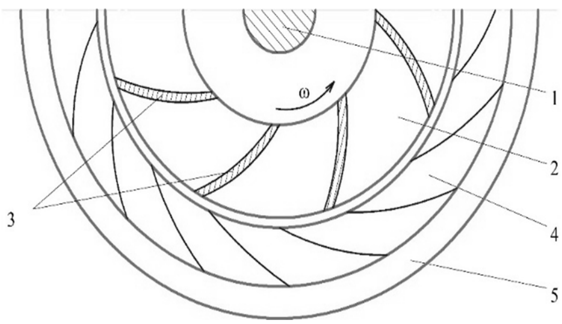


Fig. 1. Centrifugal compressor stage design: a) Schematic of a blade output with an annular cavity of centrifugal compressor: 1 – a shaft; 2 – an impeller; 3 – the impeller blades; 4 – a blade outlet (a vaned diffuser); 5 – an annular cavity; b) Compressor impeller velocity components.

There are various methods to the calculation of the main parameters of the centrifugal compressor. Computational fluid dynamics (CFD) methods and fluid structure interaction (FSI) modeling are the newest [3, 11]. Those computational tools make it possible to determine and visualize the studied physical parameters in various forms [12]. Let's consider specific works on modeling and research of the centrifugal compressor. In the paper [13] the authors presented a model for CFD analysis of turbo machinery, known as "frozen-rotor" model, that only yields satisfying results for efficiency and pressure ratio at and near the point of best efficiency. For this case, the static pressure shows a nearly uniform circumferential distribution at the inlet of the diffuser, which numerically leads to more homogeneous flow rates through the single vane channels, and thus to a more realistic time averaged flow distribution.

During the pre-design stage the compressor's parameters can be determined using various methods [14, 15]. They depend on the geometric structure of the individual parts and the calculation accuracy limits [16, 17]. The aim of the work is to develop a centrifugal compressor's model based on the proposed method of thermogasdynamic calculation with an extended determination of the number of thermogasdynamic and design parameters in each of its stages taking into account the change in the thickness of the blades of the impeller and the diffuser.

2. CALCULATION OF THE CENTRIFUGAL COMPRESSOR WITH THE DETERMINATION OF THE SECTIONS OF THE DIFFUSER AND THE IMPELLER CHANNELS

A method of approximate calculation is considered here, which gives a general idea of the geometric dimensions of a stationary-type compressor stage operating at subsonic gas speeds.

The thermodynamic and gas-dynamic calculation of the centrifugal compressor stage consists of determining the parameters of the impeller and the diffuser. It is performed at subsonic gas velocities for an inviscid working medium, since the movement of real gases in the main part of the flow approximately obeys the laws of the movement of an inviscid fluid.

The impeller is the only machine element where mechanical energy is converted into pressure energy and kinetic. Assuming the inlet to the rotor blades is radial, we have the equations for the dependence of the outlet pressure p_2 on the inlet pressure p_1:

$$p_2 = p_1\left[1+\frac{\eta_a}{2c_pT_1}\left(c_1^2-c_2^2+2u_2c_{2u}\right)\right]^{\frac{k}{k-1}}, \quad (1)$$

where η_a – isentropic efficiency of stationary the centrifugal compressors is in the range η_a = 0,80-0,90; c_p is the heat capacity of the gas at constant pressure; k – adiabatic exponent; c_1, c_2 – velocity components, u_2 – circumferential velocity; c_{2u} – actual value of the tangential component of the absolute velocity at the outlet from the impeller.

Determination of the number of impeller stages is held according to

$$Z = \frac{ln\left(\frac{\varepsilon}{\varphi}\right)}{ln(\varepsilon_1)}, \quad (2)$$

where ε – pressure ratio in the centrifugal compressor, that calculated as

$$\varepsilon = \frac{p_2}{p_1}, \quad (3)$$

φ = 0,95 – coefficient taking into account pressure losses because of internal friction and leaks, ε_1 – the degree of pressure increases in one stage. Usually this value ranges from 1,4 to 2,2.

A determined number of impeller stages is approximated to an integer. The ε_1 value will be specified using the integer value of the stages

$$\varepsilon_1 = \left(\frac{\varepsilon}{\varphi}\right)^{\frac{1}{Z}}. \quad (4)$$

The impeller outer diameter D_2 is calculated as

$$D_2 = \frac{60u_2}{\pi n}. \quad (5)$$

The circumferential velocity u_2 is taken from 150 to 250 m/s, the number of revolutions n is 3000 rpm. Respectively values of n = 3000 rpm and u_2 = 190 m/s.

The ratio of the inlet and outlet diameters is selected to be approximately 0.5. Deviations from 0.48 to 0.60 are possible, D_1/D_2 = 0.5-0.6.

Determination of the cross-sectional area of the inlet and outlet channels of the impeller blades.

First, let's define the pitch r_{d1} by the inner diameter

$$r_{d1} = \frac{\pi D_1}{Z_L}. \quad (6)$$

where Z_L = 26 – the number of impeller blades (usually 16-32).

Then we take the blade thickness δ = 0.01m and find the length of the inlet cross-section

$$l_1 = r_{d1} - \delta. \quad (7)$$

From [14] it is known that b_1 is related to D_1 as follows: b_1/D_1 = 0,05-0,1. Accepting $b_1 = 0{,}06 \cdot D_1$. And finally, the cross-sectional area of the inlet channel by the inner diameter is determined

$$F_1 = l_1 b_1. \quad (8)$$

Similarly for the outer diameter:

$$r_{d2} = \frac{\pi D_2}{Z_L}. \quad (9)$$

$$l_2 = r_{d2} - \delta. \quad (10)$$

Since b_1/D_1 = 0,01-0,02, we calculate $b_2 = 0{,}012 D_2$ and

$$F_2 = l_2 b_2. \quad (11)$$

Determination of the components of rotation velocities in impeller and thermal indicators of gas.

First, let's find the parameters at the inlet to the impeller. Find the circumferential velocity u_1

$$u_1 = \frac{\pi D_1 n}{60}. \quad (12)$$

The impeller inlet air velocity is found by the following equation

$$\omega_1 = \frac{Q}{Z_L F_1}. \quad (13)$$

Taking angles for relative velocities β_1 = 25° and β_2 = 61° (usually β_1 = 20-40°, β_2 = 60-65°) the absolute velocity c_1 is found

$$c_1 = \sqrt{\omega_1^2 + u_1^2 - 2\omega_1 u_1 \cos\beta_1}. \quad (14)$$

Then find the projection of C_1 onto U:

$$c_{1u} = c_1 \cos\left(90^o - \beta_1\right). \quad (15)$$

The parameters at the outlet from the impeller are obtained by carrying out similar calculations:

$$\omega_2 = \frac{Q}{Z_L F_2}; \quad (16)$$

$$c_2 = \sqrt{\omega_2^2 + u_2^2 - 2\omega_2 u_2 \cos\beta_2}; \quad (17)$$

$$c_{2u} = c_2 \cos\left(90^o - \beta_2\right). \quad (18)$$

After that, let's find air density. The simplest theory of compressor machines with practically acceptable accuracy is based on the thermodynamics of an ideal gas, subject to the equation

$$\rho = \frac{p_1}{RT_1}, \quad (19)$$

where R is individual gas constant of dry air.

Determination of the basic dimensions of the diffuser is listed below

$$D_3 = 1{,}1D_2\,, \tag{20}$$

$$D_4 = \left(1{,}3-1{,}55\right)D_2\,, \tag{21}$$

where D_2 is the outer diameter of the impeller.

Determination of pressure and temperature parameters in the diffuser.

Let us find the pressure at the exit from the first stage p_2 and at the exit from the impeller p_l:

$$\varepsilon = \frac{p_2}{p_1} \Rightarrow p_2 = \varepsilon p_1\,, \tag{22}$$

$$p_l = p_1\left(1+\frac{1}{2c_pT_1}\left[c_1^2-c_2^2+2\left(u_2c_{2u}-u_1c_{1u}\right)\right]\right)^{\frac{n}{n-1}} \tag{23}$$

where c_p is the heat capacity of air (c_p = 1001); c_1 and c_2 are absolute velocities in the impeller; u_2 and u_1 – circumferential velocities at the outlet and inlet to the impeller; c_{2u} and c_{1u} – projections of absolute velocities on circumferential velocities; n is the compression polytropic index (n = 1,38).

Air temperature at the outlet of the impeller and at the inlet of the blade diffuser can be find from the formula:

$$t_l = t_1\left(\frac{p_l}{p_1}\right)^{\frac{n-1}{n}}\,. \tag{23}$$

Using the same formula, we define temperature t_2 at the outlet of the blade diffuser.

Let's calculate the air density by

$$\rho_l = \left(\frac{p_l}{RT_l}\right), \tag{24}$$

and find air density ρ_2 at the outlet of the blade diffuser.

Determination of the cross-sectional area of the inlet and outlet channels.

We define a step at the beginning determination the length l_3 and l_4 as:

$$r_{d3} = \frac{\pi D_3}{Z_D}\,, \tag{25}$$

where Z_D = 22 – the number of the diffuser's blades (usually 20-28).

$$r_{d4} = \frac{\pi D_4}{Z_D}\,. \tag{26}$$

Using the obtained values, we determine the length:

$$l_3 = r_{d3} - \delta\,, \tag{27}$$

$$l_4 = r_{d4} - \delta\,, \tag{28}$$

where δ – blade thickness (0.01 m).

Channel heights h_1 and h_2 are expressed from the formula $Q = c\cdot l\cdot h\cdot\rho$. So, we get

$$h_1 = \frac{Q}{c_2 l_3 \rho_2}\,, \tag{29}$$

$$h_2 = \frac{Q}{c_4 l_4 \rho_3}\,. \tag{30}$$

Also, dimension h can be accepted based on practical data $h_1 = h_2 = (1\text{-}1{,}2)b_2$.

The area F_3 is calculated as

$$F_3 = l_3h_1. \tag{31}$$

Let us find velocity c_4 at the diffuser's outlet from the relation:

$$c_4=0{,}4c_2. \tag{32}$$

As a result, using the previously obtained values, we find the area:

$$F_4=l_4h_2. \tag{33}$$

The obtained empirical dependences of the thermodynamic and gas-dynamic calculation of the centrifugal compressor are considered to be quite massive for multistage the centrifugal compressors, therefore, in order to obtain intermediate values of the parameters in each stage, it is necessary to consider the characteristic points of the compressor working.

3. CALCULATION OF THE PARAMETERS OF THE CHARACTERISTIC POINTS

Figure 2 shows a diagram of a multistage centrifugal compressor (sectional side view). Next, the thermogasdynamic parameters (pressure, temperature, density, mass and volume) of the characteristic points of the compressor A, B, C, D, E, F, G, H in Fig. 2 are calculated. Accordingly, it is assumed that points A, C, E, G indicate the index of the input parameters of the stage centrifugal compressor, and points B, D, F, H - on the index of output parameters.

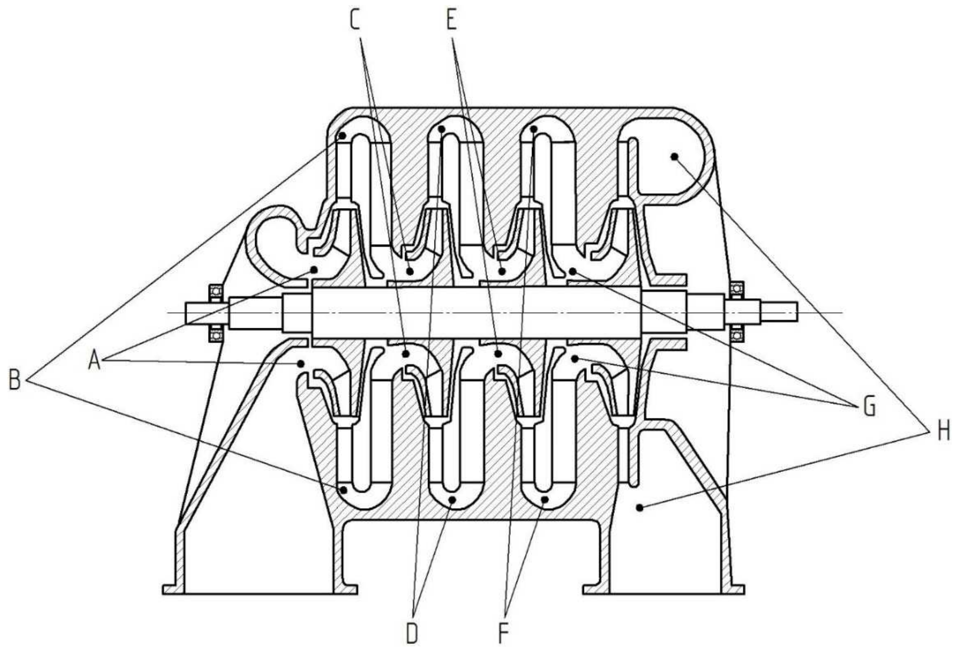


Fig. 2. Scheme of a multistage centrifugal compressor (sectional side view). A, B, C, D, E, F, G, H – characteristic points of the compressor operation, among which A and B refer to the first stage, C and D – to the second stage, G and H – for the i-th stage.

The pressure ratios *W* are added to determine the operating pressures *P* at the inlet and the outlet of the each compressor stage. Moreover, the pressures of the individual stages for the compressor are denoted as P_1, P_2 ... P_i, where the inlet pressure of the first stage is taken as the pressure P_1, and the pressure P_2 is the pressure at the exit from the first stage and the pressure at the inlet to the second stage, subsequent pressures are determined similarly.

$$\frac{P_2}{P_1} = \frac{P_i}{P_{i-1}} = W \,. \tag{34}$$

Therefore, the following formulas are used to find pressures:

$$P_i = W \cdot P_{i-1} \,. \tag{35}$$

The following formulas are used to calculate the inlet and outlet temperatures of each of the compressor stages:

$$T_i = T_{i-1} \cdot \left(\frac{P_i}{P_{i-1}} \right)^{\frac{n-1}{n}} . \tag{36}$$

In the case of intermediate coolers, the temperatures of some stages may correspond to the initial temperature.

The gas density in each compressor stage is calculated taking into account the pressure and temperature as follows:

$$\rho_i = \frac{P_i}{RT_i} \,. \tag{37}$$

To find the volume of gas in each stage of the centrifugal compressor, first, the mass of the gas is determined by the following dependence

$$m_i = \frac{Q \cdot \rho_i \cdot 60}{n} \,, \tag{38}$$

where *n* is the number of revolutions of the compressor shaft, *Q* is the volumetric productivity of gas suction, ρ is the density of gas (air).

Accordingly, the following volumes are obtained

$$V_i = \frac{m_i \cdot R \cdot T_i}{P_i} \,. \tag{39}$$

Thus, we have obtained an extended set of thermogasdynamic parameters of the centrifugal compressor in all its stages.

4. RESULTS AND DISCUSSION

The gas dynamics and thermodynamic parameters are calculated using computer means for the following specified parameters of the centrifugal compressor. Density of air ρ = 1,07 kg/m^3, individual gas constant R = 287 (J/kgK), adiabatic exponent k = 1,38, stage volumetric flow Q = 42 m^3/min, gas temperature at inlet T_1 = 293 K, pressure of outlet p_2 = 0,41 MPa, pressure of inlet p_1 = 0,09 MPa. Geometric dependencies for the impeller in dimensionless form - D_1 = 0,52D_2, for the diffuser - D_3 = 0,76D_4. The impeller blades exert a force effect on the gas flow around them. The conditions for gas entry into the blades should be considered to determine the optimal shape of the blades and their flow section. So, the first stage of the compressor was simulated changing the blades thickness (from 1 to 2 mm) of the

diffuser and the impeller. The characteristic dependences of the thickness of the blades on the cross-sectional area of the impeller and the diffuser channels taking into account the supply to the compressor stage are shown in Fig. 3.

Velocity components for different flow and blade thickness are summarized in Tables 1-3. The parameters of the characteristic points of the compressor operation at a supply of 42 m^3/min are presented in the Table 4, where can be clearly seen the ratio of the presented parameters depending on the compressor stage number. In this case, the absolute velocity c_1 when entering the channel at any point of the circle diameter D_1 is directed along the radius.

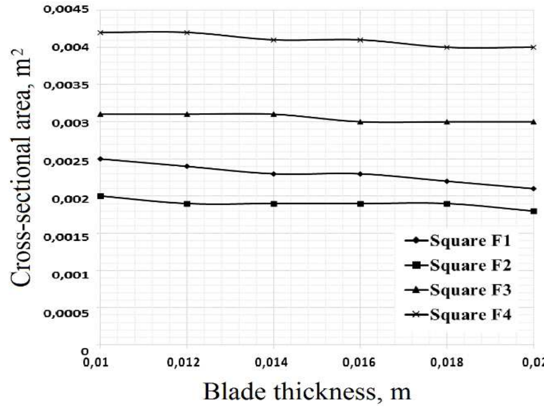


Fig. 3. Dependences of the area channel on the thickness of the blades: F1, F2 – cross-sectional areas of the inlet and outlet channels of the impeller, respectively; F3, F4 – cross-sectional areas of the inlet and outlet channels of the diffuser, respectively.

Let's decompose the velocity c_1 into components - portable u_1 and relative ω_1. The velocity is directed at a certain angle β_1 in the direction opposite to rotation. Hence it follows that if the leading edges of the blade were directed along the radius, then the gas flow could enter the working channel only with some impact. Small losses at the entrance to the channels of the impeller can be obtained only if the leading edges of the rotor blades are directed at an angle close to or equal to the angle between the directions of the relative velocity ω_1 and the portable velocity u_1.

Table 1

Velocity components of the compressor stage for 18 m^3/min gas flowrate.

Blade thickness, m	ω_1, m/s	ω_2, m/s	u_1, m/s	u_2, m/s	c_1, m/s	c_2, m/s	c_4, m/s	c_{2u}, m/s
0,01	4,63	5,83	98,8	190	94,62	187,24	74,9	163,76
0,012	4,77	5,92	98,8	190	94,49	187,199	74,8796	163,7279
0,014	4,93	6,02	98,8	190	94,3537	187,1577	74,86	163,6918
0,016	5,09	6,1077	98,8	190	94,2063	187,1115	74,85	163,65
0,018	5,27	6,2	98,8	190	94,0488	187,07	74,82	163,61
0,02	5,45	6,3	98,8	190	93,88	187,0263	74,81	163,57

Table 2

Velocity components of the compressor stage for 42 m^3/min gas flowrate.

Blade thickness, m	ω_1, m/s	ω_2, m/s	u_1, m/s	u_2, m/s	c_1, m/s	c_2, m/s	c_4, m/s	c_{2u}, m/s
0,01	10,77	13,46	98,8	190	89,16	183,85	73,54	160,8
0,012	11,147	13,83	98,8	190	88,82	183,69	73,47	160,66
0,014	11,51	14,0356	98,8	190	88,5052	183,6063	73,4425	160,5857
0,016	11,89	14,25	98,8	190	88,1672	183,51	73,4	160,5

0,018	12,3001	14,47	98,8	190	87,8	183,42	73,37	160,42
0,02	12,73	14,7	98,8	190	87,4	183,32	73,33	160,33

Table 3

Velocity components of the compressor stage for 66 m^3/min gas flowrate.

Blade thickness, m	ω_1, m/s	ω_2, m/s	u_1, m/s	u_2, m/s	c_1, m/s	c_2, m/s	c_4, m/s	c_{2u}, m/s
0,01	16,98	21,4	98,8	190	83,71	180,6	72,24	157,95
0,012	17,51	21,72	98,8	190	83,25	180,47	72,18	157,84
0,014	18,0817	22,0559	98,8	190	82,76	180,34	72,13	157,73
0,016	18,68	22,39	98,8	190	82,24	180,21	72,08	157,61
0,018	19,32	22,74	98,8	190	81,69	180,07	72,03	157,49
0,02	20,01	23,1	98,8	190	81,09	179,93	71,97	157,37

Table 4

The parameters of the characteristic points of the compressor operation at a supply of 42 m^3/min

№	P, MPa	T, K	ρ, kg/m^3	m, kg	V, m^3
I stage	0,133	328,27	1,41	0,0197	0,0139
II stage	0,197	365,72	1,87	0,0262	0,0139
III stage	0,292	407,52	2,497	0,035	0,014
IV stage	0,43	453,29	3,31	0,0463	0,014

Finally, let's consider the kinematics of the gas at the exit from the impeller. With a sufficiently large number of blades the relative velocity ω_2 has a direction close to the tangent to the trailing edge of the blade. The relative velocity is determined by the performance and the size of the impeller. In most cases, it is somewhat less than the velocity ω_1. The absolute velocity at the exit from the blades c_2 is obtained as the geometric sum of the relative and portable velocities. It is easy to see that c_2 is significantly greater than the velocity c_1 at the entrance to the impeller, since the portable velocity u_2 is approximately two times greater than u_1.

Hence it follows that when the gas passes through the impeller, the kinetic energy of the compressible medium increases. A decrease in the velocity c_2 and the conversion of the velocity energy into pressure energy is achieved in the diffusers with the least possible loss. Accordingly, from the considered tables 2 - 4, it can be seen how the thickness of the blades of the impeller and the diffuser affects the velocity indicators. With thickening of the blades a decrease in certain composite velocities is observed, which directly leads to a change in pressure indicators. Taking into account the satisfaction of weight and size indicators, the thickness of the blades was chosen in the amount of 0.016 m, which provides the necessary speed indicators at different feed rates.

5. VERIFICATION OF THE CALCULATION OF THE COMPRESSOR'S STAGE BY THE FINITE ELEMENT METHOD

The modern approach to designing the flow path of the centrifugal compressor includes mandatory numerical modeling and analysis of existing variants of turbulence models with the possibility of using a grid model with a finer grid and iteration step.

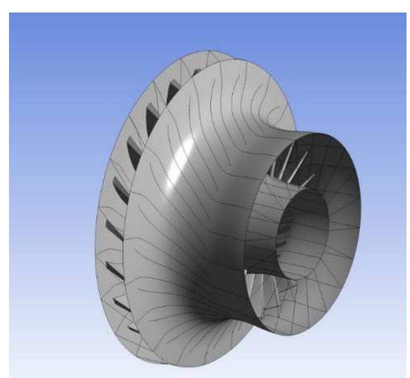
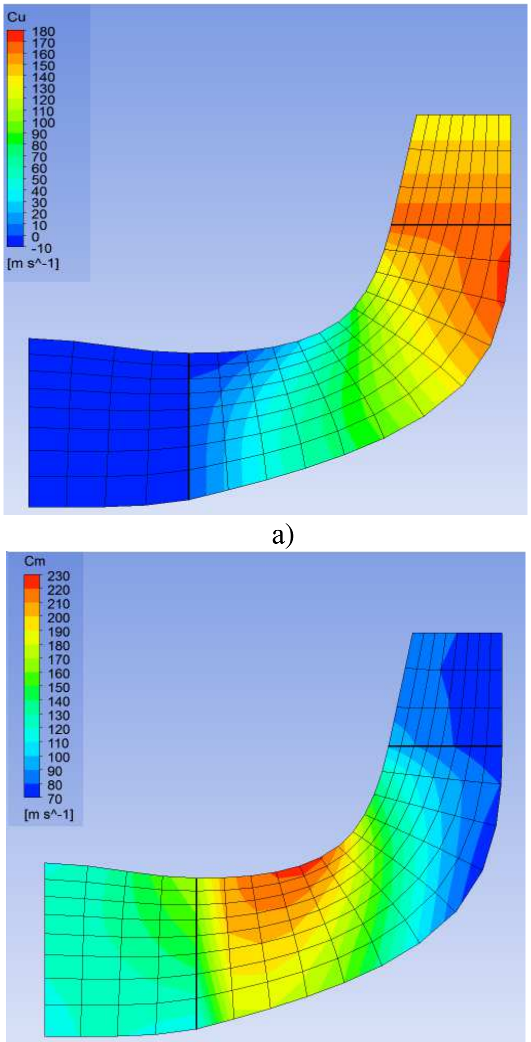


Fig. 4. Impeller simulation model with 20 vanes.

The finite element method is particularly effective in calculating the flow rate of the centrifugal compressor. The decision was made to numerically model the aerodynamics of the centrifugal compressor using the lean ANSYS system that is the leading software for modeling and research of two-dimensional and three-dimensional models of engines, turbines, transformers and other devices for various applications.



a)

b)

Fig. 5. Inter-vane channels of the impeller: distribution of C_2 (a) and C_m (b) velocities along the channel.

To verify and confirm the previously obtained data on the layout of the flow part of the centrifugal compressor and the accepted thickness of the blades, as well as to identify possible "separation" zones, a numerical simulation of the centrifugal compressor stage was carried out (Fig. 4). To build the model the necessary calculations were performed, a computational grid was created, boundary and initial conditions were set, etc.

The authors developed a simulation model that corresponds to the impeller 1st stage of the compressor, taking into account the vane diffuser. Input data: overall pressure ratio 1.48; flow of stage 42 m^3/min; inlet pressure 0.09 MPa; initial gas temperature 293 K; individual gas constant 287 (J/kgK); the number of blades is 20. Also, the following parameters are used: Turbo Mesh (T5 directory, 1.0 size factor) for FlowPath 1:1 and 1 bladerow number; minimum face angle is 0.38896 rad, maximum Face angle is 2.7534 rad. The inner and outer diameters of the impeller as well as the rotational velocity are determined in the calculation in the complex ANSYS Turbo.

Figure 5 shows the gas-dynamic calculations obtained by the simulation model of the impeller. Fig. 6 shows the thermogasdynamic calculations using the simulation model of the impeller.

When gas flows through the channels of the stage, its state changes as a result of energy transfer to the flow by the impeller, gas friction, vortex formation and heat transfer. In the impeller, the flow turns in the radial direction. Thus, an additional increase in total pressure is created in the working impeller due to centrifugal force. That is, the particles of the working fluid receive additional kinetic energy.

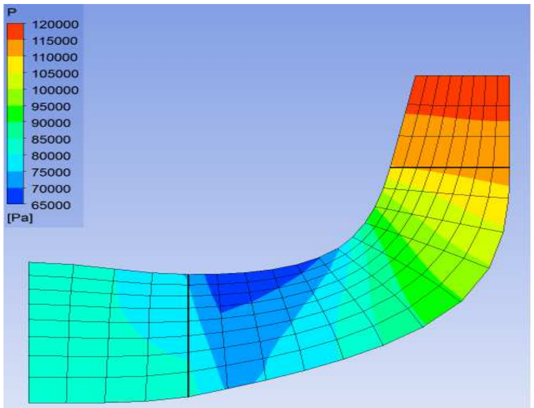
Parameters at the inlet and outlet of the impeller are summarized in Table 5.

Table 5

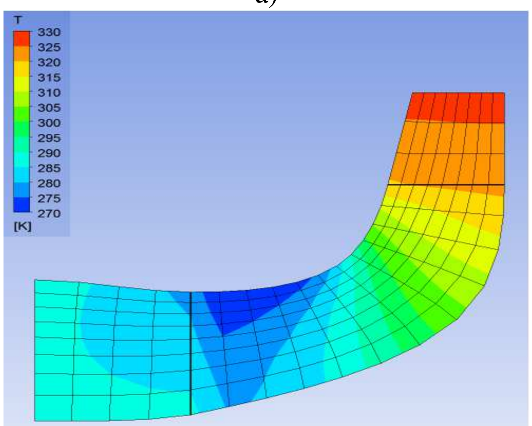
Parameters at the inlet and outlet of the impeller

Parameter	Input to the impeller	Output to the impeller
C_u, m/s	2	180
C_m, m/s	230	70
P, Pa	65 000	120 000
T, K	285	330

The analysis of the obtained by the authors modeling results confirms the method of thermogasdynamic calculation of the centrifugal compressor's stage. This is evidenced by the dependences of the pressure, temperature and velocity components at the inlet and outlet of the impeller correspond to. It should also be noted that the efficiency of the flow path can be changed depending on the shape of the impeller for a certain diffuser. The more smoothly the flow inlet to the impeller is organized, the less losses are observed, and therefore, it is possible to obtain a higher efficiency of the entire flow path of the centrifugal compressor.



a)



b)

Fig. 6. Inter-vane channels of the impeller: distribution of pressure P (a) and temperature T (b) along the channel.

6. CONCLUSIONS

The work presents a method of thermodynamic and gas-dynamic calculation taking into account the characteristic points and a detailed definition of the main geometric parameters of the impellers and the vaned diffusers as the main parts of the centrifugal compressors.

The dependences obtained and the results of calculations generally reflect the real thermodynamic processes occurring in the centrifugal compressors and can contribute to an increase in the overall efficiency of compressors.

Thus, this method of thermogasdynamic calculation of the centrifugal compressor provides an expansion of the measurement range of parameters by determining the thermogasdynamic parameters of the characteristic points of the compressor operation in each of its stages, as well as an increase in the set of geometric parameters of the passage channels, in particular, the thickness of the impeller blades and diffuser, expanding the possibilities of the proposed method for designing high-efficiency compressors, and also provides additional data for controlling the output flow or pressure of the multistage centrifugal compressor.

Numerical calculations of the finite elements of the gas dynamics of the centrifugal compressor's stage gave qualitative and quantitative results that correspond to model tests within the permissible error and theoretical ideas about the nature of the flow in the flow path of the centrifugal compressor. So, the errors of analytical calculation in comparison with CFD modeling for the first stage of the compressor are: for velocity C_2 – 3.3%, for pressure P – 2.6%, for gas temperature T – 3.7%. This indicates a high accuracy of analytical calculations of all parameters.

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Cercetarea si modelarea etapii unui compressor centrifuge multi etape in modul subsonic de functionare a gazelor

Abstract: Articolul analizează caracteristicile de proiectare și modelarea generală a treptelor compresoarelor centrifuge. Pe baza analizei se formulează principalele sarcini de modelare a unui compresor centrifugal conceput să funcționeze în domeniul subsonic al vitezelor gazului. Scopul lucrării este de a dezvolta un model de compresor centrifugal bazat pe metoda propusă de calcul termogazdinamic cu o determinare extinsă a numărului de parametri termogazdinamici și de proiectare ai unui compresor centrifugal în fiecare dintre etapele sale, precum și de a determina grosimea optimă a rotorului și a palelor difuzorului.

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