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A Methodology and Information System for Computing and Optimization of Impellers and Vanned Diffusers Geometry Parameters

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***Abstract* – The study aims to develop an information-computing complex for computer design of a centrifugal compressor with parallel calculation of stages and optimization of the geometric parameters of the impellers and the diffusers. The paper presents a universal methodology and computerized information system of the main geometry parameter determination and optimization of the centrifugal compressor impellers and vanned diffusers. Optimization of cross-sectional areas of the input and output channels of the impeller and diffuser blade channels is held using a gradient descent method by gas flowrate quadratic integral deviation criteria. The information-computing complex is built on the algorithm proposed by the authors and implemented as a computer program with a human-machine interface. Calculation data are written in the form of numerical arrays with the possibility of interpolating data and obtaining graphical dependencies.**

***Keywords* – Centrifugal compressor, diffuser, impeller, information system, methodology, optimization.**

I. Introduction

Centrifugal compressors are the equipment belonging to the group of dynamic type compressors with radial design. The main advantage of installations of this type is their high performance, which is many times higher than that of other types of compressors. Due to this advantage, centrifugal air compressors, the design of which allows for their intensive operation, are widely used on the industrial scale, i.e., in the oil refining industry, metalworking and other fields of activity. The device and principle of operation of centrifugal compressors are based on the dynamic compression of a gaseous medium. The main element of the equipment is a rotor consisting of a shaft with mounted impellers. During the operation of the equipment, the gas particles are affected by the force of inertia, which arises due to the presence of the rotational movement performed by the impeller blades. In this case, the gas moves from the centre of the compressor to the edge of the impeller and, as a result, the gas is compressed and acquires speed. Further, the gas velocity decreases and subsequent compression occurs in a circular diffuser – kinetic energy is converted into potential energy. At the next stage, the gas enters the return channel and passes to the next stage of the plant.

An important difference between centrifugal plants and other types of equipment is the absence of contact between oil and gas. In the case of units of this type, the requirements for lubrication of the working elements of the equipment are much lower than in volumetric units. At the same time, the lubricant completely protects the elements of the equipment from rust, and the oil, which has low oxidation, lubricates gears, seals and bearings as efficiently as possible.

Thus, the operation of a centrifugal compressor has a fairly simple principle of operation and is based on the rotational movement of the impeller blades, which is one of the main working elements of centrifugal group units. At the same time, this equipment is characterised by a rapid increase in the pressure level and the achievement of its maximum value in a short period of operation of the unit.

Well-known, centrifugal compressors consume a significant amount of energy from drive motors, being one of the main energy consumers in modern industry and transport. For the efficient use of electrical, thermal and mechanical types of energy, it is very important to solve the problem of optimal design of centrifugal compressors according to the selected criterion of optimality. During the preliminary design phase, the compressor performance can be determined using various techniques depending on the geometric configuration and the limits of computational accuracy.

During the operation of the impeller, the gas velocity decreases, and with the least possible losses, the energy of the velocity is converted into the energy of pressure in the diffuser. Fig. 1 shows two types of diffusers (vane and vaneless). In a stationary vane diffuser, the gas is inhibited with a corresponding increase in pressure. Then the flow in the reverse

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guide vanes passes through the interscapular channels with increasing cross-section and proceeds to the next stage.

Accordingly, determining the value of the passage areas between the blades of the impeller and the diffuser is very important, since these values affect the ongoing processes in the centrifugal compressor. Due to the selection of the design parameters of the blades, it is possible to achieve the required gas flow rates inside the stages of the centrifugal compressor.

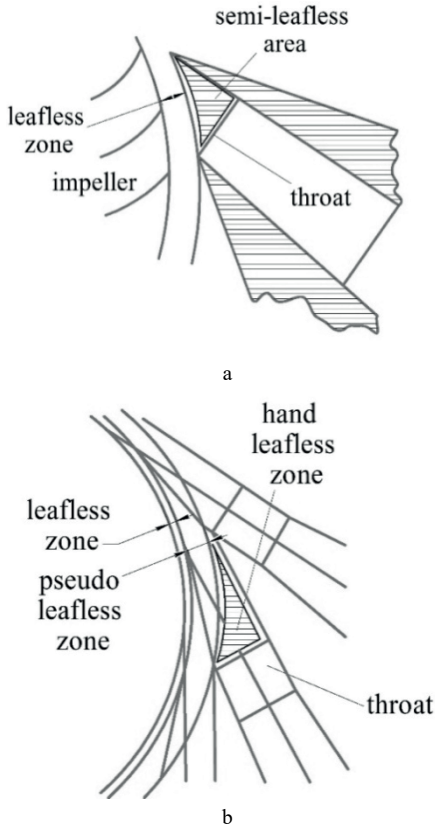


Fig. 1. Schematic diagrams of flow channel structures of vane (a) and vaneless (tube) diffusers (b).

We know about the methods of thermogasodynamic calculation of the centrifugal compressor in which the stages of calculation are decomposed into elementary processes in the main elements of the centrifugal compressor: the impeller of the centrifugal compressor and a blade diffuser. To perform thermomagodynamic calculation, a sequential calculation of the main parameters of the working process of the centrifugal compressor from element to element should be done taking into account the implementation of the basic laws of conservation of gas dynamics and thermodynamics.

There are works on the design of impellers with different blade shapes and different productivity in the literature [1]–[6]. There are three types of blades: backward curved blades with an entry angle $\beta_2 < 90°$, radial exit blades ($\beta_2 = 90°$), forward curved blades ($\beta_2 > 90°$). *The impeller* is the most important and critical part of the compressor. In works [1]–[6], 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 are carried out.

The second most important element of the centrifugal compressor is *the diffuser* [7], [8]. They are formed by two parallel walls machined via turning or electric discharge machining [8]. They have larger radial dimensions and lower efficiency than diffusers with special blades. These vanes shorten the gas path and reduce the radial dimensions of the diffuser [9]. The speed in a vaneless diffuser decreases relatively slowly, inversely with the diameter. In a vane diffuser, due to an artificial increase in the angle compared to its initial value, the speed drops much more intensively. 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. In multistage machines a *return apparatus* is used for transferring from the diffuser of the previous stage to the impeller of the next stage. The blades of the apparatus receive the gas flow coming out of the diffuser without shock and supply it uniformly around the circumference, providing a radial gas inlet to the blades of the next impeller. The combination of impeller, diffuser and return apparatus constitutes the stage of the centrifugal compressor [10].

The injection chamber is often made in the form of a volute, it is an element of the end stage discharging gas outside the flow path of the machine. The volute collects gas that flows uniformly along the outlet circumference of the diffuser and directs it either to the heat exchanger or to the consumer. The end stage can supply gas to the cooling device in the compressor. The spiral shape is taken close to the natural streamline when the gas moves in a space free from the blades. The spiral chamber ends with a conventional diffuser that goes into the machine discharge pipe [11].

In rare cases, the gas to be compressed is supplied to the machine from the environment. In most cases, the air first passes through the air filter, then through the pipeline it is supplied to the suction pipe of the machine and through the suction chamber to the first impeller. The purpose of the suction chamber is to ensure a uniform supply of air or gas to the first impeller with as little pressure loss as possible and without swirling the flow [12]. The last three compressor components (the return apparatus, the injection chamber and the suction chamber) are designed according to the technological parameters of the compressor and depend on the geometric dimensions of the impeller and diffuser, current climatic zone and other parameters [13].

Recent research efforts have primarily shown advances in the following three areas: decision support, virtual reality (VR) tool support, and integration support [7], [14].

There are different approaches for main geometry and technical parameters of centrifugal compressor calculation – from one-dimensional to three-dimensional Computational Fluid Dynamics (CFD) methods with Fluid Structure Interaction (FSI) simulation as the most new [7], [15]–[17]. Such computing instruments help determine and visualize the investigated physical parameters.

The calculation of needed parameters and their optimization are usually held by classical calculation methods. The authors of the paper calculated various physical parameters for different technical objects: thermoacoustic engines [18]; mobile robots able to move on inclined ferromagnetic surfaces [19]–[22];

adaptive sensory systems of manipulators [23]; level measurement [24] and so on. At present, special attention is paid to artificial intelligent methods: neural networks [25], hybrid neuro-fuzzy systems [26], [27], genetic algorithms [28], multi-agent learning [29]). They are also successfully implemented in centrifugal compressor designing for substantial improvement of impeller blades and diffuser channels [7], [30].

Among the information systems for designing, the centrifugal compressor should include work [31], according to which the information-calculation system of the centrifugal compressor design contains a software unit for data entry and a software unit for calculation and accumulation of data, which are connected in series. The second program unit has a separate structure in which there are subunits of compressor by means of Microsoft Office Excel software and a sub-block of three-dimensional modelling and determination of thermogasodynamic parameters of the stage of centrifugal compressor by means of SolidWorks software. In addition, the second software unit supports modelling and calculation of the centrifugal compressor stage model for strength and oscillation. Study [32] presents the information and calculation complex design of the centrifugal compressor, which contains the first software unit for data entry, the second software unit for calculation and accumulation of data and the third software unit for parametric optimization, which are interconnected by serial links. Data of designing of geometrical sizes of the stage of the centrifugal compressor are processed and then transferred to the second block of calculation and data accumulation in which the mathematical model on the basis of Ansys, Excel or Numeca software complexes is integrated into the parametric optimization software unit based on IOSO software. The parametric optimization software unit searches for optimized geometric quantities as the maximum/minimum of the object function of several variables.

The reviewed publications indicate that there are several approaches to improving the quality of centrifugal compressors. The first approach is associated with design improvements and production technologies for individual units 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 [33]. Such systems are more economical; however, they cause a certain complexity of the design, which requires more advanced technology and complicates the control of engines [34]. The second approach is based on improvements in the monitoring and control system of technological parameters, both in individual nodes of the compressor machine and at the outlet from it [35]. In this study, the main attention should be paid to the first approach, in particular, to improve the design parameters of the impeller and the diffuser as one of the main interacting and interconnected components of the centrifugal compressor.

We know about information and calculation systems for the design of centrifugal compressors in which the stages of calculation are decomposed into elementary processes in the main elements of the centrifugal compressor: the impeller of the centrifugal compressor and a blade/vane diffuser. To perform thermomagodynamic calculation, a sequential calculation of the main parameters of the working process of the centrifugal compressor from element to element should be done taking into account the implementation of the basic laws of conservation of gas dynamics and thermodynamics.

The object of study is gasthermodynamic processes occurring at each stage of the centrifugal compressor, considering the implementation of the basic laws of conservation of gas-thermodynamics, as well as the influence of the design parameters of the compressor on the flow of these processes.

The subject of study is a universal methodology for computing the main geometry parameters of centrifugal compressors, algorithmic solution for the proposed methodology computer implementation and computerized information system for the impeller and diffuser preliminary design.

The aim of the study is to develop an information-computing complex for computer design of the centrifugal compressor with parallel calculation of stages and optimization of the geometric parameters of the impellers and the diffusers.

II. Information System of Thermo-Gas Dynamic Calculation and Optimization of Centrifugal Compressor

The functional diagram of the information-calculation complex of centrifugal compressor design is presented in Fig. 2. It includes software input unit 1, software unit for calculation and accumulation of data 2, software unit for parametric optimization 3. Each of the software units is implemented in high-level programming language MATLAB and has its own internal structure of computing units.

The information and calculation complex [36] works as follows. The operator enters the initial design values, which include volumetric stage feed Q (m^3/s), inlet p_1 (MPa) and outlet p_2 (MPa) pressure, initial gas temperature T (K), adiabatic index q, universal gas constant R (J/kg·K) and gas density ρ (kg/m^3), in the initial data processing unit 4. The specified data for the design of the compressor is transmitted to the unit calculating the number of stages of the compressor 5. The number of stages depends on the parameters of gas dynamics and is determined by the formula

$$Z = \frac{\ln\left(\frac{\varepsilon}{\varphi}\right)}{\ln(G)}, \quad (1)$$

where ε – a coefficient of pressure in the centrifugal compressor; φ – a coefficient that takes into account pressure losses due to leaks and internal friction (φ = 0.95); G – a degree of increase in pressure in one stage from 1.4 to 2.2; Z – a number of stages.

Next, the data are transmitted to the data generation unit for calculating the parameters of each stage 7. It forms the task of calculating the parameters, as well as checks the integer value of the obtained number of stages.

The specified number of impeller stages is rounded to an integer. The value of G is recalculated taking into account the integer value of the impeller stages

$$G = \left(\frac{\varepsilon}{\varphi}\right)^{\frac{1}{Z}}. \quad (2)$$

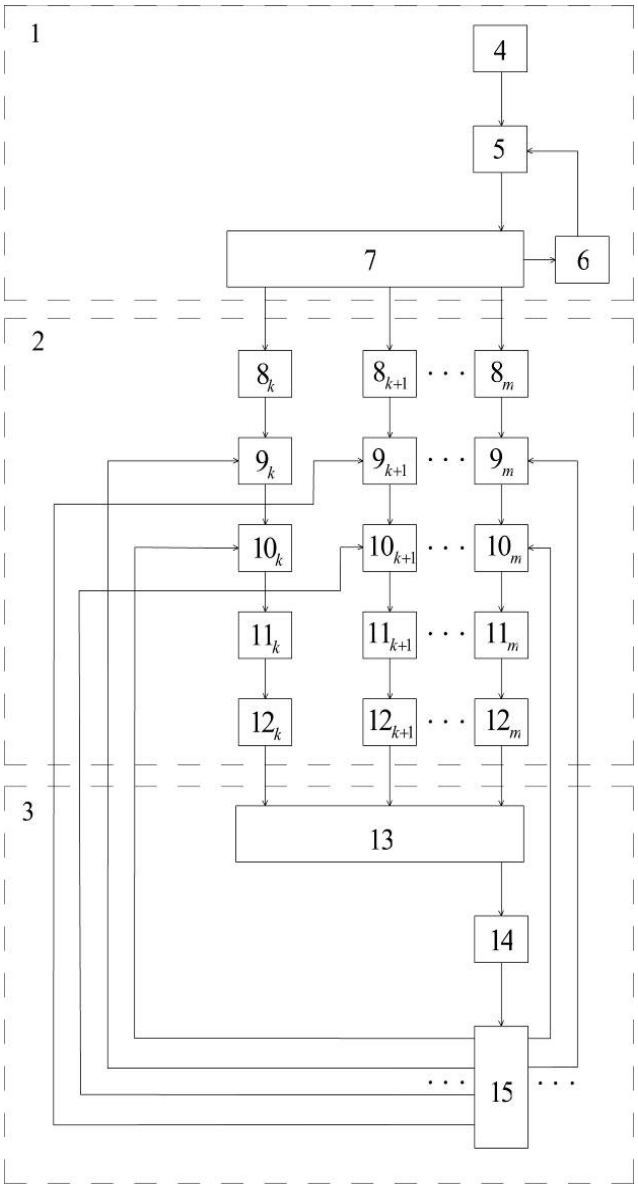


Fig. 2. Functional diagram of the information and calculation complex for calculating the centrifugal compressor: 1 – data entry program block, 2 – data calculation and accumulation program block, 3 – parametric optimization program block, 4 – initial data processing block, 5 – number of steps calculation block, 6 – step number refinement block, 7 – data generation block for calculation of the parameters of each stage, 8_k, 8_{k+1}, 8_m – blocks for determining the main dimensions of the impeller and diffuser, 9_k, 9_{k+1}, 9_m – blocks for determining the geometric parameters of the blades of the impeller, 10_k, 10_{k+1}, 10_m – blocks for determining the geometric parameters of diffuser blades, 11_k, 11_{k+1}, 11_m – blocks for determining the component speeds of rotation of the impeller and diffuser, 12_k, 12_{k+1}, 12_m – blocks for determining the thermodynamic parameters of the impeller and diffuser, 13 – block for analysing the interacting parameters of each stage, 14 – block search for optimized values and 15 – block of geometric parameter correction.

The number of stages of the compressor is recalculated with the specification of the degree of pressure increase of each stage in the unit for specifying the number of stages of the compressor 6 according to formulas (1) and (2) in the case of an integer value.

As soon as the data on the number of stages correspond to the integer value, the data generation unit for calculating the parameters of each stage 7 transmits data to the units for determining the basic dimensions of the impeller and diffuser 8_k, 8_{k+1}, 8_m. They calculate geometric parameters, including diameters and pitch blades for the compressor impeller and diffuser.

Next, the transition is made to determine the geometric parameters of the impeller. The outer diameter of the impeller D_2 is determined by the formula:

$$D_2 = \frac{60U_2}{\pi n}. \quad (3)$$

The ratio of inlet and outlet diameters is chosen to be approximately 0.5. Possible deviations from 0.48 to 0.60 are accepted $D_1/D_2 = 0.5–0.6$.

Above all, let us determine the basic dimensions of the diffuser:

$$D_3 = 1.1\, D_2, \quad (4)$$

$$D_4 = (1.3 - 1.55) D_2, \quad (5)$$

where D_3 and D_4 are the inner and outer diameters of the diffuser, respectively.

Then the cross-sectional area of the inlet and outlet channels of the impeller blades should be determined. First, the step t_{d1} on the inner diameter is calculated as follows:

$$t_{d1} = \frac{\pi D_1}{Z_l}, \quad (6)$$

where Z_l is a number of impeller blades (Z_l = 16…32).

Similarly, for the outer diameter

$$t_{d2} = \frac{\pi D_2}{Z_l}. \quad (7)$$

According to the thickness of the blade $\delta = 0.01$ m is the length of the input l_1 and output l_2 cross-section of the passage plane

$$l_1 = t_{d1} - \delta, \quad (8)$$

$$l_2 = t_{d2} - \delta. \quad (9)$$

Next, the components of the impeller speed and the thermal performance of the gas are determined. Thus, the throughput planes of the inlet and outlet channels of the vane diffuser have to be found. The steps of placing the vanes on the inner and outer diameters are, respectively:

$$t_{d3} = \frac{\pi D_3}{Z_d}, \quad (10)$$

$$t_{d4} = \frac{\pi D_4}{Z_d}, \quad (11)$$

where Z_d is a number of diffuser blades (Z_d = 20…28).

The lengths are determined using the obtained values:

$$l_3 = t_{d3} - \delta, \quad (12)$$

$$l_4 = t_{d4} - \delta. \quad (13)$$

Then the following successive blocks for determining the geometric dimensions of the impeller blades 9_k, 9_{k+1}, 9_m and the blocks for determining the geometric dimensions of the diffuser blades 10_k, 10_{k+1}, 10_m, calculate the cross-sectional areas of the inlet and outlet channels of the impeller (F_1, F_2) and the diffuser (F_3, F_4) blades.

$$F_1 = l_1 b_1, \quad (14)$$

where b_1 – a width of the inlet sections of the passage plane between the blades of the impeller. b_1 is related to D_1 as follows: b_1/D_1 = 0.05–0.1. Taking b_1 = 0.06 D_1 the passage area of the inlet channel F_1 is determined by the inner diameter. Similarly,

$$F_2 = l_2 b_2, \quad (15)$$

where b_2 – a width of the outlet sections of the passage plane between the blades of the impeller. Since b_2/D_2 = 0.01–0.02, calculated b_2 = 0.012 D_2. Therefore,

$$F_3 = l_3 b_3, \quad (16)$$

$$F_4 = l_4 b_4, \quad (17)$$

where b_3 and b_4 are the widths of the inlet and outlet sections of the passage plane between the blades of the diffuser, respectively. Moreover $b_3 = b_4$ is associated with b_2 as follows: $b_3 = b_4 = (1…1.2)b_2$. Thus, the throughput areas of the inlet and outlet channels of the vane diffuser are determined.

On the basis of the received data, the calculation of all components of speeds of the compressor is carried out (ω_1, ω_2, U_1, U_2, C_1, C_2, C_4, C_{1u}, C_{2u}) in blocks of definition of components of rotation velocities of the impeller and the diffuser 11_k, 11_{k+1}, 11_m.

Let us determine the parameters at the entrance to the impeller. Circular velocity U_1 is determined as follows:

$$U_1 = \frac{\pi D_1 j}{60}, \quad (18)$$

where j – rotor angular velocity. According to the following equation, the air velocity at the inlet of the impeller is

$$\omega_1 = \frac{Q}{Z_l F_1}. \quad (19)$$

Taking angles for relative velocities β_1 = 25° and β_2 = 61° (β_1 = 20–40°, β_2 = 60–65°), the absolute speed C_1 is

$$C_1 = \sqrt{\omega_1^2 + U_1^2 - 2\omega_1 U_1 \cos\beta_1}. \quad (20)$$

The projection of C_1 on U is obtained from the equation

$$C_{1U} = C_1 \cos\left(90^\circ - \beta_1\right). \quad (21)$$

The parameters at the exit of the impeller are determined by the following calculations:

$$\omega_2 = \frac{Q}{Z_l F_2}, \quad (22)$$

$$C_2 = \sqrt{\omega_2^2 + U_2^2 - 2\omega_2 U_2 \cos\beta_2}, \quad (23)$$

$$C_{2U} = C_2 \cos\left(90^\circ - \beta_2\right). \quad (24)$$

Let us find speed C_4 from

$$C_4 = 0.4 C_2. \quad (25)$$

Then the components of the pressures and temperatures of each stage are calculated in the units for determining the thermodynamic parameters of the impeller and the diffuser 12_k, 12_{k+1}, 12_m.

The pressure at the outlet of the first stage p_2 and the pressure at the outlet of the impeller p_l are found according to the following dependences:

$$\varepsilon = \frac{p_2}{p_1} \Rightarrow p_2 = \beta_1 p_1, \quad (26)$$

$$p_l = p_1\left(1 + \frac{1}{2C_p T_1}\left[C_1^2 - C_2^2 + 2\left(U_2 C_{2u} - U_1 C_{1u}\right)\right]\right)^{\frac{n}{n-1}}, \quad (27)$$

where n – a polytropic coefficient (n = 1.38). The air temperature at the outlet of the impeller T_l and at the outlet of the vane diffuser T_2 is calculated on the basis of the formulas:

$$T_l = T_1\left(\frac{p_l}{p_1}\right)^{\frac{n-1}{n}}, \quad (28)$$

$$T_2 = T_l\left(\frac{p_2}{p_l}\right)^{\frac{n-1}{n}}. \quad (29)$$

Accordingly, the air density ρ_l between the impeller and the diffuser is as follows:

$$\rho_l = \frac{p_l}{RT_l}. \quad (30)$$

The obtained data are processed to the block of analysis of interacting parameters of each stage 13. The operating pressures P at the inlet and outlet of each stage of the compressor are calculated as the pressure ratio W in the block 13:

$$\frac{P_{k+1}}{P_k} = \frac{P_m}{P_{m-1}} = W. \quad (31)$$

The temperatures at the inlet and outlet of each of the stages of the compressor are calculated according to the dependence:

$$T_{k+1} = T_k \cdot \left(\frac{P_{k+1}}{P_k}\right)^{\frac{n-1}{n}}. \quad (32)$$

The density of the gas in each stage of the compressor is as follows:

$$\rho_k = \frac{P_k}{RT_k}. \quad (33)$$

The volume of gas in each stage of the compressor is as follows:

$$V_k = \frac{m \cdot R \cdot T_k}{P_k}. \quad (34)$$

Then all the calculated parameters are archived in a common database. After determining and analysing the interacting parameters, the data are transmitted to the search unit for value optimization 14. The geometric parameters of the cross-sectional area of the inlet and outlet channels of the impeller blades (F_1, F_2) and the diffuser (F_3, F_4) are optimized using the gradient descent method [37]. Moreover, this method of gradient descent for the objective function uses the quadratic integral deviation of the real value of the supply of Q_R stage from a given Q according to the dependence:

$$\min_{\mathbf{H}} f(p, \mathbf{H}) = \min_{\mathbf{H}} \left[\int (Q(p) - Q_R(p))^2 dp \right]. \quad (35)$$

Vector optimization parameters are $\mathbf{H} = \{F_1, F_2, F_3, F_4\}$. The initial values of the optimization parameters are selected equal to 1 at the beginning of the optimization. The following iterative procedures are used to optimize the cross-sectional areas of the inlet and outlet channels of the impeller blades and the diffuser according to the dependence:

$$F_u[s+1] = F_u[s] - \xi[s] \frac{\partial f(\mathbf{H})}{\partial F_u}\Big|_{F_u[s]}. \quad (36)$$

After determining the optimized values, the data are sent to the unit for adjusting the geometric parameters 15. In turn, it transmits the adjusted parameters to the units for determining the geometric dimensions of the impeller blades 9_k, 9_{k+1}, 9_m and blocks for determining the geometric dimensions of the diffuser blades 10_k, 10_{k+1}, 10_m. All subsequent processes are repeated until the required efficiency of the centrifugal compressor is obtained.

These calculations contain operations of data analysis from graphs, which requires interpolation of tables of datasets. The MATLAB programming language is best suited for these purposes, as its functions are designed to analyse and process data presented as numerical arrays, which accurately describe the thermodynamic and gas-dynamic parameters of centrifugal compressors. MATLAB makes writing programs faster than traditional programming languages, because there is no need to declare variables, define types and allocate memory, and therefore it allows reducing the time of solving the proposed application problem. In addition, the MATLAB language has support for vector and matrix operations that is necessary to solve similar engineering problems of calculating the parameters of compressors. Switching to vector and matrix operations eliminates the need to use excessive "for" loops. As a result, one line of MATLAB code can often replace several lines of C/C++ code. MATLAB considerably simplifies development of new algorithms, reduces time of the decision of typical problems for an order of magnitude and gives the chance to build the big information database [20], [37] in comparison with traditional programming languages (C/C++, Java, Pascal, FORTRAN).

Multilevel software was developed based on the algorithm using array components. A large number of closely interconnected components of the functional diagram (Fig. 2) requires the use of flexible and distributed software architectures for calculating the centrifugal compressor. This, in turn, affects human-machine interaction and requires the development and use of high-quality software and user tools [38], [39]. The user interface (UI) capabilities should fully reflect the functionality of the program. The UI response time should be minimal, so that the user does not have to wait longer than necessary to perform the specified action. UI reliability is a property to keep in time within the established limits values of all parameters of gas and thermodynamics which characterise the ability to carry out the necessary functions in the set modes and conditions of application. UI maintenance is a process of improving, optimizing and eliminating the shortcomings of UI after the transfer of software into operation. Designing human-machine interaction interfaces for building a software package for calculating the impeller and diffuser of the centrifugal compressor is a rather complex and time-consuming process. The complexity of designing user interfaces is primarily due to the ambiguity of the interpretation of the requirements for computer systems and their clarity. In addition, a characteristic feature of this process is the nonlinearity of the process that is reflected in the absence of simple design algorithms, the presence of a large number of alternative solutions, disordered design stages.

The UI in MATLAB consists of graphical windows, which contain controls (buttons, lists, switches, labels, scroll bars, input areas, menus), as well as axes and text areas for displaying centrifugal compressor parameters. Creating applications includes the following main steps: the location of the desired element of the interface within the graphical window and programming events, which occur when the user accesses these objects, for example, when clicking a button. The end result is a program with the graphical user interface that is launched by specifying its name on the command line or in another MATLAB application.

To implement human-machine interaction the user interface was built in the MATLAB software package (Figs. 3, 4). This interface has two main windows. The first window is presented in Fig. 3. Simulation of the calculation of gas dynamics and thermodynamic parameters was performed for the centrifugal compressor with the following specified parameters presented in Fig. 3: volumetric flow of the 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 $q = 1.38$, individual gas constant $R = 287$ (J/kgK), air density $\rho = 1.07$ kg/m^3.

The window presented in Fig. 3 is used for entering parameters concerning the mode of operation of the centrifugal compressor. The second window is shown in Fig. 4. It is necessary for the analysis of the calculated parameters of gas dynamics and thermodynamics. The calculated and optimized values of the centrifugal compressor parameters are obtained in the form of data arrays, which can be further used in MATLABor third-party applications.



Fig. 3. The first window of the UI for entering the main parameters of centrifugal compressor.

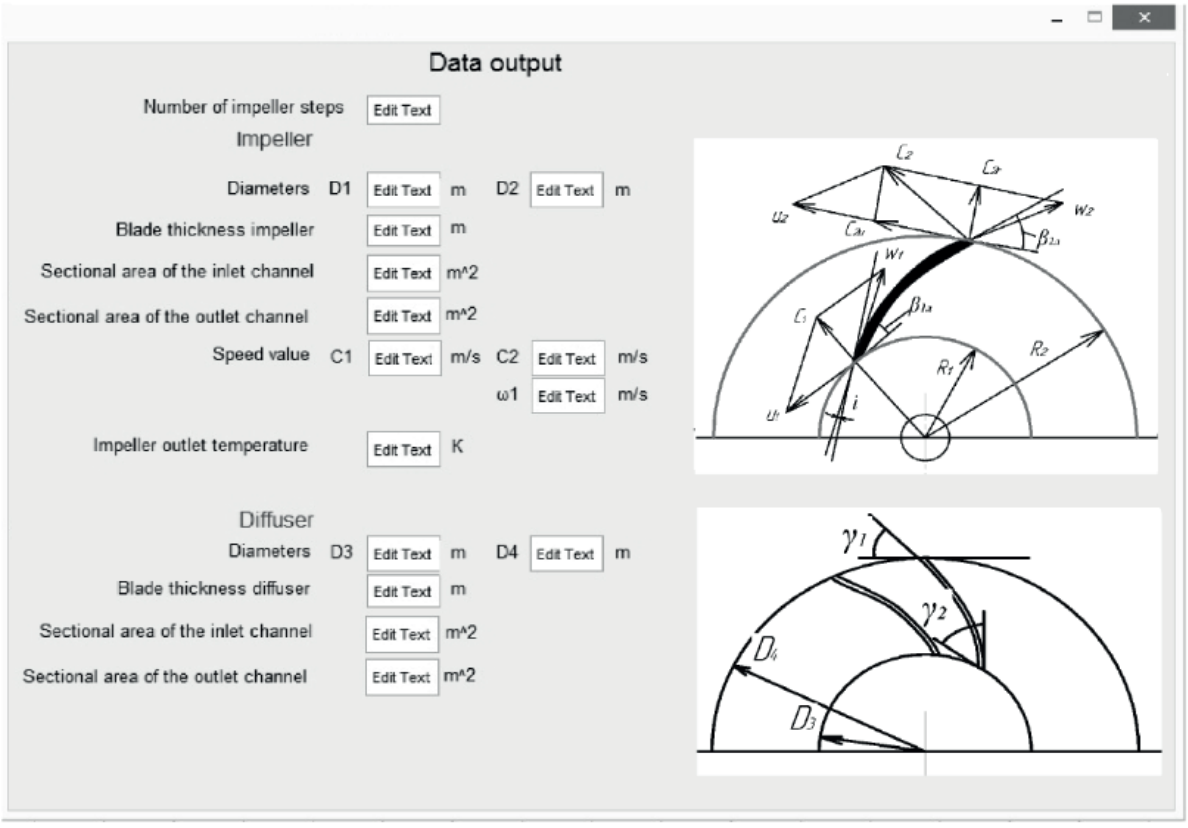


Fig. 4. The second window of the developed UI for the analysis of the calculated parameters of gas dynamics and thermodynamics in the centrifugal compressor.

III. RESULTS

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 exchange with the medium surrounding the compressor. The dependences of the densities and temperatures between the impeller and the diffuser are shown in Fig. 5.

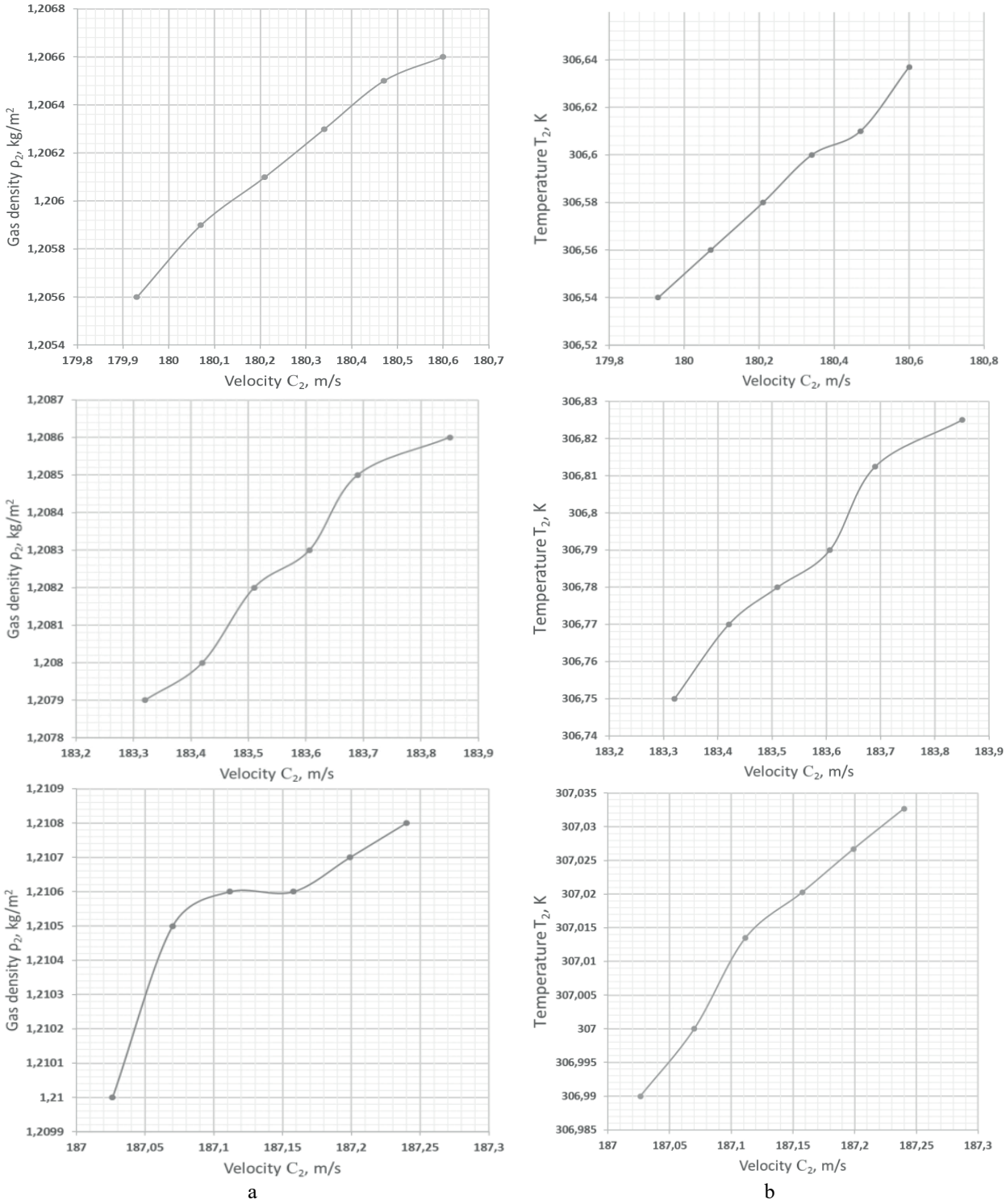


Fig. 5. Velocity C_2 dependences on: (a) Densities between the impeller and the diffuser; (b) Temperatures between the impeller and the diffuser.

The blades exert a force effect on the gas flow around them. To determine the optimal shape of the blades and their flow section, we will consider the conditions for gas entry into the blades. For this purpose, the centrifugal compressor stage was simulated with a change in the thickness of the impeller and diffuser blades (from 0.01 m to 0.02 m). Velocity components for different feed and blade thickness are summarised in Table I.

The design and gas-dynamic parameters of the impeller and diffuser have a significant impact on the efficiency of not only the entire stage. These parameters are not independent. The main parameters of the compressor impeller correspond to the optimal combination of the remaining design and gas-dynamic parameters. Therefore, the design mode of the impeller and diffuser for the given operating conditions is obtained with certain combinations of the values of these parameters. The dependence of the ratio of the impeller vanes and diffuser vanes to the flow area of the channels is shown in Fig. 6.

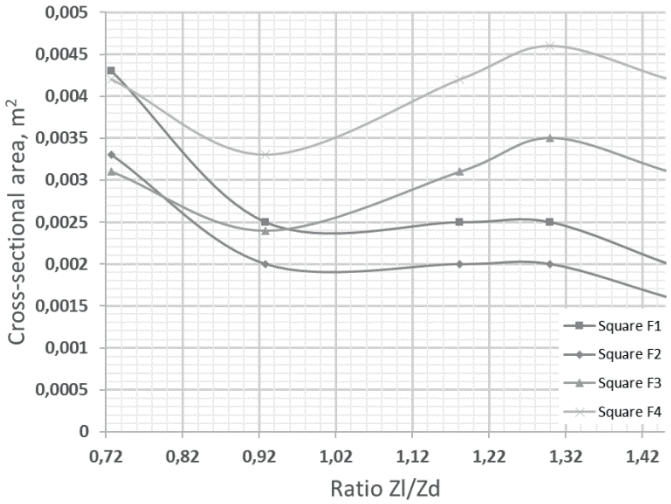


Fig. 6. The dependence of the ratio of the impeller vanes and diffuser vanes to the flow area of the channels.

TABLE I

SPEED COMPONENTS OF THE STAGE FOR 42 M³/MIN GAS FLOWRATE

Blade thickness δ, m	ω_1, m/s	ω_2, m/s	C_1, m/s	C_2, m/s	C_4, m/s	C_{1U}, m/s	C_{2U}, m/s
0.01	10.77	13.46	89.16	183.85	73.54	37.68	160.8
0.012	11.15	13.83	88.82	183.69	73.47	37.53	160.66
0.014	11.51	14.04	88.50	183.61	73.44	37.4	160.59
0.016	11.89	14.25	88.16	183.51	73.4	37.26	160.5
0.018	12.30	14.47	87.8	183.42	73.37	37.1	160.42
0.02	12.73	14.7	87.4	183.32	73.33	36.94	160.33

The correspondence between the parameters is established on the basis of the analysis of the physical flow diagram, theoretical calculations, recommendations arising from the generalized experimental data. The quantitative ratio of the blades is the defining design parameter of the centrifugal impeller and the diffuser. Therefore, the following studies must be carried out in the optimization of gas-dynamic parameters due to the criteria of the areas of the passage sections of the channels of the impeller and diffuser blades.

In our case, the required area of the channels of the impeller and the diffuser can be obtained by the ratio of the number of blades of the impeller and the diffuser. The number of the impeller and the diffuser blades affects the theoretical head. Along with this, the number of blades also affects the efficiency of the compressor. With a decrease and while maintaining a constant pressure, the load on each blade increases; the pressure and velocity gradients on the surface of the blades increase, and the intensity of the diffuser zones on the airfoil increases. At the same time, the opening angles of the interscapular canal increase. All this leads to the appearance of shear zone with a strong decrease in the number of impeller blades and, consequently, to a decrease in the efficiency of the wheel. On the other hand, as the number of blades increases, the surface area increases, and friction losses increase. From the above, it is clear that the optimal design of the centrifugal compressor stage to ensure efficiency and pressure will be achieved with a certain number of blades. The obtained data of modelling the influence of the number of blades on the area are shown in Fig. 7.

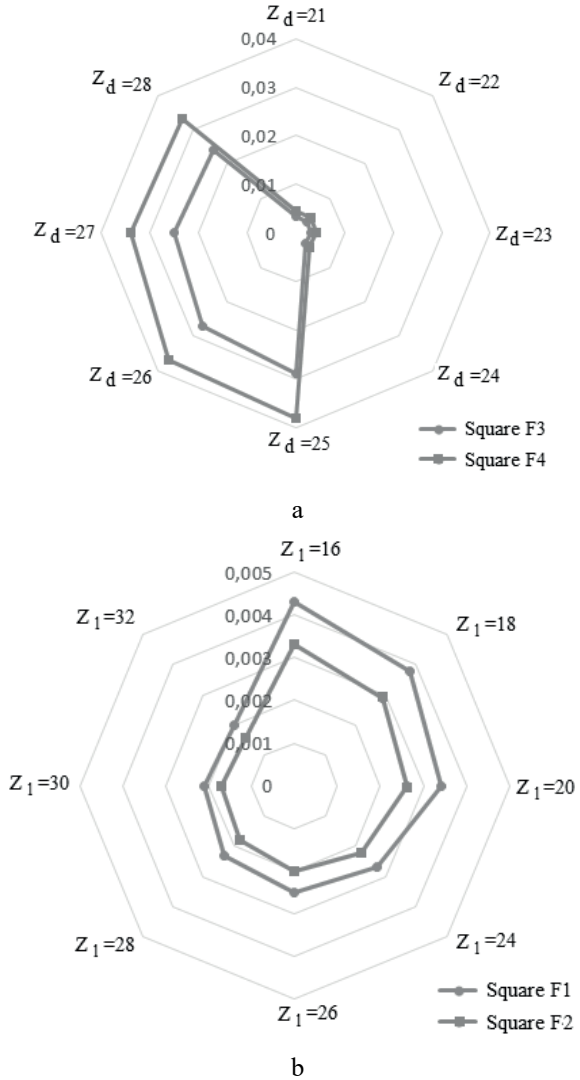


Fig. 7. Influence of the number of blades on the area of channels F_1, F_2, F_3, F_4: (a) with a fixed value of the impeller blades $Z_l = 26$ and a variable value of the diffuser blades; (b) with a fixed value of the diffuser blades $Z_d = 22$ and a variable value of the impeller blades.

At the same time, such important parameters of centrifugal compressor as Q or M feed of the stage; initial p_1 and final p_2 pressures directly depend on the composing speeds in the compressor.

The work spent on the rotation of the impeller is lost not only on increasing the potential pressure energy, but also on increasing the velocity energy of the gas. Speed reduction and conversion of speed energy into pressure energy is achieved in diffusers with as little losses as possible. On the other hand, the gas velocity directly affects the values of its flowrate. The dependences of the gas velocities on the ratio of the impeller and diffuser blades are shown in Fig. 8.

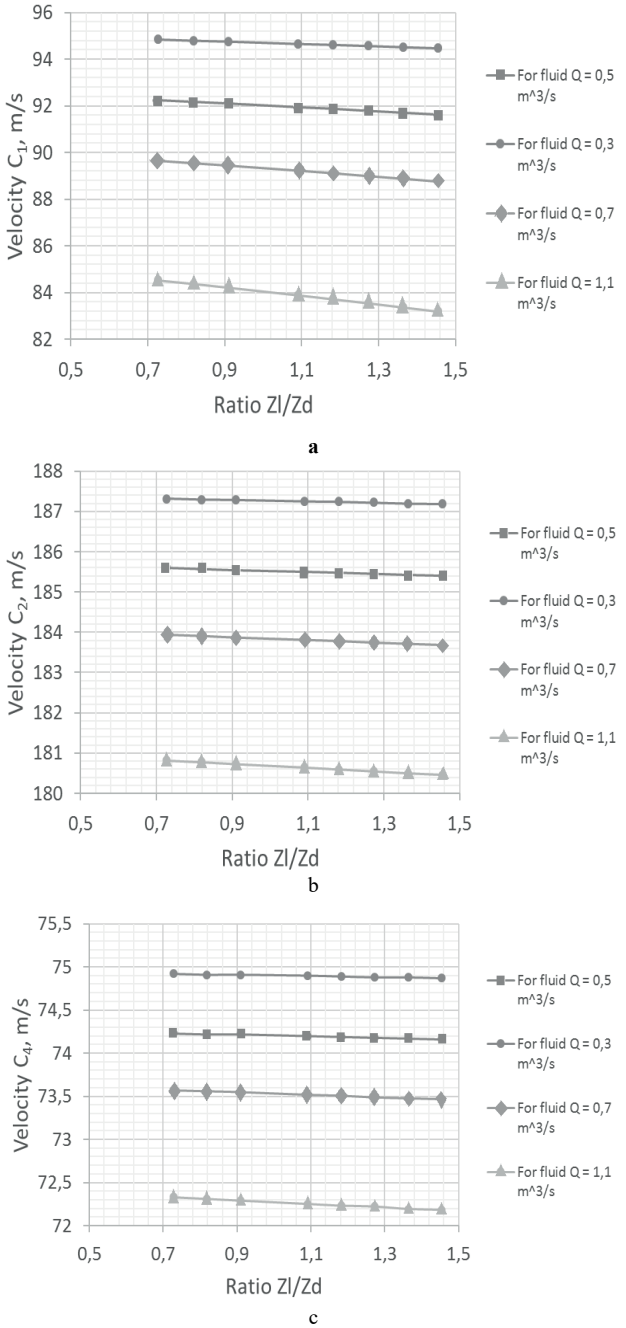


Fig. 8. Dependences of the gas velocities on the ratio of the impeller and the diffuser blades: (a) for the gas velocity at the inlet to the impeller; (b) for the gas velocity between the impeller and the diffuser; (c) for the gas velocity at the outlet of the diffuser.

The air parameters in the elements of the centrifugal compressor depend on the ratio of the impeller blades and the diffuser and change as follows. In front of the impeller, the air speed increases, the pressure and temperature drop. In the impeller, due to the energy consumption, the air is compressed and the speed of its movement increases, the air temperature rises both due to compression and due to friction heat. Gas pressure and temperature increase in the diffuser due to a decrease in the air flowrate. Better organization of the flow is achieved in the diffuser, as a result of which, despite the additional surfaces, the total losses decrease, and the efficiency increases.

IV. Discussion

Summarising the mentioned results, the following should be noticed. Vane diffusers make it possible to obtain maximum stage efficiency values, but in a narrow range of initial conditions (flowrate, gas composition, etc.). Vaneless diffusers provide high efficiency in a wide range of initial conditions, but this efficiency is lower than for stages with vane diffusers. If we install the drive to change the angle of inclination of the vane diffuser, then we get the maximum efficiency of the stage, which remains in a wide range of initial conditions. However, adjustable vane diffusers are extremely rarely used due to structural difficulties, but this type of regulation has a great future, especially when using jointly regulated vane diffusers and the section chamber (see Fig. 2), which makes it possible to obtain high efficiency over the entire range of operating performance. Such an increase in efficiency with combined regulation is 2.6 % in comparison with a stage equipped only with the section chamber [40]. The combined control methods also include the control of the simultaneous rotation of the diffuser blades and the change in the rotational speed, which are also widely considered in [41], [42].

Combined methods of control hide a great potential to improve consumer properties of industrial centrifugal compressors. The creation of complex mechanical devices and electrical systems, with additional manufacturing costs and maintenance difficulties, is required for such modern combined control methods. But most importantly, it is necessary to research and develop advanced complex algorithms for coordinated control of these devices and systems with elements of automatic search for extrema (maximum efficiency or minimum power consumption); and, on this path, the creators of the compressor technology of the future are waiting for interesting discoveries and solutions.

V. Conclusion

The study provides a systematic engineering methodology and information system for numerical calculations of the main geometry parameters of impellers and vanned diffusers as the main parts of centrifugal compressors. Firstly, basic features of centrifugal compressors are highlighted during a preliminary literature review. In particular, an impeller and a vane diffuser are defined as the most important and interrelated parts of the compressors. Secondly, the summarized engineering methodology of thermodynamic and gas-dynamic calculation of the blade impeller and the vane diffuser of the centrifugal

compressor is presented. It considers volumetric or mass feed of the stage, initial and final pressures, the initial temperature of the gas and thermodynamic characteristics of the gas under normal conditions. As a result, we obtain the main geometry parameters of the impeller and the diffuser. Thirdly, the multi-staged algorithm for the computer implementation of the given methodology is proposed. It determines rotation speeds and cross-sectional areas for the diffuser and the impeller. Fourthly, a computerized information system for pipe diffuser preliminary design is developed in accordance with the proposed algorithmic solution. The presented computer program has discreet intuitive to the user multi-window UI with only required fields and buttons. Finally, optimization function based on gradient descent method is added to the proposed computer system to find the best correspondence with gas, expressed through the tracking gas velocities in key places of the compressor (expressing corresponding pressures), volume flowrates, cross-sectional areas of impeller and diffuser, expressed through the ratio of the numbers of their blades. The resulting dependences and the obtained computational results generally reflect the real thermodynamic processes occurring in the centrifugal compressors for initial and optimized samples and can help increase the whole efficiency of compressors.

The visual computer implementation of the developed algorithm significantly simplifies computing and optimization of the main geometry parameters that influence both the impeller and diffuser during preliminary design. Moreover, the developed software reduces time spent on optimization, including a built-in efficient gradient method, and allows designing centrifugal compressors with improved quality thermodynamic parameters.

The scientific novelty of the paper lies in the proposed design approach and circuit implementation of the computer information and calculation complex for the centrifugal compressor development, which are additionally able to provide parametric optimization by high-level programming written in a single language. Optimization of centrifugal compressor design data in single software considering all its stages will allow expanding the range of measuring parameters by determining the geometric and thermodynamic parameters at each stage of the centrifugal compressor.

The practical significance of the obtained results is in developed software that implements the method of thermogasodynamic calculation of the centrifugal compressor with support for optimization of the geometric parameters of the blades, and experiments on the simulated centrifugal compressor. The results of the model experiment allow us to recommend the proposed achievements for use in practice, as well as to determine the optimized parameters of the centrifugal compressor to achieve better efficiency.

Prospects for further research are to expand capabilities of the program in the field of design of the multistage centrifugal compressors, optimization of the compressor parameters according to efficiency criteria, CFD-modelling of gas flows in centrifugal compressors, improvement of production technology of individual parts, and adjustment of blade angles to improve overall efficiency and productivity of centrifugal compressor.

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