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Design of the Centrifugal Compressor Vaned and Channel Diffusers Based on the Given Velocity Distribution

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Abstract: Traditional methods of designing channels of turbomachines are focused on the use of simple geometric lines and surfaces, such as a straight line, arc, plane, cylindrical surface, etc. The design is performed according to recommendations based on experimental data. The other way of designing the channels 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. Of course, channels designed in this way can be more efficient than channels designed by the traditional method. This paper describes the design principles for 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 is based on the preseparation condition of the boundary layer along one of the outer surfaces. Numerical and experimental research was performed for vaned and channel diffusers. 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. A new method of designing the vaned diffusers and channel diffusers with the predetermined velocity distribution on vane surfaces provides an improvement in the gasdynamic characteristics of diffusers compared to traditional geometry diffusers.

Keywords: Centrifugal Compressor, Method of the Design, Vaned Diffuser, Channel Diffuser, Research

1. Introduction

Centrifugal compressors are widely used in many industries [5]. One way to reduce the energy consumed by the compressor during its operation is to increase the efficiency of the compressors. Improving the efficiency of compressors can be achieved by improving the gasdynamic characteristics of the elements of the compressor stages, including diffusers.

Much of the energy (40–60%) transmitted by the gas in the impeller of the centrifugal compressor is kinetic. Diffusers are used to convert the kinetic energy of the gas stream into the potential one, that is, to reduce the velocity and increase the pressure. The most common types of diffusers are vaneless, vaned and channel.

The traditional design of the centrifugal compressor diffusers can be characterized as geometric, in which simple surfaces and lines are used – a plane, a straight line, an arc of a circle, etc. So, for vaned diffusers, the center line of the vanes is defined by an arc of a circle. With such the design, the separation of the flow on the pressure surfaces of the vane is characteristic, while the separation point is located close to the inlet edge. For channel diffusers with straight walls the recommendations are used, which are obtained by studying the direct-axis gas flow in diffuser channels. But the gas flow after the impeller is swirling, and therefore the use of such recommendations for swirling gas flow is fundamentally wrong. It is obvious that the creation of new methods for designing vaned and channel diffusers based on the 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, provides diffusers with better gasdynamic characteristics, which increases the performance of compressors [1-4, 6-9, 11-17].

For developing methods for designing vaned and channel diffusers the ideas of Stratford's [15, 16] were used, according to which the separation criterion could be used to choose the form of the surface of the flowing body in such the way that the surface shear stress would be close to zero. Under this condition the flow still at the edge of separation, still remains attached to the surface of the body. This allows for not only minimal total pressure loss but also very low heat flux, since convective heat exchange is associated with surface friction. At the point of flow separation, the maximum pressure gradient is reached.

2. The Mathematical Model

Methods of diffuser vanes designing based on the use of the given velocity distribution, which ensures the preseparation state of the boundary layer on the vane surfaces, are promising.

The swirling flow of compressible viscous gas is considered. The mathematical model for solving the inverse problem of the fluid dynamics has been developed for the steady, adiabatic, unseparated flow.

The angular momentum changes from the diffuser inlet to the current values about the axis for the annular element of fluid:

$$\Delta M = \Delta\dot{m}\left(r_{in}C_{in}\cos\alpha_{in} - rC\cos\alpha\right), \tag{1}$$

where $\Delta\dot{m}$ is mass flow rate for the stream flow; r_{in}, C_{in}, α_{in} are radius, velocity and flow angel at the diffuser inlet; r, C, α are the current values of the radius, velocity and flow angel.

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

$$\Delta M = \Delta b z_{\text{v}} \int_{r_{in}}^{r} \Delta p r dr, \tag{2}$$

where ΔM is the change of the moment of momentum concerning an axis z for the stream with the mass flow rate $\Delta\dot{m}$; Δb is the width of the stream flow; z_{v} is the quantity of vanes; $\Delta p = \left(p_{ps} - p_{ss}\right)$ is the pressure difference on the pressure and suction surfaces of the vane.

After the transformations of (2) using gasdynamic functions for isentropic flow we get

$$r_{in}\lambda_{in}\cos\alpha_{in} - r\lambda\cos\alpha = \frac{bz_{\text{v}}p_{in}^{*}\sigma}{\dot{m}a_{cr}}\int_{r_{in}}^{r}\left[\pi\left(\lambda_{ps}\right) - \pi\left(\lambda_{ss}\right)\right]rdr, \tag{3}$$

where a_{cr} is critical speed of sound, $\lambda = C/a_{cr}$ is the reduced velocity (coefficient of velocity), p_{in}^{*} is the pressure of adiabatically stagnated flow at the diffuser inlet, $\sigma = 1 - (1-\sigma_{out})\left(r - r_{in}\right)/\left(r_{out} - r_{in}\right)$ is the total pressure loss factor, $\sigma_{out} = p_{out}^{*} / p_{in}^{*}$, $\pi(\lambda) = \left(1 - \frac{k-1}{k+1}\lambda^2\right)^{\frac{k}{k-1}}$ is gasdynamic pressure function, k is the adiabatic index; λ_{ps} is the reduced velocities on the pressure surface of the vane; λ_{ss} is the reduced velocities on the suction surface of the vane.

The flow equation for the annular diffuser element width is as follows

$$\dot{m} = 2\pi r C_r \rho b \tau, \tag{4}$$

where C_r is velocity projection; ρ is density; τ is the blockage factor.

The blockage factor is determined by the formula

$$\tau = 1 - \frac{\delta' z_{\text{v}}}{2\pi r \sin\alpha}, \tag{5}$$

where $\delta' = \delta_{\text{v}} + \Sigma\delta^{*}$ is the given blade thickness; $\Sigma\delta^{*}$ is the total displacement thickness of the boundary layers in the interblade channel.

The flow equation using gasdynamic functions is:

$$\dot{m} = \lambda\varepsilon(\lambda) a_{cr}\rho^{*} 2\pi r b \tau \sin\alpha, \tag{6}$$

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

Equations (3) and (6) form the system of equations with unknowns $\alpha, \lambda, \lambda_{ps}, \lambda_{ss}$. 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 a linear law, the pressure changes along the grid step are $\pi\left(\lambda_{ps}\right) + \pi\left(\lambda_{ss}\right) = 2\pi(\lambda)$. Then

$$\pi\left(\lambda_{ss}\right) = 2\pi(\lambda) - \pi\left(\lambda_{ps}\right). \tag{7}$$

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

An important part of the vane diffuser design method is to determine the optimum velocity distribution. The velocity distribution is determined by the condition that the turbulent boundary layer is predetermined.

2.1. Determination of Velocity Distribution

The estimation of the possibility of appearance of a viscous flow separation can be carried out by integrating the angular momentum equation for the boundary layer (Karman equation), which has the form:

$$\frac{d\theta}{dl} + \frac{1}{C}\frac{dC}{dl}\theta(2+H) + \frac{1}{\rho}\frac{d\rho}{dl}\theta = \frac{\tau_w}{\rho C^2}, \tag{8}$$

where θ is momentum thickness; $H = \delta^{*}/\theta$ is the formparameter; δ^* is displacement thickness; C, ρ are velocity and density at the edge of the boundary layer at $y=\delta$; τ_w is shear stress. To determine the separation, the criteria for separation are as follows:

$$f = \frac{dC}{dl}\frac{\theta}{C} G\left(\mathrm{Re}^{**}\right). \tag{9}$$

The condition of separation is $f \le f_{cr}$. In this paper, the Loitcyanskii method [10] was adopted according to which $G\left(\mathrm{Re}^{**}\right) = \left(\mathrm{Re}^{**}\right)^{1/6}$ and $f_{cr} = -0,02$, $\mathrm{Re}^{**} = C \cdot \theta / \nu$.

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

$$\bar{\theta} = 0,0159\,\mathrm{Re}^{-0,15}\,\lambda^{-3,55}\left(\int_0^{\bar{l}} \lambda^4 d\bar{l}\right)^{0,85}, \tag{10}$$

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

The formula for determining the distribution of velocities that ensures the preseparation state of the turbulent boundary layer is obtained by integrating the momentum equation for the boundary layer and has the form [9]:

$$\lambda = \lambda_1\left[1 + \frac{\left(\bar{l} - \bar{l}_1\right)\left(2 + H_s\right)\left(-f_{cr}\right)}{\bar{\theta}_1}\right]^{-\frac{1}{2+H_s}}, \tag{11}$$

where λ_1 and $\bar{\theta}_1$ are the reduced velocity (coefficient of velocity) and the relative momentum thickness at the distance from the inlet edge of the vane, $\bar{l}_1 = l_1/L = 0,02...0,05$, H_s is the formparameter.

2.2. Designing the Vaned Diffuser

The solution system of equations (3) and (6) is performed by the method of successive approximations. The integral in (3) is solved by any numerical method, for example, the method of central rectangles.

Equation (11) is used to determine the velocities at the pressure surfaces of the vane. Then, using (7) with (3) the unknown λ_{ss} is eliminated.

The result of the solution system of equations (3) and (6) for the number of sections from the inlet to the outlet of the diffuser is the distribution of the rate-average flow angles and velocities along the vane.

Based on this data, the geometric parameters of the vane and the velocity distributions on its surfaces are determined. Boundary layers are taken into account in the calculations. The area of the boundary layer is replaced by the displacement thickness, which is determined by the ratio $\delta^* = H_s \delta^{**}$. Equation (10) is used to determine the values of the relative momentum thickness and then, using (9) determines the value of the formparameters on the suction surfaces of the vane. Often, on the suction surfaces of the vanes have the flow separation, but the separation occurs much further from the inlet edges of the vanes than in the diffusers with vanes with the traditional shape of the profile. Obviously, head losses will be smaller.

2.3. Designing the Channel Diffuser

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 (11).

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 (3) and (6) 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 (10), and the value of the formparameter is calculated by the (9). After calculating the boundary layer displacement thicknesses in the channel, the thickness of the diffuser segments is determined by (5). 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.

3. Numerical Calculation

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 $>10^5$ and Mach numbers 0,2…0,6. The calculations were performed for 8 modes of operation of stages, which differed by mass flow.

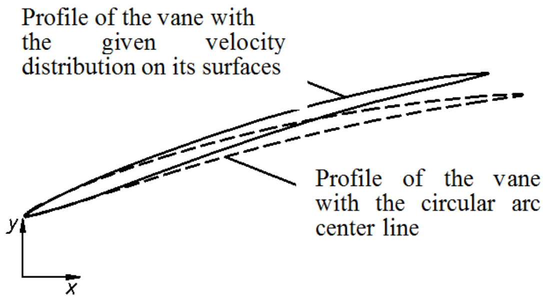

Figure 1. *Profiles of vanes.*

Figure 1 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.

The initial data for the design of the vaned diffusers are the flow parameters at the inlet to the diffuser, determined on the basis of experimental data of the model stage: $\alpha_3 = 21,6°$; $p_3 = 121\ kPa$; $p_3^* = 139\ kPa$; $T_3 = 309,7\,K$;

$T_3^* = 322,2\ K$; $C_3 = 156\ m/s$; $\rho_3 = 1,36\ \text{kg}/\text{m}^3$.

The design value of the mass flow rate is $2\ kg/s$. The geometrical parameters of the diffusers are as follows: $l/t = 1,5$; $b_3 = b_4 = 1,08 \cdot b_2$; $r_3 = 1,14 \cdot r_2$; $r_4 = 1,45 \cdot r_2$.

Figure 2 shows the values of total pressure loss (ζ) and static pressure rise (C_p) coefficients for diffusers designed by different methods.

Vaned diffuser, designed with the preseparation velocities distribution, has better performance in nominal mode ($\Phi_0 = 0,0502$) 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.

In the ANSYS CFX software, the channel diffuser (CD) was calculated using the proposed method. 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 ($r_3 = 253\ mm$, $r_4 = 358\ mm$, $b_3 = b_4 = 20\ mm$, the thickness of the segments at the inlet $\delta_{sin} = 7\ mm$, $z_s = 14$, $\alpha_{in} = 14^0$, $\alpha_{out} = 44^0$), but different segment profiles. Figure 3 shows segment profiles and diffuser characteristics obtained in ANSYS CFX.

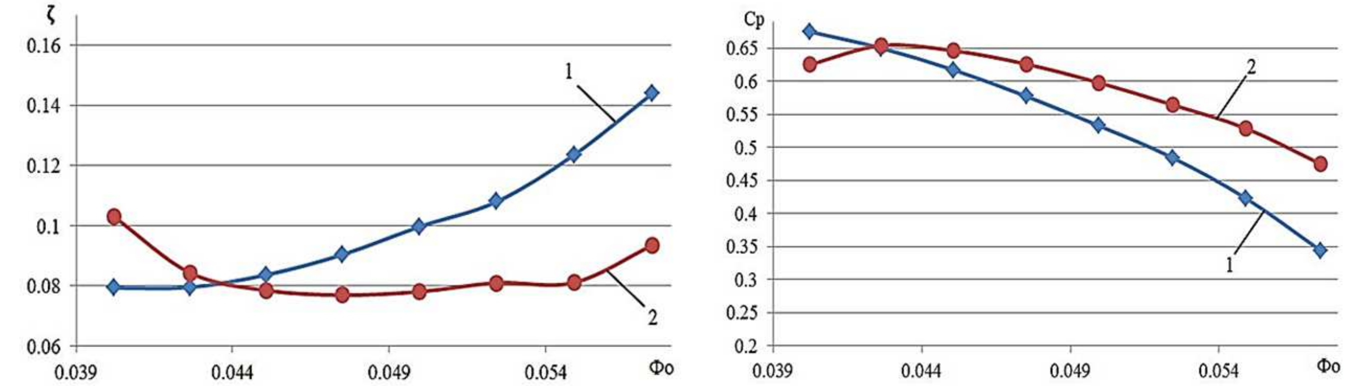

Figure 2. Dependences of loss coefficients (ζ) and static pressure rise coefficients (C_p) on flow coefficient Φ_0); 1–the center line of the vane is the arc of a circle; 2–given velocity distribution on the surface of the vane.

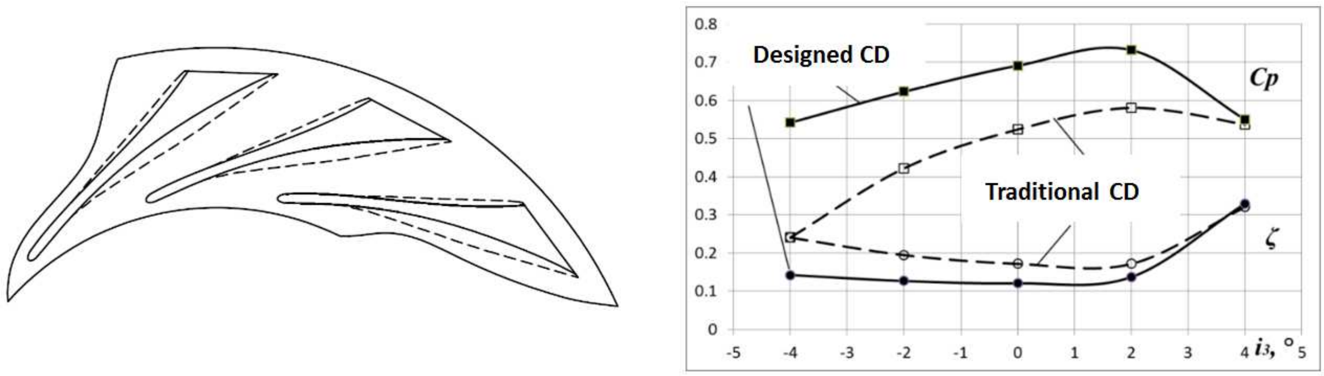

Figure 3. Profiles and characteristics of traditional geometry CD (dashed line) and designed using the above method (solid line).

Designed channel diffuser is more efficient than traditional, practically in all modes. In the calculated mode ($i_3 = \alpha_{3s} - \alpha_3 = 0^0$) for this diffuser, the value of the static pressure rise coefficient $C_p = \left(p_{out} - p_{in}\right)/\left(p_{in}^* - p_{in}\right)$ is higher by 32% and the total pressure loss coefficient $\zeta = \left(p_{in}^* - p_{out}^*\right)/\left(p_{in}^* - p_{in}\right)$ is lower by 30%. At negative angles of attack ($i_3 < 0^0$) the losses in the designed CD are reduced by almost 2 times in comparison with the traditional one; at positive angles of attack, the difference of the values of the coefficients C_p and ζ is less significant for these diffusers.

4. Experimental Research

Experimental investigations of the diffuser model, which was designed using presented method, made at the aerodynamic stand. Figure 4 shows scheme of experimental model. The model includes: confuser 1, axial channel 2, impeller 3, vaned diffuser 4, crossover 5, return cannel 6, axisymmetric channel 7, an outlet device 8.

The design pressure distributions along the center line and along the surfaces of the channel diffuser segments were confirmed experimentally. Figure 5 shows the distribution of relative static pressures along the surfaces of segments.

The discrepancy between theoretical and experimental values of the pressures at the corresponding points does not exceed 2%.

Therefore, the method presented in this paper allows channel diffuser segments to be profiled with sufficient accuracy for engineering calculations. The calculated pressure distributions can be used to qualitatively and quantitatively evaluate the flow structure in the diffuser, which is important at the design stage.

Figure 6 shows the distributions of relative static pressures and total pressures at different values of the flow rate 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 $\overline{p} = p/p_{in}$.

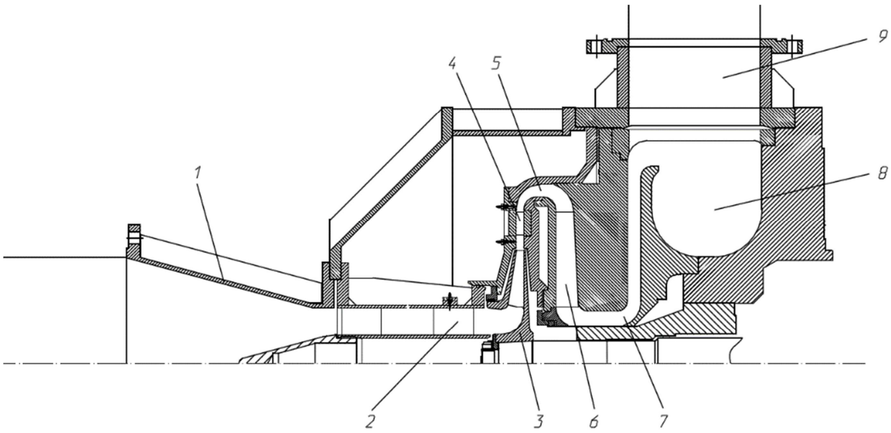

Figure 4. Scheme of the experimental model.

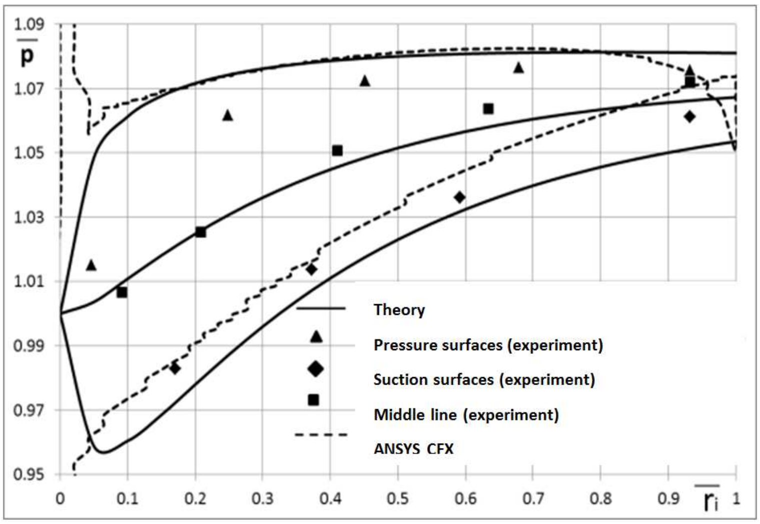

Figure 5. The distribution of relative static pressures along the surfaces of segments.

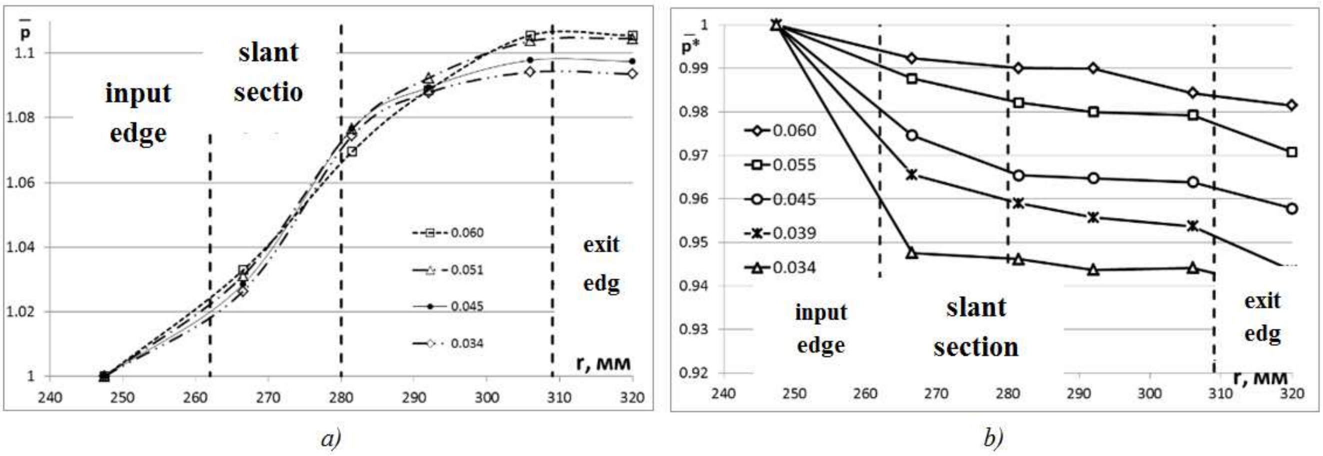

Figure 6. The distributions of relative static pressures (a) and relative total pressures (b) on the center line along the radius CD at different values of the flow rate.

The pressure is increasing most intensively in the slant section. The values of the pressure at the vaneless section in front of the inlet edges of the segments and in the slant section are independent from the flow. The effect of the flow rate on the pressure becomes noticeable from the center of the diffuser`s channel. As the flow rate increases, the pressure increases. This may be due to the fact that the flow at the outlet of the impeller in the case of high flow rates is more even in the grid step. Therefore, the conversion of velocity to pressure is more efficient with a low level of energy loss associated with flow levelling. The pressure at the exit edges is hardly at all increasing.

A significant drop in the total pressure (up to 5%) occurs on the vaneless section before the diffuser.

The lower the mass flow rate is, the greater is the pressure drop. This is mainly due to the mixing of "jets" and "traces" formed on the impeller.

The total pressure drop in the diffuser channel is close to 1% for almost all compressor modes, which indicates the high efficiency of the segment profiles. Losses in the channel diffuser are also increased due to the mixing of "traces" along the exit edges of the segments.

Figure 7 shows experimental and theoretical loss coefficients obtained for different sections of the channel diffusers. The discrepancy between the experimental and theoretical values of the loss coefficients ζ_c, ζ_Σ does not exceed 15% for all modes of operation.

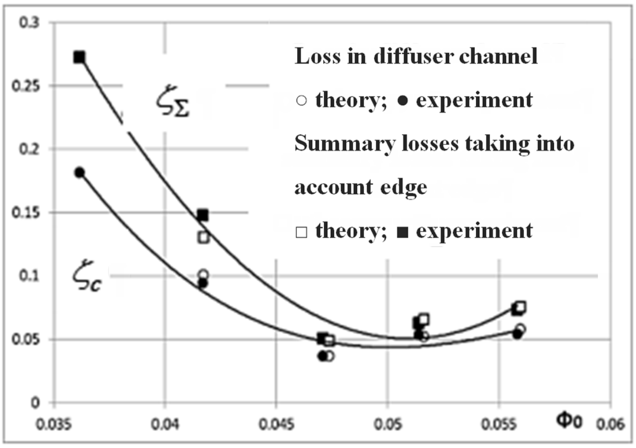

Figure 7. *Dependences of loss coefficients CD on flow coefficient.*

The lowest channel diffuser loss is close to the nominal mode with $\Phi_{0n} \approx 0{,}047$. With the reduction of the mass flow of gas $\Phi_0 < \Phi_{0n}$ the losses increase more intensively than in the case of the increase in flow rate.

5. Conclusion

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. 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.

The designing of double-row diffusers using the proposed method makes it possible to obtain the completely unseparated gas flow in the diffuser [9].

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