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EFFICIENCY IMPROVEMENT OF A CENTRIFUGAL COMPRESSOR STAGE WITH THE PARAMETRIC OPTIMIZATION OF THE IMPELLER BLADES

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KEY WORDS: simulation, stage, thermogasdynamics, centrifugal compressor, impeller, diffuser, methodology, efficiency.

ABSTRACT:

This work relates to the energy industry, in particular, complements the existing knowledge about a problem of increasing the overall efficiency of a centrifugal compressor stage by improving the geometric parameters of an impeller. The design features of the centrifugal compressors are analysed and the issues of their general modelling are considered in the article. The aim of the work is to increase the efficiency of the centrifugal compressor stage by improving the method of thermogasdynamic calculation with an extended determination of the number of its thermogasdynamic gas parameters and subsequent optimization of the design parameters of the impeller blades. This goal is achieved by solving the following problems: a) development of a model of the centrifugal compressor stage based on a method of thermogasdynamic calculation; b) study of the influence of the geometric parameters of the impeller and a diffuser on the components of gas velocities; c) improvement of the design methodology by optimizing the number of impeller blades, their thickness and angle of inclination to increase the efficiency of the stage. The most significant scientific results of the work are the construction method and the model of the centrifugal compressor stage, which allow taking into account real thermogasdynamic gas processes and calculating the geometric parameters of the impeller blades that are optimal in terms of efficiency.

1. INTRODUCTION

Currently the centrifugal compressor machines are widely used in many sectors of the national economies of different developed countries: in ferrous and nonferrous metallurgy, chemical, oil and gas industries, long-distance gas supply, oxygen production, etc. (Stansel, 2018; Seleznyov at. al. 1968). They are usually designed to compress and move gases and vapours (Kalinkevych et. al., 2018).

The reduction of unproductive energy costs plays a significant role at most industrial enterprise. Since the share of compressor and injection stations in the balance of energy consumption is 25–30%, and a decrease in the efficiency of centrifugal machines in operation due to wear increases the cost of compressed air. Naturally, reducing costs in the production of compressed air gives a tangible economic effect.

Reducing the energy costs of the centrifugal compressor (Jiang et. Al. 2018; Ghoreyshi 2018; Medic et. al., 2014) is possible due to the following components: designing highly efficient units and parts of the centrifugal compressor; the use of soft starters to start the units; expansion of the working area of the compressor by removing the restriction on closing the throttle valve in the operating mode; reducing the load on the compressor in idle mode by transferring it to deep throttling; increase in the total efficiency of the station due to the implementation of group regulation of pressure and productivity; reduction of cooling water consumption; identification of reserves due to the analysis of the process flow; reduction of losses from downtime due to advanced diagnostic tools, etc.

Today the issues of increasing the efficiency of work, improving (optimizing) the design, reducing the cost of production of both individual components and assemblies, and the entire compressor plant as a whole remain relevant (Ghoreyshi, 2018; Galerkin et al., 2020).

The design and principle of operation of the centrifugal compressors are based on dynamic compression of a gaseous medium. Fig. 1 shows a centrifugal compressor stage design. The impeller blades have an expanding shape, and this leads to an increase in pressure at rotation with angular velocity. Usually, the development and modelling of the centrifugal compressor stage, in particular the impeller and the diffuser, involves two stages. First, a preliminary design is carried out using the one-dimensional analysis based on previous experience, drawn up in a sketch to indicate the angles of the inlet and outlet blades and the "skeletal" dimensions. This is followed by a simulation in which the complete blades and channels geometry is determined. Model validation is performed through the successive aerodynamic analysis and stress analysis (Wan at al., 2017).

There are works on the design of impellers with different blade shapes and different productivity in the literature (Wan at al., 2017; Lyashkov, 2005). The impeller is the most important and critical part of the compressor. In work (Mangani, et al. 2012) considerable attention is paid to the design of impellers, the improvement and optimization of their geometric and operating parameters, gas-dynamic calculation and CFD-modelling (Kondratenko et al. 2019) are carried out.

The impeller is used to convert mechanical energy into pressure energy and kinetic energy. The impeller is the most important and critical part of the compressor: the impeller blades exercise force on the gas flow around them. There are three types of blades: backward curved blades with an entry angle, radial exit blades and forward curved blades. In works (Xie at al., 2018; Kim at al., 2010; Mojaddam and Pullen, 2019) considerable attention is paid to the design of impellers, the improvement and optimization of their geometric and operating parameters, as well as gas-dynamic calculation.

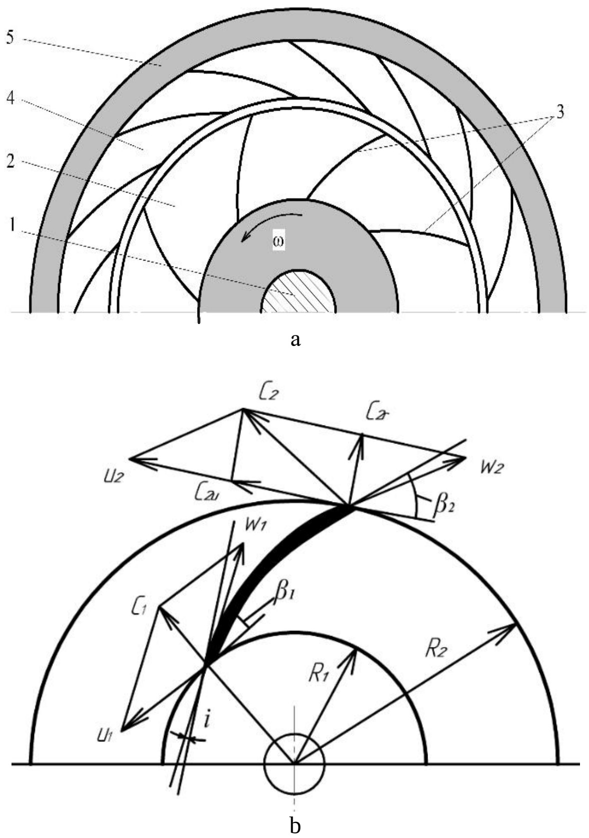

Figure 1. The centrifugal compressor stage design: a) Schematic of a blade output with an annular cavity of the centrifugal compressor: 1 – a shaft; 2 – an impeller; 3 – the impeller blades; 4 – a blade outlet (a vaned diffuser); 5 – an annular cavity; b) Compressor impeller velocity components

The second most important element of the centrifugal compressor is the diffuser (Emre and Cadirci, 2022; Seyed AmirHosein and Kosasih, 2014). At the same time, much less attention in literature is paid to the design, thermodynamic and gas-dynamic calculation of the vane diffusers of the centrifugal compressors than to the impellers.

The paper (Seyed AmirHosein and Kosasih, 2014) presents studies of the dependence of the degree of augmentation on the shape and geometry of the diffuser, such as the length and angle of expansion. However, the study of the flow field over the rotor blades in a closed turbine has not received much attention. The papers (Xie, et al., 2018) propose an optimal design method, in which the hub line in the meridional plane is changed and optimized based on numerical simulation data. In addition, the aerodynamic characteristics are compared in detail between the prototype and three modified impellers.

As a result of reviewed publications, it was noticed that there are several approaches to improve the quality and the efficiency of the centrifugal compressors through their modelling and research. The first approach is associated with the improvement of the design and production technologies of individual components of such machines. This also includes the construction of multi-compressor stations as a combination of several compressor machines of the same type, driven by their own turbines (Wan, et al., 2017). Such systems are more economical, but this causes a certain complexity of the design, which requires more advanced technologies and complicates the control of the engines. The second approach is based on the improvement of the system of control of technological parameters (Kalwar et al., 2017), both in individual units of the compressor machine, and at the exit from it. In this paper, the focus should be on the first approach, in particular, improving the design parameters of the impeller and the diffuser as one of the main interacting and interconnected components of the centrifugal compressor by mathematical modelling (Kondratenko and Kozlov, 2012).

The centrifugal compressors consume a significant amount of energy from drive motors, being one of the main consumers of energy in modern industry and transport. For the efficient use of electrical, thermal and mechanical forms of energy, it is very important to solve the problem of optimal design of the centrifugal compressor stage by modelling internal processes in terms of overall efficiency. The aim of the work is to increase the efficiency of the centrifugal compressor stage by improving the method of thermogasdynamic calculation with an extended determination of the number of thermogasdynamic gas parameters and subsequent optimization of the design parameters of the impeller blades.

2. THERMODYNAMIC AND GASDYNAMIC CALCULATION OF THE CENTRIFUGAL COMPRESSOR

The existing calculation methods are based on the provisions of the jet theory and similarity conditions with extensive use of experimental data on the thermodynamics and aerodynamics of stage elements (Strahovich at. al. 1961; Van Ness 1983). A calculation method (Cherkasskiy, 1984) is considered that gives a general idea of the geometric dimensions of the compressor stage of a stationary type operating at subsonic gas velocities.

The calculation specifies:
1) volumetric Q or mass M flow of the stage;
2) initial p_1 and final p_2 pressures;
3) the initial temperature T_1 of the gas;

4) thermodynamic characteristics of the gas under normal conditions k, R, ρ (k - adiabatic exponent, R - individual gas constant, ρ - air density).

The calculation of inviscid flow for these parameters is widely used for the primary qualitative assessment of the flow pattern and the initial assessment of the pressure of the impellers. Volumetric losses (leaks) are caused by the flow of liquid (gas) through the gaps between the impeller and the machine body from the high pressure zone into the suction cavity. The internal efficiency takes into account the volumetric and hydraulic losses in the machine, in addition to losses due to disc friction. Viscosity affects the hydraulic resistance and determines the expenditure of energy to overcome it. The flow, pressure and efficiency developed by the pump will decrease with an increase in the viscosity of the liquid. On the other hand, the viscosity of a liquid (gas) directly affects the hydraulic resistance of flows in the gaps: the higher the viscosity, the less leakage through the gaps and the higher the efficiency. Theoretically, it is impossible to take into account the influence of viscosity on the operating parameters and the shape of the characteristics. The simplest way to recalculate operating parameters and rebuild characteristics with a change in viscosity is based on the application of correction factors obtained empirically.

In this case, the thermogasdynamic calculation of the centrifugal compressor stage consists of determining the design parameters of the flow path. It is performed for one stage at subsonic gas velocities for a non-viscous working medium, since the movement of real gases in the main part of the flow approximately obeys the laws of motion of a non-viscous liquid. At high speeds and with a large number of stages the viscosity of the gas will make its own adjustments to the actual operation of the compressor.

2.1 Impeller parameters calculation

The outer diameter of the impeller D_2 is determined by:

$$D_2 = \frac{60u_2}{\pi n} \tag{1}$$

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. Therefore D_1 we find from the expression:

$$D_1 = 0.55D_2 \tag{2}$$

Determine the geometric dimensions of the channels of the impeller. First, let's define the pitch r_{d1} and r_{d2} by the inner diameter:

$$r_{d1} = \frac{\pi D_1}{Z_L} \tag{3}$$

$$r_{d2} = \frac{\pi D_2}{Z_L} \tag{4}$$

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 and output cross-section

$$l_1 = r_{d1} - \delta \tag{5}$$

$$l_2 = r_{d2} - \delta \tag{6}$$

The heights of the cross-sectional area of the impeller channel are determined from the relation

$$b_1 = 0.06 \cdot D_1, \tag{7}$$

$$b_2 = 0.012 D_2. \tag{8}$$

Let's consider determination of the components of rotation velocities in impeller. The circumferential speed u_1 is calculated as:

$$u_1 = \frac{\pi D_1 n}{60} \tag{9}$$

The air velocity at the inlet to the impeller is found by the following equation:

$$\omega_1 = \frac{Q}{Z_L F_1} \tag{10}$$

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

$$c_1 = \sqrt{\omega_1^2 + u_1^2 - 2\omega_1 u_1 \cos\beta_1} \tag{11}$$

The air velocity at the outlet to the impeller is found by the following equation:

$$\omega_2 = \frac{Q}{Z_L F_2} \tag{12}$$

Similarly for c_2 :

$$c_2 = \sqrt{\omega_2^2 + u_2^2 - 2\omega_2 u_2 \cos\beta_2}\ . \tag{13}$$

Then find the projection of c_{1u} and c_{2u} onto U:

$$c_{1u} = c_1 \cos\left(90^\circ - \beta_1\right) \tag{15}$$

$$c_{2u} = c_2 \cos\left(90^\circ - \beta_2\right) \tag{16}$$

The density of the working medium (gas) is determined by the following relationship

$$\rho = \frac{p_1}{RT_1} \tag{17}$$

where: R is individual gas constant of dry air.

2.2 Diffuser parameters calculation.

Determination of the basic dimensions of the diffuser is listed below according to expressions:

$$D_3 = 1.1D_2 \tag{18}$$

$$D_4 = \left(1.3 - 1.55\right) D_2 \tag{19}$$

where: D_2 is the outer diameter of the impeller.

Next, the geometric dimensions of the channels of the diffuser are determined. First, let's define the pitch r_{d3}, and r_{d4} by the inner diameter:

$$r_{d3} = \frac{\pi D_3}{Z_D} \tag{20}$$

$$r_{d4} = \frac{\pi D_4}{Z_D} \tag{21}$$

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

Using the obtained values, we determine the lengths cross-section diffuser channel:

$$l_3 = r_{d3} - \delta \tag{22}$$

$$l_4 = r_{d4} - \delta \tag{23}$$

where: δ – blade thickness (0.01 m).

The heights of the cross section of the diffuser channel will be determined from the following relations:

$$h_1 = \frac{Q}{c_2 l_3 \rho_2} \tag{24}$$

$$h_2 = \frac{Q}{c_4 l_4 \rho_3} \tag{25}$$

Also, dimension h can be accepted based on practical data $h_1 = h_2 = (1\text{-}1.2)b_2$.

Velocity c_4 at the outlet of the diffuser is determined from the relation:

$$c_4 = 0.4 c_2 \tag{26}$$

To determine the parameters of pressure and temperature in the diffuser, we first calculate the pressure between the impeller and the diffuser.

$$\varepsilon = \frac{p_2}{p_1} \Rightarrow p_2 = \varepsilon p_1 \tag{27}$$

$$p_l = p_1 \left(1 + \frac{1}{2 c_p T_1} \left[c_1^2 - c_2^2 + 2\left(u_2 c_{2u} - u_1 c_{1u} \right) \right] \right)^{\frac{n}{n-1}} \tag{28}$$

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 found from the formula:

$$t_l = t_1 \left(\frac{p_l}{p_1} \right)^{\frac{n-1}{n}} \tag{29}$$

Using the same formula, we define temperature t_2 at the outlet of the blade diffuser. Determine the air density by:

$$\rho_l = \left(\frac{p_l}{R T_l} \right) \tag{30}$$

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

3. STUDY OF THE EFFECT OF THE GEOMETRIC PARAMETERS OF THE IMPELLER AND DIFFUSER ON THE VELOCITY COMPONENTS

Simulation of the calculation of gas dynamics and thermodynamic parameters was performed for the centrifugal compressor with the following specified parameters: volumetric flow of stage $Q = 42$ m^3/min, inlet pressure $p_1 = 0.09$ MPa, outlet pressure $p_2 = 0.41$ MPa, initial gas temperature $T_1 = 293$ K, adiabatic exponent $k = 1.38$, individual gas constant $R = 287$ (J/kgK), air density $\rho = 1.07$ kg/m^3. Geometric dependencies for the impeller in dimensionless form are $D_1 = 0.52D_2$ for the impeller and $D_3 = 0.76D_4$ for the diffuser. The impeller blades exert a force effect on the gas flow around them. We will consider the conditions for gas entry into the blades to determine the optimal shape of the blades and their flow section. So, the centrifugal compressor's first stage was simulated. Let's consider the dependence of the effect of supply on pressure to evaluate the considered mathematical model for designing the centrifugal compressor stage. Let's consider the dependence of the effect of supply on pressure to evaluate the considered mathematical model for designing the centrifugal compressor stage. The gas-dynamic characteristic of the influence of the supply on the pressure P_l between the compressor's impeller and diffuser is presented in Fig. 2.

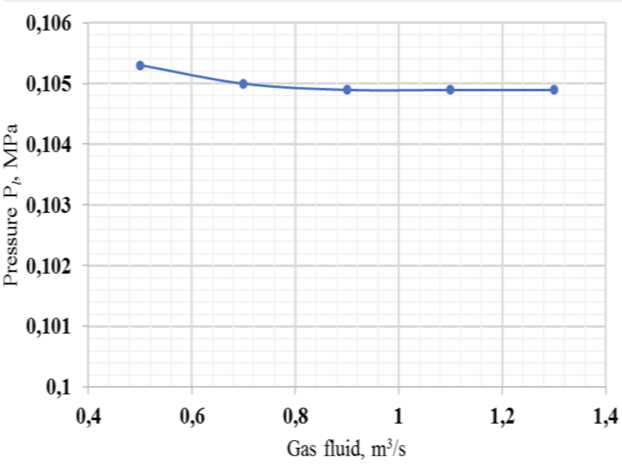

Figure. 2. Dependence of the pressure between the impeller and the diffuser on the gas volume flowrate

The graph in Fig. 2 shows that when the supply changes, the pressure practically does not change, which indicates the high quality of the proposed mathematical model of the centrifugal compressor stage. The components of the velocity for various variants of the geometric parameters of the impeller are summarized in Table 1. The thickness of the blades, their number and the exit blade angle at which they are bent are taken as the geometrical parameters of the impeller. Moreover, during the calculations, the number and thickness of the diffuser blades do not change and amount to 22 units with a thickness of 0.01 m. When gas moves through the channels of the impeller, its velocity c_1 should be decomposed into components - portable u_1 and relative ω_1. The velocity is directed at some angle β_1 in the direction opposite to the rotation (Fig. 1, b). So, if the leading edges of the blade are directed along the radius, then the gas flow could enter the working channel only with some impact. Small losses at the inlet of the impeller channels 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 transferred velocity u_1. So, if the leading edges of the blade are directed along the radius, then the gas flow could enter the working channel only with some impact. Small losses at the inlet of the impeller channels 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 transferred velocity u_1.

Accordingly, the thickness, the exit angle of the blades and their number affect the speed characteristics as can be seen from the Table 1. A decrease in some compound velocities is observed with thickening of the blades and an increase in their number. Conversely, an increase in compound velocities directly leads to a change in pressure indicators with an increase in the exit angle. These values are determined by the gas kinematics in the centrifugal compressor stage. The relative speed is determined by the capacity and size of the impeller with a sufficiently large number of the blades. The absolute speed at the exit of the blades is obtained as the geometric sum of the relative and portable speeds.

Influence of the number of the impeller's blades with 60° blade angle and 0.01 m thickness									
Influence indicator	**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_{1u}, m/s**	**c_{2u}, m/s**
20	47.104	56.7666	98.8	190	59.55	168.93	67.572	25.17	146.3
21	47.71	57.1027	98.8	190	59.12	168.85	67.54	24.98	146.24
22	48.332	57.4428	98.8	190	58.69	168.78	67.512	24.8	146.17
23	48.97	57.787	98.8	190	58.24	168.7	67.48	24.61	146.1
24	49.625	58.1353	98.8	190	57.78	168.63	67.452	24.42	146
25	50.26	58.45	98.8	190	57.32	168.5	67.4	24.22	145.97
26	50.989	58.8446	98.8	190	56.85	168.47	67.388	24.02	145.9
27	51.699	59.2059	98.8	190	56.37	168.4	67.36	23.82	145.84
28	52.43	59.5715	98.8	190	55.88	168.32	67.328	23.62	145.77
Influence of the blade thickness for 25 impeller's blades with 60° outlet blade angle									
0.006	44.27	55.11	98.8	190	61.58	169.3	67.72	26.02	146.62
0.008	47.07	56.74	98.8	190	59.55	168.93	67.57	25.17	146.3
0.01	50.26	58.45	98.8	190	57.32	168.5	67.4	24.22	145.97
0.012	53.9	60.28	98.8	190	54.88	168.17	67.27	23.2	145.64
0.014	58.14	62.23	98.8	190	52.24	167.77	67.1	22.08	145.3
0.016	63.1	64.3	98.8	190	49.43	167.38	66.95	20.89	144.96
Influence of the outlet blade angle β_2 for 25 impeller blades and 0.01 m blade thickness									
60	50.26	58.45	98.8	190	57.32	168.5	67.4	24.22	145.97
61	47.05	56.74	98.8	190	57.32	169.55	67.82	24.22	148.6
62	47.05	56.74	98.8	190	57.32	170.55	68.22	24.22	150.6
63	47.05	56.74	98.8	190	57.32	171.56	68.62	24.22	152.86
64	47.05	56.74	98.8	190	57.32	172.57	69.03	24.22	155.1
65	47.05	56.74	98.8	190	57.32	173.58	69.43	24.22	157.32
66	47.05	56.74	98.8	190	57.32	174.6	69.8	24.22	159.5
67	47.05	56.74	98.8	190	57.32	175.6	70.24	24.22	161.6

Table 1. Velocity components of the compressor stage for 42 m3/min gas flowrate

4. IMPROVING THE CENTRIFUGAL COMPRESSOR'S EFFICIENCY IN TERMS OF PRESSURE CHANGES

Power is supplied from a third-party source for the operation of the compressor. The process of energy transfer can be conditionally divided into two stages. At the first stage energy is transferred from the drive through the shaft to the blades of the impeller, and from the blades it is transferred to the flow at the second stage. In this case, part of the power is lost to overcome the friction of the impeller against the gas and mechanical losses associated with the deformation of the rotor elements and friction in the bearings. Another part of the energy is lost with leakage of the working gas. Not all the power transferred to the impeller goes to increase its potential or kinetic energy. Part of the power is spent on overcoming friction in the flow path of the compressor. Part of the energy is wasted due to the fact that it is necessary to compress the more heated gas (due to losses). The remaining energy is supplied to the working fluid and is used to compress the working gas.

The compressor efficiency is calculated according to the following equation

$$\eta = \frac{\bar{H}_k}{2\mu + \alpha} \tag{31}$$

where: $\bar{H}_k$ – head coefficient; μ – head change coefficient; α – impeller friction coefficient.

The head coefficient of the compressor depends on the type of diffuser used in the compressor and ranges from 1.25 to 1.45 for an impeller with a diameter of 85...640 mm. Fig. 3 shows the average values of head coefficient changes for the compressor with the vaned and vaneless diffuser. We use the characteristic for the vaned diffuser with a number of blades of 22 units.

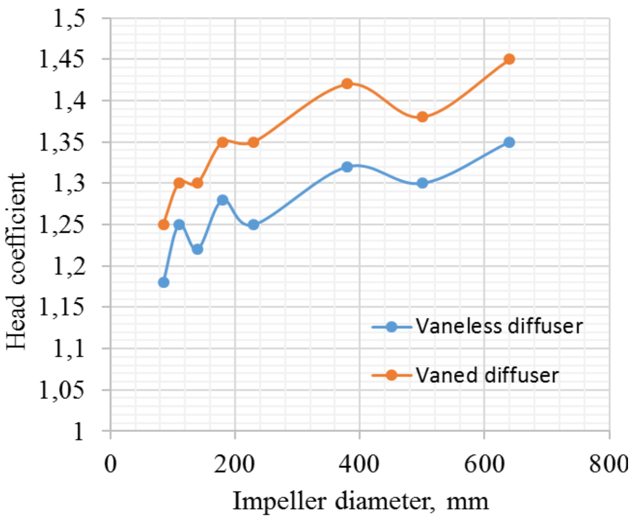

Figure 3. Average values $\bar{H}_k$ of the turbocompressors

The head change coefficient depends on the geometrical parameters of the impeller. The smaller it is, the more efficiently the centrifugal compressor stage works and its efficiency increases.

The number of the impeller blades directly affects the value of theoretical head created by the impeller. Along with this, the number of the blades also affects the efficiency of the impeller: the load on each blade increases – the pressure and velocity gradients on the surface of the blades increase – the intensity of diffuser zones on the profile increases with a decrease in the number of blades while maintaining a constant pressure. The surface area of the blades increases, and friction losses increase with an increase in the number of blades. Therefore, the optimal design of the impeller to ensure efficiency will be achieved with a certain range of the minimum number of blades which can withstand the load with satisfied head values.

Let`s determine the influence of the number of the blades on the value of the head change coefficient. To do this, we take the thickness of the blades 0.01 m and calculate the value of the head change coefficient for a certain number of the impeller blades of the considered compressor stage as (Baturin, 2013).

$$\mu_1 = \frac{1}{1 + \frac{2\pi}{3z_k} \cdot \frac{1}{1 - \left(D_1 / D_k \right)^2}} \tag{32}$$

The results of the calculation are summarized in Table 2. The maximum value of the efficiency with a set number of the impeller blades is determined based on the obtained data. In this case, there is a tendency to increase the efficiency with a decrease in the number of blades, and therefore the possible minimum value of the blades in the region of 20 pieces should be taken. A further decrease in the number of blades can lead to a loss in the strength of the impeller.

Z_L	D_0	D_H	D_1	μ_1	η
20	151.2	154.60	152.91	0.8994	0.7639
21	151.2	154.42	152.82	0.9038	0.7604
22	151.2	154.26	152.73	0.9077	0.7571
23	151.2	154.10	152.66	0.9114	0.7542
24	151.2	153.97	152.59	0.9148	0.7515
25	151.2	153.84	152.52	0.9179	0.7490
26	151.2	153.72	152.46	0.9208	0.7467
27	151.2	153.61	152.41	0.923	0.744
28	151.2	153.5	152.36	0.926	0.742

Table 2. Head change coefficient and efficiency for a certain number of the blades

The compressor's hub diameter is obtained as

$$D_0 = (0,2...0,3) \cdot D_k \tag{33}$$

The impeller's outer diameter is found as:

$$D_H = \sqrt{D_0^2 + \frac{4F_1 \cdot 10^2}{\pi}} \tag{34}$$

The impeller's average inlet diameter is:

$$D_1 = \sqrt{\frac{D_H^2 + D_0^2}{2}} \tag{35}$$

After that, let's consider a blade thickness effect to the head change coefficient and efficiency. The corresponding results are given in Table 3. In this case, the thickness of the impeller blades does not have a significant effect on the efficiency value, since this value varies within 5 decimal places. Therefore, an average value of 0.01 m should be applied in view of the strength of the impeller.

Δ	D_0	D_H	D_1	μ_1	η
0.006	151.2	154.6	152.9	0.899431	0.763909
0.008	151.2	154.4	152.8	0.899437	0.763903
0.01	151.2	154.3	152.7	0.899443	0.763898
0.012	151.2	154.1	152.7	0.89945	0.763893
0.014	151.2	153.9	152.6	0.899456	0.763888
0.016	151.2	153.8	152.5	0.899462	0.763882

Table 3. Head change coefficient and efficiency for a certain number of blades thicknesses

Further research in the field of increasing efficiency is associated with determining the optimal angle of inclination of the blades. A different equation for finding the head change coefficient, that takes into account the outlet angle of the impeller blades, is used for this purpose (Baturin, 2013).

$$\mu_2 = \frac{1}{1 + \dfrac{1.2 \cdot (1 + \sin \beta_2)}{z_l \left(1 - {D_1}/{D_k}\right)^2}} \tag{36}$$

The results of head change coefficient and efficiency calculations for different blade angles are presented in Table 4. Significant changes in the indicators are noticeable in the range from 60° to 67°.

β_2	D_0	D_H	D_1	μ_2	η
60	151.2	154.30	152.76	0.96	0.715
61	151.2	154.30	152.76	0.99	0.690
62	151.2	154.30	152.76	0.98	0.699
63	151.2	154.30	152.76	0.93	0.733
64	151.2	154.30	152.76	0.9	0.761
65	151.2	154.30	152.76	0.94	0.757
66	151.2	154.30	152.76	0.991	0.725
67	151.2	154.30	152.76	0.994	0.694

Table 4. Head change coefficient and efficiency for a certain number of blades angles

Now, the results of finding the efficiency taking into account the head change coefficient using the previously mentioned equations should be combined to determine the maximum efficiency of the compressor stage. Fig. 4 shows comparison of the stage efficiency and the head change coefficient characteristics. The intersection of the two curves determines the best geometry of the compressor impeller. In this case, 20 blades curved at the outlet at an angle of 64 degrees should be chosen for the further construction.

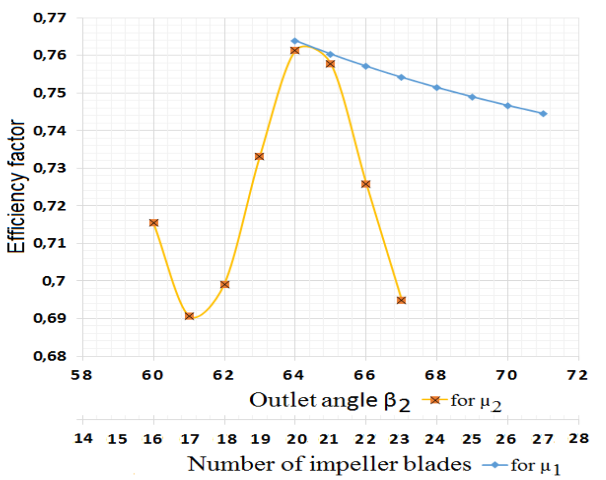

Figure. 4. Stage efficiency vs. head change coefficient dependencies

We will determine the thermogasdynamic gas parameters for the centrifugal compressor stage with the obtained optimal parameters. The number of diffuser and impeller blades is 20, the blade thickness is 0.01 m and the outlet impeller blades angle is 64°. The components of the obtained velocities are ω_1 = 47.1 m/s, ω_2 = 56.77 m/s, u_1 = 98.8 m/s, u_2 = 190 m/s, c_1 = 59.55 m/s, c_2 = 172.82 m/s, c_4 = 69.13 m/s, c_{1u} = 25.17 m/s, c_{2u} = 155.33 m/s; P_1 = 117 000 Pa, T = 318 °C.

The construction and gas-dynamic parameters of the impeller and the diffuser have a significant impact on the efficiency of the whole stage. The main parameters of the compressor's impeller and diffuser correspond to the optimal combination of other design and gas-dynamic parameters. Therefore, the design mode of the impeller and the diffuser for the given operating conditions is obtained with the certain combinations of these parameters. The dependence of the influence of the geometrical parameters of the compressor stage on the thermogasdynamic gas parameters can be estimated using CFD-modeling methods.

5. CONCLUSIONS

The paper presents the development and study of the hydrodynamic model of the centrifugal compressor stage based on the method of thermodynamic and gas-dynamic calculations with a detailed determination of the main geometric parameters of the impellers and the bladed diffusers as the main parts of such machines. Firstly, a preliminary review of the literature highlights the main characteristics of the centrifugal compressors. In particular, the impeller and the vaned diffuser are identified as the most important and interrelated parts of compressors. Secondly, the methodology for thermodynamic and gas-dynamic calculation of the bladed impeller and the bladed diffuser of the centrifugal compressor is presented. It takes into account the volumetric or mass flow of the stage, the initial and final pressures, the initial gas temperature, and the thermodynamic characteristics of the gas under normal conditions. As a result, the authors obtain the main geometric parameters of the impeller and the diffuser.

The influence of blades' thickness, number and angle at the compressor's efficiency is investigated. The calculation methodology is improved in the way of obtaining the optimal values of these parameters. The dependences are 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. The practical value of the obtained results lies in the possibility of applying the proposed methodology and model for calculating and modelling highly efficient single- and multi-stage centrifugal compressors, which will improve the quality of their design in terms of overall efficiency for different operating modes.

Thus, this approach to the thermogasdynamic calculation of the centrifugal compressor provides an expansion of the range of the geometric parameters of the flow channels of the impeller and the vaned diffuser, which expands the scope of the proposed method for designing the highly efficient compressors.

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