ХОЛОДИЛЬНА ТЕХНІКА ТА ЕНЕРГОТЕХНОЛОГІЇ

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Fluid dynamic design of centrifugal compressor diffusers

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Diffusers are used to convert the kinetic energy of the gas flow into potential energy, i.e. to reduce the velocity and to increase the pressure. The most common types of diffusers are vaneless, vaned, and channel ones. Each type of diffuser has its design features and its performances. Improving the operational characteristics of vaneless diffusers is possible by using stepped diffusers with flow injection. The developed mathematical model of gas flow in a vaneless diffuser with flow injection provides design of the stage of centrifugal compressor with the wider range of stable operation. Traditional methods of designing vaned and channel diffusers of turbomachines are focused on the use of simple geometric lines and surfaces, such as a straight line, arc of a circle, plane, cylindrical surface, etc. The design is performed according to recommendations based on experimental data. The other way of designing the vaned and channel diffusers of turbomachines is related to solving the inverse problem of fluid dynamics, when the shape of the surfaces is determined by the given distribution of velocities along the surfaces of the channel. This paper describes the design principles for the centrifugal compressor diffusers based on physical and mathematical models of the flow of swirling viscous compressible fluid. According to the presented method, the designing diffusers are based on the preseparation condition of the boundary layer along one of the surfaces. Such a design ensures a reduction in separation zones in the channels of the diffusers and, accordingly, a reduction in total pressure losses and an expansion of the range of stable operation. A new method of designing the vaned and channel diffusers provides an improvement in the gasdynamic characteristics of diffusers compared to traditional geometry diffusers.

Keywords: *Centrifugal Compressor; Method of the Design; Vaneless Diffuser; Vaned Diffuser; Channel Diffuser; Research*

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1. Introduction

A significant part of the energy (40-60%) transmitted to the gas in the centrifugal compressor impeller is kinetic. Diffusers are used to convert the kinetic energy of the gas flow into potential energy, i.e. to reduce the velocity and to increase the pressure. The most common types of diffusers are vaneless, vaned, and channel ones. Each type of diffuser has its design features and its performances.

The vaneless diffuser (VLD) has the simplest design. VLD is the axisymmetric channel formed by two walls. Most often, the walls are flat, parallel to each other, and perpendicular to the axis of the rotor rotation.

The vaned diffuser (VD) is another type of diffuser. The profiled vanes are placed between the walls of the casing (disks) VD. Usually, the walls of the diffuser are parallel to each other.

The channel diffusers are rectangular or circular channels evenly spaced around the circumference.

The expediency of choosing the diffuser type for the centrifugal compressor is determined by the operating conditions of the compressor. The magnitude of the gas flow angle between the absolute and tangential velocities at the exit of the impeller is an important factor in choosing the diffuser type.

The advantages of the VLD are the simplicity of design and the ability to provide a wider range of the operating modes of the compressor stage compared to the stage with the VD.

The stages with vaned diffusers have smaller radial dimensions and higher values of the maximum efficiency in design mode compared to VLD. At the same time, the construction of the compressor becomes more complicated, and the range of steady operating modes gets narrow.

The channel diffusers are most often used in transport unit compressors, compressors with built-in gas coolers, in low mass flow rate machines with small impeller diameters at large values of the Mach numbers.

2. Design of vaneless diffusers

The gas flow in the vaneless diffuser, despite the simple shape of the channel, has a complex spatial nature. The flow entering the diffuser is formed by the rotating impeller of the compressor.

The flow coming from the impeller is unsteady in absolute motion, and the parameters change both along the channel width and in the circumferential direction. It is generally accepted that the flow scheme at the impeller outlet is of the «jet-wake» type. In the inlet region of the VLD there is the intense mixing of «jets» and «wake», which is accompanied by the loss of the total pressure [1]. In the inlet region of the diffuser the main role is played by mixing and friction losses, and in the main part – separation losses and friction losses.

Promising ways to expand the range of the steady operation of the centrifugal compressor stages with VLD are the use of the stepped diffusers, as well as a gas injection into the VLD.

<u>Vaneless diffuser with injection</u>

In the mathematical model of the fluid flow in the vaneless diffuser, the supply of the mass, momentum, moment of momentum, and energy due to injection is taken into account. The method is developed by analogy with [2] and allows you to calculate the average flow parameters along the radius of the diffuser.

The steady flow is considered, and the flow in VLD is considered to be axisymmetric. The fluid of the mainstream and the injection stream is the compressible ideal gas with constant specific heat.

The flow calculation method is based on the boundary layer theory [3,4,5], according to which the flow region is divided into the non-viscous core and near-wall boundary layers.

The calculation of the flow involves 2 stages: the calculation of the averaged flow parameters and the calculation of the parameters of the boundary layer.

The calculated dependences are obtained from the basic conservation laws recorded for the elementary control volume (Fig. 1).

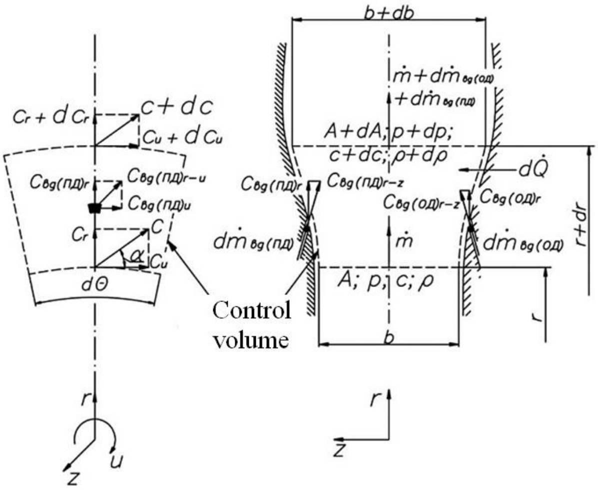


***Figure 1** – Control volume*

In the considered control volume gas is injected from the side of the diffuser main disk with mass flow dm_{inj1}, velocity C_{inj1} and temperature T_{inj1}. Similarly, gas is injected from the side of the cover disk with mass flow dm_{inj2}, velocity C_{inj2} and temperature T_{inj2}.

The system of differential equations consists of the continuity equation, the momentum equation, the angular momentum equation, the energy equation in the form of the first law of thermodynamics, the ideal gas state equation, the equation that establishes the relation between total and static temperature and the equation that establishes the relation between velocity components.

After transformations we have the system of equations:

$$\frac{d\rho}{\rho}+\frac{dC_{\mathrm{r}}}{C_{\mathrm{r}}}=\frac{d\dot{m}_1}{\dot{m}}+\frac{d\dot{m}_2}{\dot{m}}-\frac{dA}{A}; \qquad (1)$$

$$\frac{dC_{\mathrm{r}}}{C_{\mathrm{r}}}+\frac{p}{\rho\cdot C_{\mathrm{r}}^{2}}\cdot\frac{dp}{p}=\frac{dr}{r}\cdot\mathrm{ctg}^{2}\,\alpha-\frac{1}{2}C_{\mathrm{f}}\cdot\frac{1}{\sin\alpha}\cdot\frac{dA_{\mathrm{w}}}{A}-$$
$$-\left(1-\frac{C_{\mathrm{inj1r}}}{C_{\mathrm{r}}}\right)\cdot\frac{d\dot{m}_1}{\dot{m}}-\left(1-\frac{C_{\mathrm{inj2r}}}{C_{\mathrm{r}}}\right)\cdot\frac{d\dot{m}_2}{\dot{m}}; \qquad (2)$$

$$\frac{dT}{T}+\frac{C^2}{c_p\cdot T}\cdot\frac{dC}{C}=\left(1+\frac{C^2}{2\cdot c_p\cdot T}\right)\cdot\frac{dT^*}{T^*};\qquad(3)$$

$$\frac{dC}{C}-\sin^2\alpha\cdot\frac{dC_r}{C_r}=\cos^2\alpha\cdot\frac{dC_u}{C_u};\qquad(4)$$

$$\frac{dp}{p}-\frac{d\rho}{\rho}-\frac{dT}{T}=0.\qquad(5)$$

This system of equations with five unknowns $d\rho/\rho$, dp/p, dT/T, dC_r/C_r, dC/C is easy to solve by Cramer's method.

Numerical research of the vaneless diffusers

Calculations of the flow parameters in the vaneless diffusers with different meridional profiles with and without injection are performed using the software package developed based on the above method [6].

The performed numerical researches allow us to draw the following conclusions: the use of stepped VLD compared to VLD with parallel walls with the same relative width at the inlet and the same radial dimensions allows for reducing the total pressure loss.

This can be explained by the fact that the total pressure losses that occur in stepped diffusers due to the flow separation by sudden expansion are smaller than the friction losses in the VLD with the constant width. In addition, it was found that the flow separation in the wide part of the stepped VLD occurs at lower mass flow rates, which allows for expansion of the region of stable operation of the stages at low mass flow rate.

Since the flow with the injection is the process with the supply of energy, its evaluation must be performed taking into account the power expended on the injection. The power of the injected jet is determined by the kinetic energy:

$$N_{inj}=\dot{m}_{inj}\frac{C_{inj}^2}{2}=\dot{m}_{inj}\frac{C_{injr}^2}{2\sin^2\beta}\qquad(6)$$

Equation (1) shows that at the same flow of injected air, the power spent on injecting is minimal at $\beta = 90°$. To evaluate the characteristics of the VLD with injection, we use the loss coefficient and the static pressure rise coefficient, which are calculated taking into account the dynamic head of the injected air flow:

$$\zeta=\frac{p_3^*-p_4^*+\rho_{inj}C_{inj}^2/2}{p_3^*-p_3+\rho_{inj}C_{inj}^2/2}\qquad(7)$$

$$C_p=\frac{p_4-p_3}{p_3^*-p_3+\rho_{inj}C_{inj}^2/2}\qquad(8)$$

The results of calculating the coefficient of total pressure ζ and the static pressure rise coefficient C_p for VLD with parallel walls, stage VLD without injection, and with injection at different values m_{inj} and β are shown in Fig. 2.

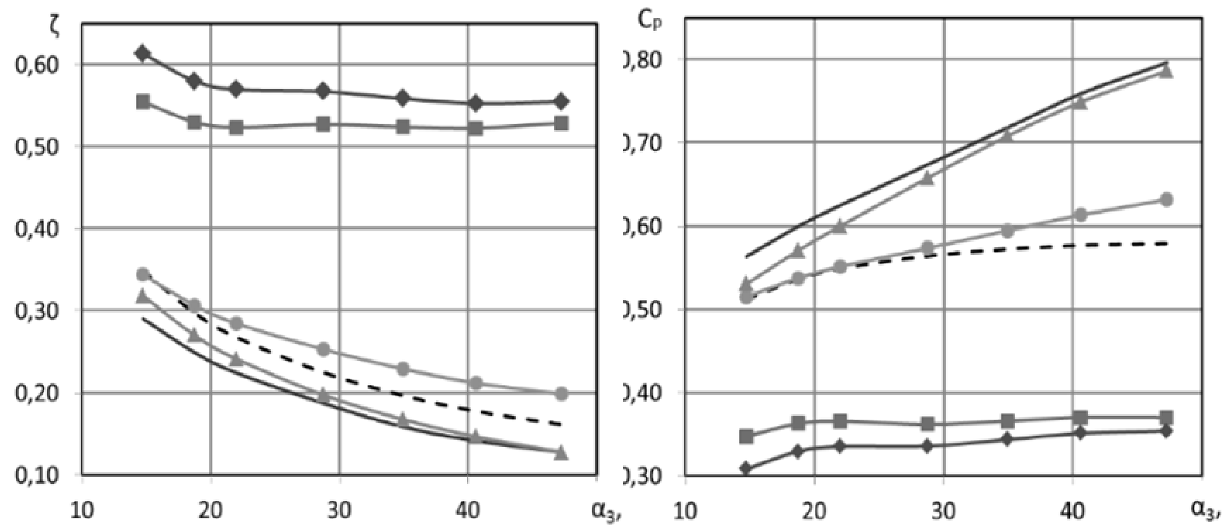


***Figure 2** – Performances of VLD with injection (m_{inj} = 0,04m):*
▬ ▬ VLD with parallel walls without injection; ▬▬ stepped VLD without injection;
▬■▬ β = 5°; ▬●▬ β = 15°; ▬▲▬ β = 90°; ▬ж▬ β = 165°; ▬♦▬ β = 175°;

Based on the results of the VLD calculations with injection, the following conclusions can be made:

– when using an injection in the VLD flow separation occurs at a lower mass flow rate, which confirms the possibility of expanding the range of stable operation of the stages by injection. The mass flow rate at which the return flow occurs decreases with increasing the mass flow rate, or with increasing the angle of the injected flow;

– in the case of using the injection in the VLD there are losses in the mixing of the main flow with the injected flow. Mixing losses increase with increasing the mass flow rate, or with increasing the injected flow angle. The lowest total pressure loss and the greatest flow stagnation are provided for radial injection.

Experimental research of vaneless diffusers

The experimental research was conducted at the aerodynamic setup of AT 400 research complex of «Sumy Machine–Building Research and Production Association» [7]. The setup is made according to the open loop, the working environment is air. To research the effect of injection on the performances and nature of the flow in the vaneless diffuser, the construction of the stage was changed. The design of the diffuser after reconstruction is shown in Fig. 3.

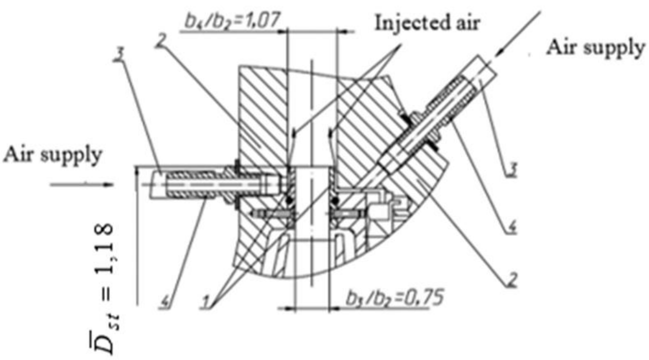


***Figure 3** – Sketch of the diffuser construction with injection: 1 – nozzle apparatus; 2 – diffuser disk; 3 – pipe; 4 – fitting*

At the diffuser inlet, ring 1 is installed, which allows you to narrow the diffuser at the inlet to the value of the relative width $\bar{b}_3$ = 0,045 (b_3/b_2 = 0,75). The relative outer diameter of the ring $\bar{D}_{st}$ = 1,18. At $\bar{D}_{st}$ = 1,18 there is the sudden expansion of the diffuser channel to the width equal to the width of the diffuser at the outlet $\bar{b}_4$ = 0,065 (b_4/b_2 = 1,07). On the surfaces of ring 1 adjacent to the disks of diffuser 2, the grooves are made that act as injectors. 90 grooves, evenly spaced in the circle, are made on each of the ring. The width of the grooves is 4 mm, and depth is the 1,5 mm.

The flow separation in the vaneless diffuser is caused by the pressure gradient along the radius. To shift the separation of the flow in the region of lower mass flow rate, the injection is carried out in the radial direction.

The air used for the injection into the VLD is supplied through two independent tubes: one leads the air to the main diffuser disc (right), and the other one leads to the cover disc (left). This allows different injection options to be explored from the side of each of the discs.

Fig. 4 shows how the use of injection from the cover side allowed the compressor to be taken out of the surge zone.

During the experiment, the control valve was smoothly closed until the pre-surge pressure pulsations appeared on the diaphragm.

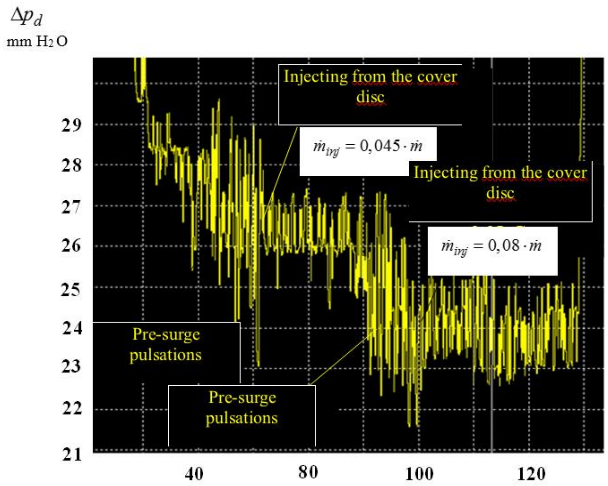
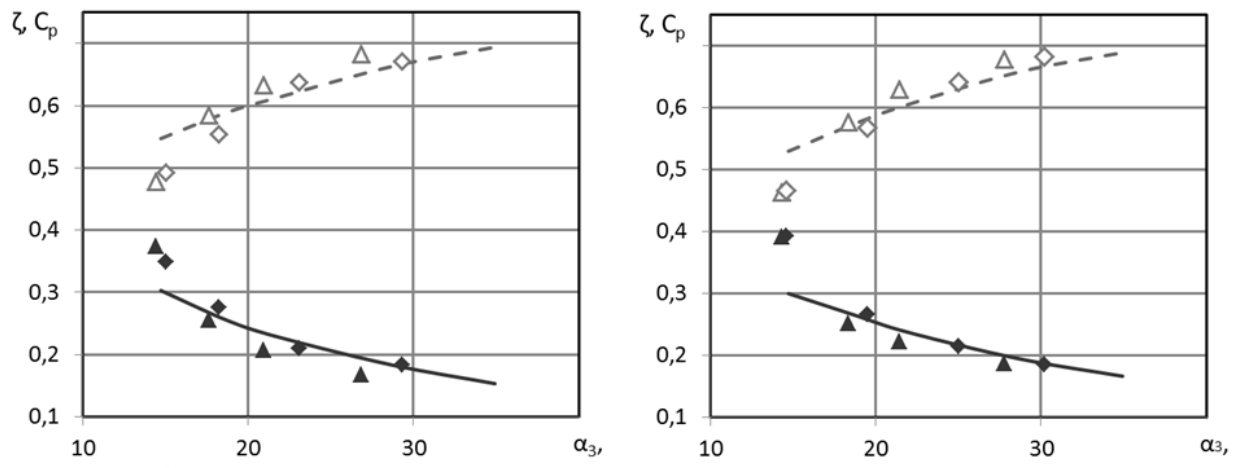


***Figure 4** – Pulsation of the pressure drop at the diaphragm at different mass flow rates of the injected air from the side of the cover disk, $m_{inj\ cd}$ = var .*

After their detection, the control valve was opened and the mass flow rate value of the injected air was set to 4,5% of the capacity of the stage at the design mode. As the result, the amplitude of the pressure drop oscillations was decreased, and the operation of the stage was stabilized. Then the mass flow rate of the stage was again reduced to the occurrence of pre-surge pulsations, after which the flow of injected air was increased to 8,0%, and the operation of the stage was stabilized again.

Fig. 5 shows the calculated and experimental performances of the stepped VLD for different injection modes.

The discrepancy between the calculated and experimental data for the values of the flow angles at the inlet to the VLD does not exceed 7 %. At lower values of the flow angles in the diffuser, there are large sepa-



a) b)

***Figure 5** – Calculated and experimental performances of stepped VLD at different injection modes: a) m_{inj} = 0,045m; b) m_{inj} = 0,08m; ▬▬ ζ (theory); ▬ ▬ C_p (theory); ▲ ζ (experiment, injection from the main disk); ♦ ζ (experiment, injection from the cover disk); Δ C_p (experiment, injection from the main disk); ◊ C_p (experiment, injection from the cover disk);*

tion zones, and the unsteady nature of the flow increases, resulting in a significant increase in losses. In general, the comparison of the calculated and experimental data confirms the correspondence of the mathematical and computer models to the actual processes of fluid flow in the VLD.

3. Design of vaned diffusers

Stages with vaned diffusers in comparison with stages with vaneless diffusers have smaller radial dimensions and higher values of maximum efficiency in design mode. But the construction of the compressor becomes more complicated, and the range of the stable operating modes is narrowed.

It is better to use stages with VD in the case when the centrifugal compressor operates in the narrow range of the mass flow rate near the design mode. At the same time, we can expect an efficiency that is 2-3 % higher than in the case of using VLD.

The average flow velocity in the channel is determined by the formula

$$C_4 = C_3 \cdot \frac{A_3}{A_4} \cdot \frac{\rho_3}{\rho_4} \cdot \frac{\sin\alpha_3}{\sin\alpha_4}, \quad (9)$$

where A_3 and A_4 are the areas at the inlet and outlet of the channel, which are determined by the formula $A = \pi \cdot D \cdot b$. Then formula (9) takes the form

$$C_4 = C_3 \cdot \frac{D_3}{D_4} \cdot \frac{b_3}{b_4} \cdot \frac{\rho_3}{\rho_4} \cdot \frac{\sin\alpha_3}{\sin\alpha_4}. \quad (10)$$

In the vaned diffuser, the decrease in velocity can be achieved not only by increasing the diameter but also by increasing the flow angle α. The flow angle can be increased due to the influence of the diffuser vanes on the gas flow.

In traditional diffuser design, the centerline of the vanes is the arc of the circumference. In this case, the shape of the vane does not depend on the flow parameters, the properties of the working fluid, and the solidity. For almost all modes of operation in such VD, flow separation is observed on the surfaces of the vanes.

The use of the diffuser design methods based on solving the inverse problem of hydrodynamics using the velocities distribution, which is set on the surfaces of the vanes, makes it possible to achieve better performances in comparison with diffusers designed by the geometric method.

Mathematical model of fluid flow in the vaned diffuser with the given velocity distribution

For developing methods for designing vaned diffusers the ideas of Stratford's [8].

The mathematical model for solving the inverse problem of fluid dynamics has been developed for the steady, adiabatic, unseparated flow.

The moment of momentum equation for the flow stream has the form

$$\Delta M = \Delta b \cdot z_v \cdot \int_{r_{in}}^{r} \Delta p \cdot r \cdot dr, \quad (11)$$

where ΔM is the change of the moment of momentum concerning an axis z for the stream with the mass flow rate Δm; Δb is the stream flow width; z_v is the number of vanes; $\Delta p = p_{ps} - p_{ss}$ is the pressure difference on the vane surfaces.

The change in the moment of momentum is equal to

$$\Delta M = \Delta \dot{m} \cdot \left(r_{in} \cdot C_{in} \cdot \cos \alpha_{in} - r \cdot C \cdot \cos \alpha \right), \qquad (12)$$

where r_{in}, C_{in}, α_{in} are the radius, velocity, and flow angle at the diffuser inlet; r, C, α are the current values of radius, velocity, and flow angle.

After transformations of equation (11) using gasdynamic functions, we obtain

$$r_{\text{in}} \cdot \lambda_{\text{in}} \cdot \cos \alpha_{\text{in}} - r \cdot \lambda \cdot \cos \alpha =$$
$$= \frac{\Delta b \cdot z_{\text{v}} \cdot p_{\text{in}}^{*} \cdot \sigma}{\Delta \dot{m} \cdot a_{\text{cr}}} \cdot \int_{r_{in}}^{r} \left[\pi\left(\lambda_{\text{ps}}\right) - \pi\left(\lambda_{\text{ss}}\right) \right] \cdot r \cdot dr. \qquad (13)$$

In equation (4.33) $\lambda = C/a_{cr}$ is the reduced velocity; a_{cr} is the critical speed of sound; p_{in}^{*} is the stagnation pressure at the diffuser inlet; $\sigma = 1 - (1 - \sigma_{\text{out}}) \cdot \dfrac{r - r_{\text{in}}}{r_{\text{out}} - r_{\text{in}}}$ is the total pressure loss factor; $\pi(\lambda) = \left(1 - \dfrac{k-1}{k+1} \cdot \lambda^2\right)^{\frac{k}{k-1}}$ is the gasdynamic function of pressure; k is the adiabatic index.

The flow equation for the diffuser element with the width Δb is as follows

$$\Delta \dot{m} = C_{\text{r}} \cdot \rho \cdot 2\pi \cdot r \cdot \Delta b \cdot \tau, \qquad (14)$$

where τ is the blockage factor.

The blockage factor can be determined by the formula

$$\tau = 1 - \frac{\delta \cdot z_{\text{v}}}{2\pi \cdot r \cdot \sin \alpha}, \qquad (15)$$

where $\delta' = \delta_v + \Sigma\delta^*$ is the reduced vane thickness; $\Sigma\delta^*$ is the total displacement thickness of the boundary layers in the vane passage.

The flow equation using gasdynamic functions is:

$$\Delta \dot{m} = \lambda \cdot \varepsilon(\lambda) \cdot a_{\text{cr}} \cdot \rho^{*} \cdot 2\pi \cdot r \cdot \Delta b \cdot \tau \cdot \sin \alpha, \qquad (16)$$

where $\varepsilon(\lambda) = \left(1 - \dfrac{k-1}{k+1} \cdot \lambda^2\right)^{\frac{1}{k-1}}$ is the gasdynamic function of density; ρ^* is the stagnation density.

Equation (13) and equation (16) form the system of equations with unknowns α, λ, λ_{ps}, λ_{ss}. The reduced velocities on the pressure (λ_{ps}) and suction (λ_{ss}) surfaces of the vane and the average velocity are interconnected. For example, for the linear law, the pressure changes along the cascade pitch are $\pi(\lambda_{ps}) + \pi(\lambda_{ss}) = 2\pi \cdot \lambda$.

With the given velocity value on the pressure surface of the vane, the velocity value on the suction surface of the vane is determined by the ratio

$$\pi\left(\lambda_{\text{ss}}\right) = 2 \cdot \pi(\lambda) - \pi\left(\lambda_{\text{ps}}\right). \qquad (17)$$

Thus, the system of two equations has two unknown values – the average velocity λ and the flow angle α, that can be determined by any numerical method.

The important part of the vaned diffuser design method is to determine the optimum velocity distribution. The velocity distribution is determined from the condition of the preseparation state of the turbulent boundary layer.

Determination of the velocity distribution that provides the preseparation state of the turbulent boundary layer is performed using the momentum equation.

The moment of momentum equation for the boundary layer (Karman equation), has the form

$$\frac{d\delta^{**}}{dl} + \frac{1}{C}\frac{dC}{dl}\delta^{**}\left(2 + H\right) + \frac{1}{\rho}\frac{d\rho}{dl}\delta^{**} = \frac{\tau_{\text{w}}}{\rho C^2}, \qquad (18)$$

where $H = \delta^*/\delta^{**}$ is the formparameter; C, ρ are the velocity and density at the boundary layer at ; $\gamma = \delta$.

To determine the separation, the criteria for separation are as follows:

$$f = \frac{dC}{dl}\frac{\delta^{**}}{C} G\left(\text{Re}^{**}\right), \qquad (19)$$

where $\text{Re}^{**} = C \cdot \delta^{**}/\nu$.

The condition of separation is $f \le f_{cr}$. The Lojcjanskij method was adopted according to which $G(\text{Re}^{**}) = (\text{Re}^{**})^{1/6}$ and $f_{cr} = -0{,}02$, $\text{Re}^{**} = C \cdot \theta/\nu$ [3].

The relative momentum thickness of the boundary layer is determined from the ratio:

$$\bar{\delta}^{**} = 0{,}0159 \cdot \text{Re}^{-0{,}15} \cdot \lambda^{-3{,}55} \cdot \left(\int_0^{\bar{l}} \lambda^4 \cdot d\bar{l} \right)^{0{,}85}, \qquad (20)$$

where $\text{Re} = C \cdot l/\nu$ is the Reynolds number.

The formula according to which it is expedient to determine the velocity distribution that provides the preseparation state of the turbulent boundary layer can be obtained by the method of integrating the momentum equation. The momentum equation using the unseparated flow condition (19) according to [3] should be converted to the form

$$\frac{d}{dl}\left[f\cdot\frac{C}{C'}\right]=(1+m)\zeta-\left[2+m+(1+m)H\right]\cdot f\,, \qquad (21)$$

where $m=\dfrac{\mathrm{Re}^{**}\cdot G'\left(\mathrm{Re}^{**}\right)}{G\left(\mathrm{Re}^{**}\right)}$; $\zeta=\dfrac{\tau}{\rho C^2}$.

The formula according to which it is expedient to determine the velocity distribution that provides the preseparated state of the turbulent boundary layer has the form

$$\lambda=\lambda_1\cdot\left[1+\frac{\left(\bar{l}-\bar{l}_1\right)\cdot\left(2+H_s\right)\cdot\left(-f_{\mathrm{cr}}\right)}{\bar{\delta}_1^{**}}\right]^{-\frac{1}{2+H_s}}. \qquad (22)$$

Formula (22) can be used to determine the values of velocities at the outer edge of the boundary layer. The resulting velocity distribution provides the unseparated flow with a maximum of the diffuser factor.

Design of vaned diffuser with the given velocity distribution

The diffuser vane profile is determined for the given meridional contour. Most VD has a constant width. The diameters on which the inlet and outlet edges of the vanes are located and the value of the average flow angle at the inlet are determined by preliminary calculations.

The above diffuser design methodology can be used for a number of different design options.

Design of single-row vaned diffuser

The system of equations (13) and (16) is solved by the method of successive approximations. The integral in equation (13) can be found by any numerical method, such as the method of central rectangles.

To determine the velocities on the pressure surface of the vane, it is necessary to the formula (14). Then with the help of formula (17) from equation (13), it is possible to eliminate the unknown λ_{ss}.

The result of the joint solution of equations (13) and (16) for the cross-sections from the inlet to the outlet of the VD is the distribution of the average flow angles and average velocities along the vane. Based on these data, you can determine the geometric parameters of the vane and the distribution of velocities on its surfaces.

Displacement thickness is determined by the ratio $\delta^* = H_s\cdot\delta^{**}$. The values of the momentum thicknesses are determined by the formula (20) and then the values of the form parameters on the vane suction surface are determined by the formula (19).

For diffuser vanes designed this way, flow separation can be observed at the suction surface of the vanes, but separation occurs much farther from the entrance edges of the vanes than in diffusers with vanes with the traditional profile. The losses of the total pressure will be less.

Increasing the vane solidity (increasing the number of vanes) reduces the load on the vane and shifts the separation point to the exit of the channel.

Design of vaned diffuser with maximum velocity limitation

This method of vanes design can be used for diffusers with a high value of the flow velocity at the inlet to the diffuser (λ_{in} = 0,6-0,8).

At such flow velocities at the inlet to the diffuser on the suction surface of the vanes, the velocity can reach the speed of sound (see Fig. 6, a). To prevent this, the velocity at the suction surface of the vanes should be limited, for example $\lambda_{\max} = \lambda_{in}$ (Fig. 6, b).

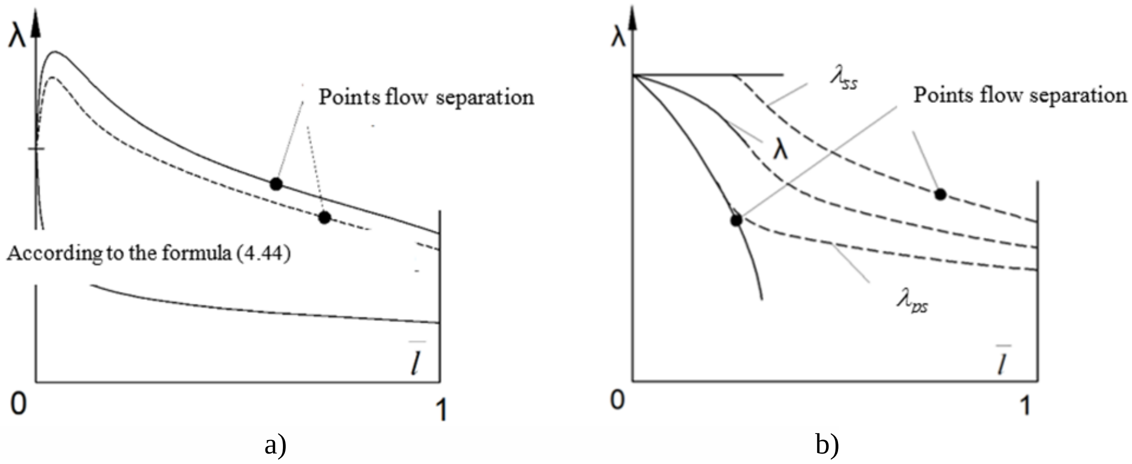


Figure 6 *– Distribution of the velocities along the diffuser vane surfaces*

The average velocity and velocity on the pressure surface of the vanes are determined then. The coordinate of the separation point on the pressure surface of the vane is determined. Starting from the coordinate located in front of the coordinate of the flow separation point, the velocity distribution on the vane pressure surface is calculated by the formula (14).

By solving equations (13) and (16) for the number of cross-sections, we obtain the distribution of the average flow angles and velocities along the vane. Next, the calculation is performed as shown above. In the diffuser designed this way, the flow velocities in the cascade do not exceed the value of the velocity at the inlet to the diffuser.

Design of double-row vaned diffuser

Diffuser with a completely unseparated flow can be designed. To do this, it is necessary to reduce the outlet diameter of the vanes until the flow becomes unseparated. But in this case, the flow velocity at the diffuser outlet will be higher than the required one, i.e. the diffuser factor of the flow will decrease. To further reduce the flow velocity, the second row of vanes must be designed. The second row of vanes is designed similarly to the first.

The scheme of the double–row vaned diffuser and the distribution of velocities along the surfaces of the vanes are shown in Fig. 7. The velocity distribution for the preseparated state of the boundary layer on the vane pressure surface is determined by the formula (14) taking into account that the flow velocity at the inlet to the second row is equal to the flow velocity at the exit from the first row.

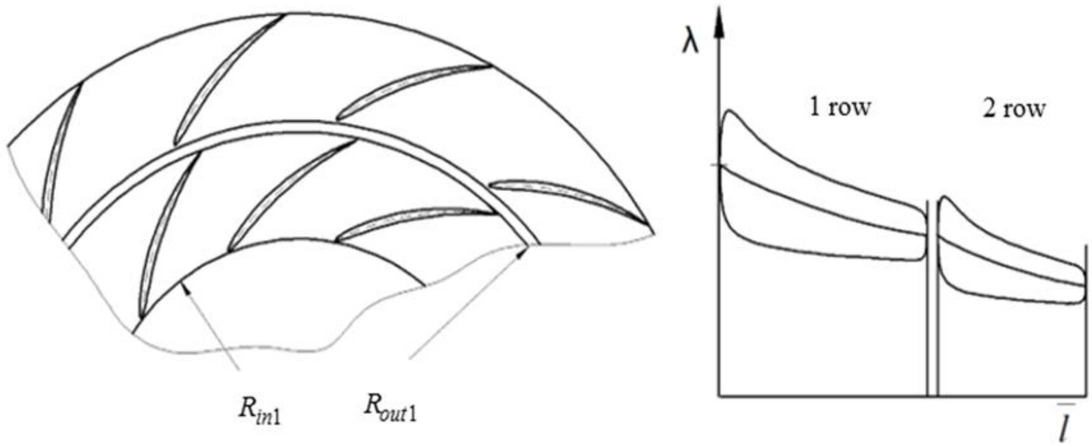


Figure 7 *– Scheme of the double–row vaned diffuser and the distribution of velocities along the surfaces of the vanes*

The thin boundary layer and correspondingly small values of the momentum thicknesses at the inlet section of the second row allow large velocity gradients without flow separation.

Research of vaned diffusers with the given velocity distribution

Simulation of gas flow in the flow path of the centrifugal compressor stage and determination of diffuser efficiency was performed in the ANSYS CFX software package for Reynolds numbers Re >105 and Mach numbers 0,2…0,6. The calculations were performed for 8 modes of operation of stages, which differed by mass flow.

Figure 8 shows the vane profiles for diffuser designed according to the given distribution of velocities and diffuser, in which the center line of the vanes is the arc of the circumference are presented.

Figure 9 shows the values of total pressure loss (ζ) and static pressure rise (C_p) coefficients for diffusers designed by different methods.

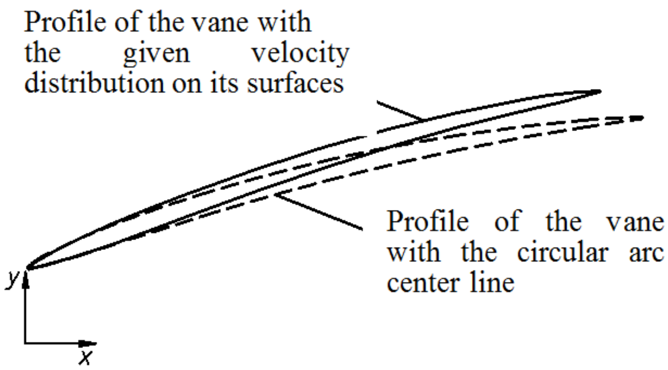


Figure 8 *– Profiles of vanes*

Vaned diffuser, designed with the preseparation velocities distribution, has better performance in nominal mode (Φ_0 = 0,0502) up to 28%, as well as in all modes with flow rate higher than nominal up to 54%. The static pressure rise coefficient of the vaned diffuser, designed using the proposed method, has a higher value at nominal mode of 9% and all modes with the flow rate of more than 21%. The prospect of the proposed vaned diffuser profiling method with the given velocity distribution on the vanes surfaces is obvious.

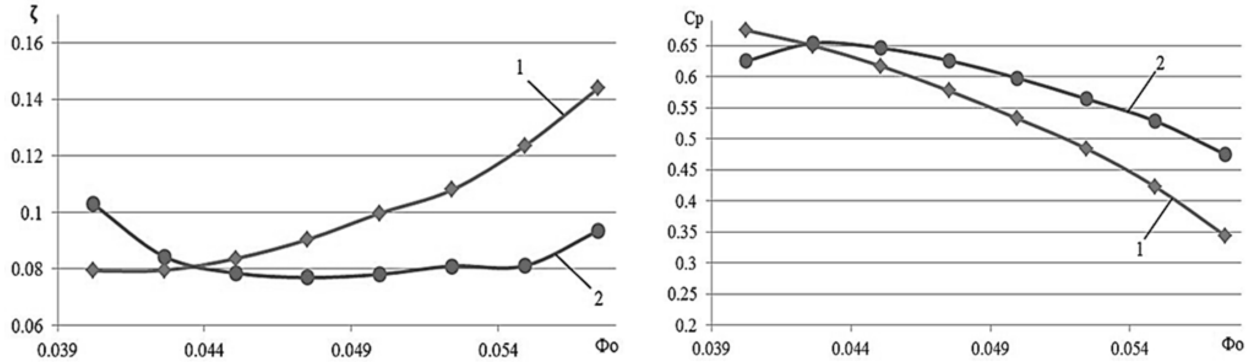


Figure 9 *– Dependences of loss coefficients (ζ) and static pressure rise coefficients (C_p) on flow coefficient $Ф_0$: 1 – the center line of the vane is the arc of a circle; 2 – given velocity distribution on the surface of the vane*

<u>Experimental research of vaned diffusers</u>

Experimental research was conducted at the aerodynamic setup of AT 400 research complex of «Sumy Machine-Building Research and Production Association». The scheme of the experimental model is shown in Fig. 10. The model includes confuser 1, axial channel 2, impeller 3, the vaned diffuser 4, crossover 5, return channel 6, axisymmetric channel 7, an outlet device 8.

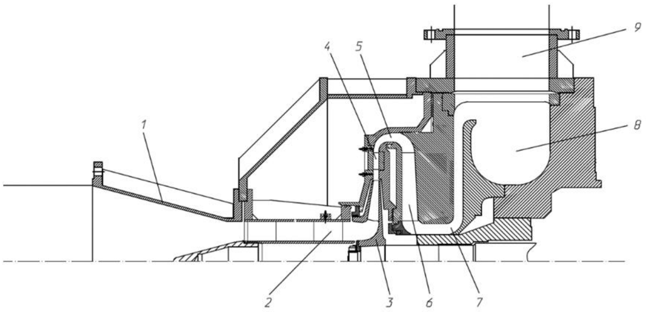


Figure 10 *– Scheme of the experimental model*

The confuser 1 is designed to connect the pipeline Ø 400 mm with the axial channel 2 of the model stage. The impeller 3 is located behind the axial channel. The reduction of the flow between the casing and the impeller cover is ensured by the labyrinth seal. The vaned diffuser 4 is located behind the impeller.

Fig. 11 shows the calculated and experimental gas-dynamic performances of the centrifugal compressor stage.

The difference between the values of the polytropic efficiency and the coefficient of the polytropic head of the stage is due to the assumptions made during the simulation of gas flow in the stage.

Comparison of experimental and design characteristics of diffusers and flow parameters in diffusers confirm the high efficiency of the diffuser, which was designed using the presented method and the adequacy of mathematical models with the physical processes of fluid flow in diffusers.

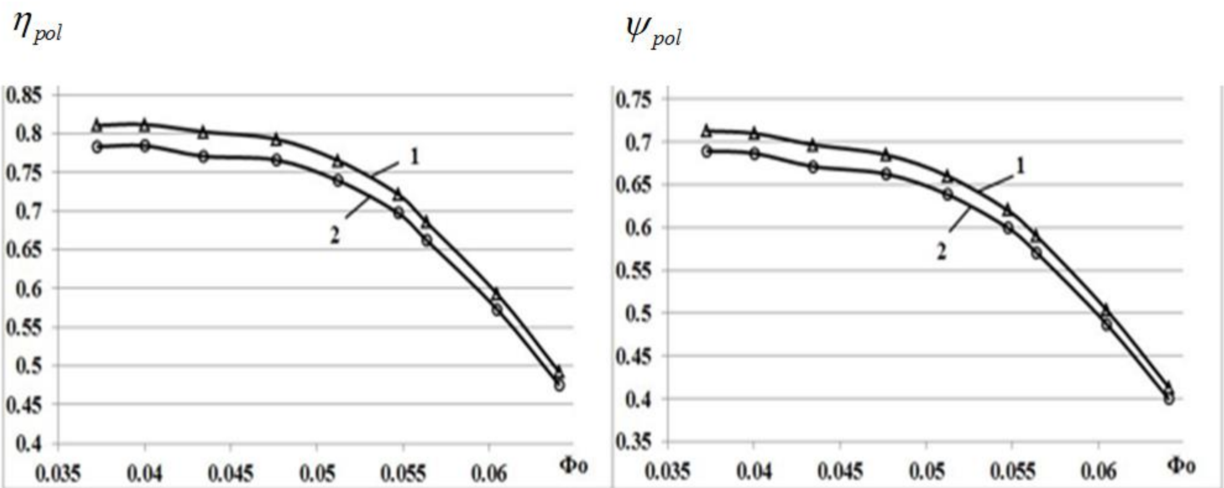


Figure 11 *– Comparison of gasdynamic performances of the stage: a — the polytropic efficiency (η_{pol}); b — the coefficient of the polytropic head (ψ_{pol}); 1 — numerical simulation; 2 — physical experement*

The new design method of vaned diffusers with the given velocity distribution on the surfaces of the vanes provides improved gasdynamic performances of diffusers compared to diffusers with traditional geometry. The values of the total pressure loss factors decrease by 28-50%, and the values of the static pressure rise factors increase by 9-21% for the wide range of operating modes.

4. Design of channel diffusers

The use of the channel diffusers (CD) in comparison with other types of diffusers is more effective under the following conditions: 1) small values of flow angles at the diffuser inlet; 2) supply of the gas from the segment passage to separate chambers; 3) if the diffuser channels go into the channels of the return channel.

The traditional geometry of the channel diffusers provides an inlet part made in the logarithmic spiral on one of the sides of the segment, behind which is the part with straight walls. The flat side surfaces of the channels of such diffusers simplify their manufacture, but have a significant disadvantage in terms of fluid dynamics. The gas flow enters the diffuser after the impeller is swirled, so the design of the channels without taking into account the swirling of the flow is erroneous. In the straight-line diffusers, due to significant values of the pressure gradients, the flow separation occurs at the diffuser inlet part, which leads to an increase in losses.

Designing the channel diffusers according to the given distribution of the flow velocities will reduce losses in them [9]. Of course, channels designed in this way can be more efficient than channels designed by the traditional method.

<u>Mathematical model of fluid flow in channel diffuser with the given velocity distribution</u>

The mathematical model of the fluid flow in the channel diffuser with the given velocity distribution, as well as for the vaned diffuser, is developed for steady, adiabatic, unseparated fluid flow.

The system of equations, the solution of which allows us to find the shape of the profile of CD segments, has the form

$$\begin{cases} r_{\text{in}} \cdot \lambda_{\text{in}} \cdot \cos\alpha_{\text{in}} - r \cdot \lambda \cdot \cos\alpha = \dfrac{b \cdot z_{\text{s}} \cdot p_{\text{in}}^* \cdot \sigma}{\dot{m} \cdot a_{\text{cr}}} \cdot \displaystyle\int_{r_{\text{in}}}^{r} \left[\pi(\lambda_{\text{ps}}) - \pi(\lambda_{\text{ss}})\right] \cdot r \cdot dr, \\ \dot{m} = \lambda \cdot \varepsilon(\lambda) \cdot a_{\text{cr}} \cdot \rho^* \cdot 2\pi \cdot r \cdot b \cdot \tau \cdot \sin\alpha. \end{cases} \tag{23}$$

In the given formulas the total pressure loss factor

$$\sigma = 1 - (1 - \sigma_{\text{out}}) \cdot \frac{r - r_{\text{in}}}{r_{\text{out}} - r_{\text{in}}}, \; \sigma_{\text{out}} = p_{\text{out}}^* / p_{\text{in}}^* . \tag{24}$$

The blockage factor is determined by the formula

$$\tau = 1 - \frac{\delta' \cdot z_{\text{s}}}{2 \cdot \pi \cdot r \cdot \sin\alpha}, \tag{25}$$

where $\delta' = \delta_s + \Sigma\delta^*$ is the conditional segment thickness (determines the thickness of the diffuser segment, provided that it flows around the non-viscous flow); δ_s is the segment thickness; $\Sigma\delta^*$ is the total displacement thickness of the boundary layers on the pressure and suction surfaces of the segment.

On the pressure surfaces of the segments, the law of change of velocity is given for the preseparation state of the turbulent boundary layer according to (22).

The law of change of pressure in the circular direction in the channel and the linear law of change of the average angle of flow along the channels are used $\alpha = f(r)$. Then, as the result of solving the system of equations (23) the change in the average flow velocity and the blockage factor along the radius of the diffuser are determined. The calculation is performed using numerical methods.

The calculation of the momentum thickness in the boundary layer on the surfaces of the segments is performed by the (20), and the value of the formparameter is calculated by the (19). After calculating the boundary layer displacement thicknesses in the channel, the thickness of the diffuser segments is determined by (25). The end result of the design calculation is to get dependencies $\delta_s = f(r)$. Assuming the angle of the midline of the segments is equal to the average angle of the flow ($\alpha_s = \alpha$), we have the completely specified shape of the segments of the channel diffuser.

Losses in the CD are determined based on the calculation of the boundary layer. The loss of total pressure in the diffuser channel due to friction and mixing of the flow is determined by the formula

$$\zeta_{\text{c}} = \frac{\left(\dfrac{\sum\delta_{\text{out}}^*}{W \cdot \sin\alpha_{\text{out}}}\right)^2 + \dfrac{2 \cdot \sum\delta_{\text{out}}^{**}}{W \cdot \sin\alpha_{\text{out}}}}{\left(1 - \dfrac{\sum\delta_{\text{out}}^*}{W \cdot \sin\alpha_{\text{out}}}\right)^2} \cdot \left(\frac{\sin\alpha_{\text{in}}}{\sin\alpha_{\text{out}}}\right)^2, \tag{26}$$

where $\Sigma\delta^*_{\text{out}}$ and $\Sigma\delta^{**}_{\text{out}}$ are respectively the total displacement thickness and momentum thickness on the pressure and suction surfaces of the segments at the outlet of the CD; W is the circular distance between the pressure and suction surfaces of the segments at the inlet of the CD.

Total pressure losses due to the decline in static pressure at the trailing edges (edge losses) are determined by the formula

$$\zeta_{edg} = -C_{edg} \cdot \frac{\dfrac{\delta_{s\,out}}{\sin\alpha_{out}}}{\left(W + \dfrac{\delta_{s\,out}}{\sin\alpha_{out}}\right)}, \quad (27)$$

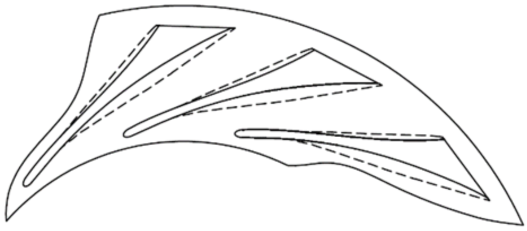
where $C_{edg} = (p_{edg} - p_{cor})/0{,}5p_{cor}C^2_{cor}$ is the coefficient that takes into account the pressure difference behind the trailing edge and in the flow core; δ_{out} is the thickness of the trailing edge. Taking into account the experimental data, the values of the coefficient C_{edg} are taken in the range (-0.1)…(-0.25).

Total losses in the diffuser $\zeta_{sum} = \zeta_c + \zeta_{edg}$.

Research of channel diffusers

The ANSYS CFX v.12.1 software package was used for numerical flow modeling in the investigated elements of the flow path of the channel diffuser. The calculated model of the stage centrifugal compressor contains the impeller and the channel diffuser. The calculation zone covers the diffuser with the uniform distribution of the flow parameters at the inlet [9]. Hexahedral computational grids were constructed in ANSYS Turbogrid. Grids of the impeller and the channel diffuser of the model stage of the CC contain 690 690 and 637 296 elements accordingly. The specified number of grid elements is sufficient for the analysis of fluid flow in diffusers [10].

The characteristics of the traditional CD and CD, whose profile was obtained using the above method, are compared. The diffusers have the same geometric dimensions, but different segment profiles. Fig. 12 shows the segment profiles and diffuser performances.



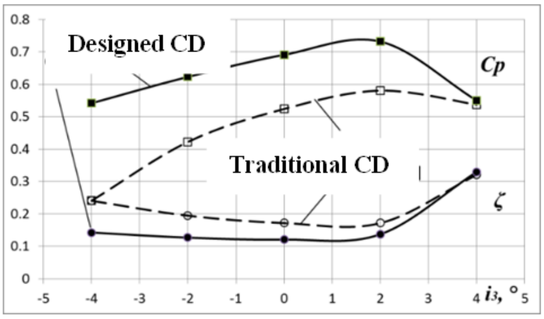


Figure 12 *– Profiles and performances of the traditional geometry CD (dashed line) and one designed using the above method (solid line)*

The designed channel diffuser is more efficient than the traditional one, practically in all modes. In the calculated mode ($i_3 = 0°$) for this diffuser, the value of the static pressure rise coefficient C_p is higher by 32 % and the total pressure loss coefficient ζ is lower by 30 %. At negative incidence ($i_3 < 0°$) the losses in the designed CD are reduced by almost 2 times in comparison with the traditional one; at positive incidence, the values of the coefficients C_p and ζ for these diffusers differ insignificantly.

Numerical studies made it possible to determine the influence of the values of the flow angles at the diffuser inlet and the values of the angle of attack on the performances of the CD. Investigations show that the proposed method is universal, as it allows you to profile effective channel diffusers for a wide range of flow angles from the inlet to the diffuser [9].

Experimental research of channel diffusers

Experimental investigations of the diffuser model, which was designed using a presented method, were made at the aerodynamic setup. Fig. 13 shows the channel diffuser with holes for measuring static pressure on the surfaces of the segments and the diffuser wall along the channel center line, as well as holes for traversing the flow of total pressure tubes along the segment surfaces and the channel center line.

Figure 13 *– Channel diffuser and holes for measuring static and total pressures*

The design pressure distributions along the centerline and the surfaces of the CD segments were confirmed experimentally. Fig. 14 shows the distribution of relative static pressures along the surfaces of the segments.

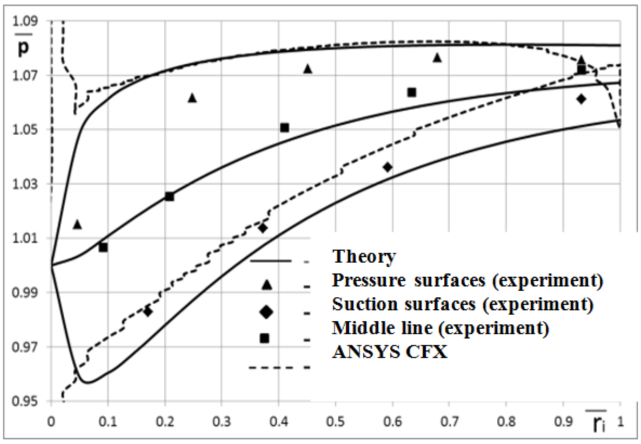
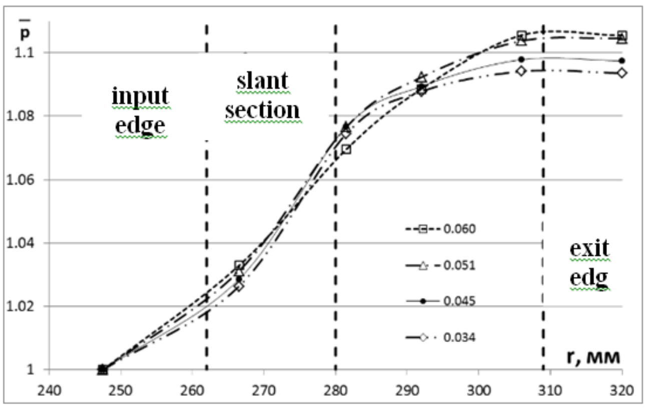


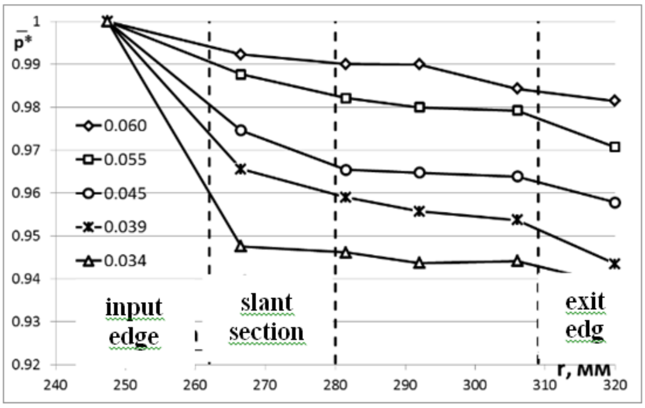
Figure 14 *– Channel diffuser and holes for measuring static and*

Fig. 15 shows the distributions of relative static pressures and total pressures at different values of the flow coefficient Φ_0. The relative static pressure is defined as the ratio of the static pressure to the static pressure at the inlet to the channel $\bar{p} = p/p_{\text{in}}$.

The pressure is increasing most intensively in the «slant section». The values of the pressure at the vaneless region in front of the inlet edges of the segments and in the «slant section» are independent of the mass flow rate. The effect of the mass flow rate on the pressure becomes noticeable from the middle of the diffuser`s channel. As the mass flow rate increases, the pressure increases. This may be because the flow at the impeller outlet in the case of high mass flow rates is more even in the pitch. Therefore, the conversion of the velocity to pressure is more efficient with the low level of energy loss associated with flow leveling. The pressure at the exit edges is hardly at all increasing.



a)



b)

Figure 15 *– The distributions of relative static pressures (a) and relative total pressures (b) on t he middle line along the radius of a CD at different values of the flow rate*

5. Conclusions

The article describes the concept of increasing the energy efficiency of centrifugal compressors through the gasdynamic improvement of vaneless, vaned and channel diffusers, which is based on the developed methodological principles with the solution of the inverse problem of gas dynamics and the design of diffusers with the determination of rational gasdynamic parameters, in particular the the velocity distribution.

Research has confirmed the possibility of improving the characteristics of the stages of centrifugal compressors in the case of the use of stepped vaneless diffusers and diffusers with gas injection. Diffusers with gas injection provide an expansion of the range of stable operation of the stages of centrifugal compressors with the help of radially directed injection into the BLD.

The new method of designing vaned diffusers with the velocity distribution that provides the preseparation state of the turbulent boundary layer on vane surfaces provides an improvement in the gasdynamic characteristics of diffusers compared to traditional geometry diffusers, which is reduced losses from 28% to 50% and increased values of static pressure rise coefficients from 9% to 21% in the wide range of modes.

A fundamentally new method of profiling duct diffusers of centrifugal compressors has been developed, according to which the geometry of the ducts is determined from the condition of ensuring the preseparation state of the boundary layer. The proposed method for profiling centrifugal compressor channel diffusers allows to reduce the losses by 30% and to increase the value of the static pressure rise coefficient by 32% compared to traditional geometry channel diffusers.

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Гідродинамічна конструкція дифузорів відцентрових компресорів

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Дифузори служать для перетворення кінетичної енергії газового потоку в потенціальну, тобто для зменшення швидкості і збільшення тиску. Найпоширенішими типами дифузорів є безлопатковий, лопатковий і канальний. Кожен тип дифузора має свої конструктивні особливості і свої характеристики. Доцільність вибору типу дифузора для відцентрового компресора визначається умовами експлуатації компресора. Поліпшити експлуатаційні характеристики безлопаткових дифузорів можливо за допомогою ступінчастих дифузорів із вдуванням потоку газу. Розроблена математична модель течії газу в безлопатковому дифузорі з вдуванням потоку забезпечує конструювання ступеня відцентрового компресора з більш широким діапазоном стабільної роботи. Традиційні методи проектування каналів турбомашин орієнтовані на використання простих геометричних ліній і поверхонь, таких як пряма, дуга кола, площина, циліндрична поверхня та ін. Проектування виконується згідно з рекомендаціями на основі експериментальних даних. Інший спосіб проектування лопаткових та канальних дифузорів турбомашин пов'язаний з розв'язанням оберненої задачі гідродинаміки, коли форма поверхонь визначається заданим розподілом швидкостей уздовж поверхонь каналу. У статті описані принципи проектування дифузорів відцентрового компресора на основі фізичних і математичних моделей потоку закрученої в'язкої стисливої рідини. Відповідно до представленої методики проектування дифузорів базується на умові передвідривного стану примежового шару на одній із поверхонь. Така конструкція забезпечує зменшення відривних зон в каналах дифузорів і, відповідно, зменшення втрат повного тиску і розширення діапазону стабільної роботи. Новий метод *проектування лопаткових і канальних дифузорів забезпечує поліпшення газодинамічних характеристик дифузорів порівняно з дифузорами традиційної геометрії.*

Ключові слова: *Відцентровий компресор; Метод проектування; Безлопатковий дифузор; Лопатковий дифузор; Канальний дифузор; Дослідження*

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