

# Analysis of the gas-dynamic performance of a vaned diffuser with given velocity distribution along the vane's surfaces

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**Abstract.** Extension of the effective range of vaned diffusers is one of the promising ways to improve the centrifugal compressor's stages which are used in numerous fields of industry. The new method of profiling of the diffuser vanes has been developed using Stratford's results and boundary layer theory by Loytsanskiy. The developed method is based on the solution of the inverse task of gas-dynamic using given velocity distribution along the vane's surface. Comparison of the results of numerical simulations for different diffusers has shown that the performance of the diffuser designed with the resulting velocity distribution are better. Influence of the vane profile, number of the vanes, diffuser outlet diameter and the diffuser width on diffuser characteristics has been investigated. The results of the simulations have been used to formulate recommendations on the design of high-effectiveness vaned diffusers for centrifugal stages of different types.

## NOMENCLATURE

$\bar{c}$	absolute velocity, m/s;	$p$	pressure, Pa;
$\alpha$	flow angle, °;	$T$	temperature, K
$R_v$	radius of mean camberline vane, m;	$R$	gas constant, J/(kg·K);
$\delta^{**}$	momentum thickness;	$\pi(\lambda)$ , $\varepsilon(\lambda)$	gas-dynamic function;
$\tau$	tangential stress	$\zeta$	loss factor;
$U_\delta$	velocity of the gas in a potential flow;	$C_p$	static pressure recovery coefficient;
$H, f$	form-parameters;	$F_0$	flow coefficient;
$\nu$	kinematical viscosity, m <sup>2</sup> /s;	$\bar{m}$	mass flow rate, kg/s;
$\lambda = c/a$	coefficient of velocity;	$V$	volume flow rate;
$a$	critical sonic speed, m/s;	$U_2$	impeller tip speed;
$\bar{l} = l/L$	relative length of the vane;	$M$	Mach number;
$r$	radius, m;	<b>Subscripts</b>	
$b$	height of the vane, m;	in	diffuser inlet;
$z_v$	number of vanes in diffuser;	out	diffuser outlet;
$D_2$	tip diameter of the impeller;	ps	pressure surface of the vane;
$\delta$	vane thickness, m;	ss	suction surface of the vane;
$\rho$	density of the gas stream, kg/m <sup>3</sup> ;	<b>Superscript</b>	
		*	parameters of adiabatically stagnated flow.



## 1. Introduction

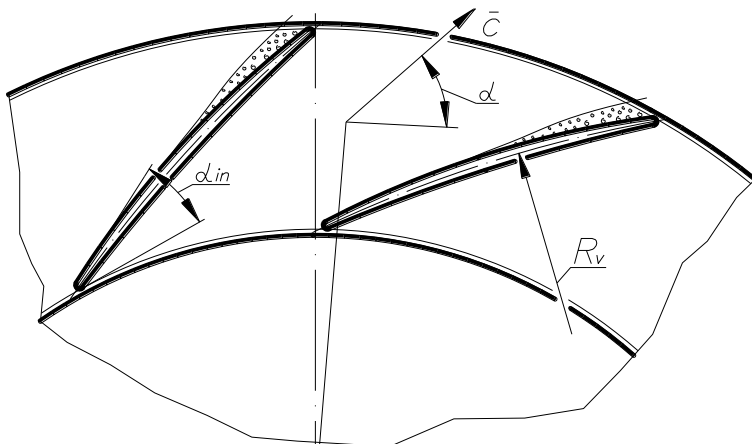
Centrifugal compressors are widely used in numerous fields of industry including transportation of natural gas, petroleum and chemical industries. Power consumed by centrifugal compressors may reach  $32MW$ . Effectiveness of centrifugal compressors depends on aerodynamic flow quality of the elements of the stages.

The power of the compressor's drive is consumed to compress and to move the working gas and part of energy is consumed to overcome the aerodynamic resistance to the movement of gas. Losses of mechanical energy in centrifugal compressor are approximately uniformly shared between the impeller, vaned diffuser and return channel [1]. Therefore, the increase of efficiency of elements of the flow path of the centrifugal compressors is an essential task in engineering.

## 2. Vaned diffuser design procedure

Vaned diffusers with circular centrelines and symmetric vane's profile C-4 are widely used in centrifugal compressors [2]. These diffusers have simple geometry and are usually designed ignoring parameters of the compressed gas, real kinematics of the flow at the vane's inlet, vane-spacing ratio and geometrical parameters of the flow path between the vanes. As a result vaned diffusers with such geometry are characterized by flow separations from the pressure surface of the vanes at almost all conditions of operation. All these facts lead to narrow effective range of the stages with such diffusers (Figure 1).

Flow separation from the front surface of the vanes at all conditions of operation leads to an increase of the loss coefficient of vaned diffusers and decrease of the static pressure recovery coefficient.



**Figure 1** - flow separation from the pressure surface of the circular-arc mean camber line vane [3].

Based on the analysis of drawbacks of well-known design methods of vaned diffuser offered a method of profiling vanes of the diffuser with a given velocity distribution, based on Stratford's idea of profiling a flat diffuser with unseparated gas flow [4]. Unseparated gas flow provided by profiling the shape of the diffuser from the condition of zero tangential stresses on the walls of the diffuser. The boundary layer structure of gas flow on surfaces of the diffuser vanes was taken

into account in the model suggested by Loitsianskiy [5]. Based on the above stated, the method of solving the inverse task of gas-dynamics for profiling high-performance vane diffuser was suggested.

Law of speed change of gas flow on the front surface of the vane is determined from the Karman's integral impulses ratio:

$$\frac{\delta^{**}}{dx} = \frac{1}{U_\delta} \frac{U_\delta}{dx} (2 + H) \delta^{**} + \frac{\tau}{\rho U_\delta^2}, \quad (1)$$

According to the recommendations [5], the condition of unseparated flow to turbulent flat flow is as following:

$$-\frac{\delta^{**}}{U_\delta} \frac{dU_\delta}{dx} G(Re^{**}) \leq f_s, \quad (2)$$

where  $f_s$  is the critical value of the form-parameter,  $f_s = -0.02$ ;  $G(Re^{**}) = (Re^{**})^{\frac{1}{6}}$ .

Reynolds number for the momentum thickness is determined from the expression:

$$Re^{**} = \frac{\lambda \cdot a \cdot \delta^{**}}{\nu}, \quad (3)$$

By integrating the equation (4) we obtain:

$$\frac{d}{dx} \left( f \frac{U_\delta}{U_\delta} \right) = (1+m)\zeta - [2+m+(1+m)H]f, \quad (4)$$

where  $f = -\frac{\delta^{**} U_\delta}{U_\delta} G(Re^{**})$ ;  $m = \frac{Re^{**} G(Re^{**})}{G(Re^{**})}$ ;  $\zeta = \frac{\tau}{\rho U_\delta^2}$ .

The boundary layer will have the preseparation nature in the case when the form parameter  $f$  is equal to its critical value  $f_s$ , while  $\zeta = 0$ ,  $m = 0$  and  $H = H_s$ , where  $H_s$  is the critical value of the form parameter.

According to [5], the amount of turbulent and viscous friction depends solely on the form parameter, then for  $f = -0.02$ , equation (4) can be integrated in quadrature and written in dimensionless form [6]:

$$\lambda_{psi} = \lambda_{in} \cdot \left[ 1 + \frac{(\bar{l}_i - \bar{l}_1) \cdot (2 + H_s) \cdot (-f_s)}{\delta_{in}^{**}} \right]^{-\frac{1}{2+H_s}}, \quad (5)$$

In order to determine the shape of the vane profile with given by velocity distribution on its surface the equations system has been considered. It consisted of the equation of the moment of impulse, mass flow rate and state.

Momentum from the impulse equation for current streams is as following:

$$r_{in} \cdot \lambda_{in} \cdot \cos(\alpha_{in}) - r_i \cdot \lambda_i \cdot \cos(\alpha_i) = \frac{b \cdot z_v}{\bar{m} \cdot a} \cdot \int_{r_{in}}^{r_i} \Delta P \cdot r \cdot dr, \quad (6)$$

Mass flow rate equation using gas-dynamic functions is as following:

$$\bar{m} = \lambda \cdot \varepsilon(\lambda) \cdot a \cdot \rho_{in}^* \cdot \sigma \cdot b \cdot (2 \cdot \pi \cdot r - \delta \cdot z_v), \quad (7)$$

where  $\rho^*$  is density of the adiabatically stagnated flow;  $\sigma$  is function of total pressure loss;  $\delta$  is thickness of the vane ( $\delta = \delta_v + \delta^*$ );  $\delta^*$  is displacement thickness.

As the equation of state the ideal gas law was used:

$$\frac{P}{\rho} = R \cdot T, \quad (8)$$

where  $R$  is gas constant of the working fluid;  $T$  is the temperature of the gas flow.

Nature of change in pressure between the surfaces of the vanes were accepted linear [7]:

$$\Delta P = P_{ps} - P_{ss} = P_{in}^* \cdot [\pi(\lambda_{ps}) - \pi(\lambda_{ss})], \quad (9)$$

The system of equations (6), (7) and (8) is produced by successive approximations. Definite integral in equation (6) is solved numerically by the central rectangles.

According to [5] region of the boundary layer is replaced by thickness repression, in which stands out thickness of the momentum of impulse losses.

Displacement thickness is determined from the expression:

$$\delta^* = H_s \cdot \delta^{**}. \quad (10)$$

The relative thickness of the loss of momentum of impulse is determined by the expression:

$$\bar{\delta}^{**} = 0.0159 \cdot Re^{-1,15} \cdot \lambda^{-3,55} \cdot \left( \int_0^{\bar{l}} \lambda^4 \cdot \bar{l} \right)^{0,85}, \quad (11)$$

Solving the inverse task of gas-dynamics is the distribution of the mean flow rate angle of flow in the space between vanes of diffuser that forms by the middle line of the vane.

### 3. Criteria of efficiency of the vaned diffuser

The main purpose of vaned diffuser as part of a centrifugal compressor stage is to convert the kinetic energy of the flow emerging from the impeller into static pressure with minimal losses in all conditions of operation. Criteria of efficiency for vaned diffuser are: loss coefficient and coefficient of recovery of static pressure depending on the nominal flowrate [8].

Loss coefficient is defined from the relation:

$$\zeta = \frac{P_3^* - P_4^*}{P_3^* - P_3}, \quad (12)$$

where  $P_3^*$  and  $P_4^*$  is the adiabatically stagnated pressure before and after the vane system of diffuser;  $P_3$  is static pressure of gas flow before the vaned diffuser.

The coefficient of static pressure recovery is defined from the relation:

$$c_p = \frac{P_4 - P_3}{P_3^* - P_3}, \quad (13)$$

where  $P_4$  is static pressure of gas flow after vanes of the diffuser.

Flow coefficient, which characterizes the operation conditions of centrifugal compressor stage, is defined from the relation:

$$F_o = \frac{4 \cdot V_o}{\pi \cdot D_2^2 \cdot U_2} \quad (14)$$

#### 4. Numerical modelling of gas flow in the vaned diffuser

To solve the direct task of gas-dynamics the software package ANSYS CFX was used.

Verification of numerical simulation technique was conducted by comparing the integrated gas-dynamic characteristics of centrifugal compressor stages derived from numerical and experimental investigation. The experiment was carried out at a test rig AD-400 of the PJSC "Frunze NPO" (Sumy, Ukraine). The value of impeller tip speed Mach number for experimental investigation was of 0.6. On the nominal operation mode (mass flow rate of 2 kg / s) stage provides a pressure ratio of 1.3. The results of comparison of gas dynamic characteristics of centrifugal compressor stages, obtained in numerical and experimental way, presented in [9].

Settings for modelling of process of gas flow in the flow path of the centrifugal compressor stages were cyclically symmetric and stationary.

At the inlet to the computational domain as the boundary conditions have been set parameters of adiabatically stagnated flow before of the impeller; at the outlet - proportion of the mass flow passing through the segment of the flow path.

The flow path of the centrifugal compressor stage has been replaced by a block-structured computational grid [10]. Computational grid was meshed near the walls for the application of SST turbulence model, which is able to predict the locations of occurrence of flow separation the most accurately. The mean value of  $y^+$  on all test conditions of operation was not exceeding 2.

The choice of integration step was defined from the relation:

$$q = \frac{30}{\pi \cdot n}, \quad (15)$$

where  $n$  is impeller rotating speed, rev/min.

Fluctuations in the values of polytropic efficiency and coefficient of polytropic head were adopted as criteria for convergence of the numerical investigation. Their value for the centrifugal compressor stage is within the error of the model experiment. Processing of results of numerical simulation was performed in ANSYS CFX Post processor. It included determination of the parameters that are required to design the vaned diffuser and calculation of its criterions of efficiency.

Numerical simulation of gas flow passed in two stages. The purpose of the first stage of simulation was to determine the initial data for the design of the vaned diffuser. For this purpose two-element (impeller ( $D_2=460 \text{ mm}$ ,  $\overline{b}_2 = 0.0319$ ,  $\beta_{v_1} = 30^\circ$ ,  $\beta_{v_2} = 104^\circ$ ) and vaneless diffuser ( $D_4=759 \text{ mm}$ ,  $b_3 = b_4 = 15.5 \text{ mm}$ ) centrifugal compressor stage has been calculated. Flow simulation in vaneless diffuser was due to the necessity of obtaining parameters of the gas flow behind the impeller without inverse influence of the diffuser vane system [2]. In the inlet of the vaneless diffuser were defined the following flow parameters at all study conditions of the centrifugal compressor stage: the absolute velocity of the air flow, the value of the static pressure and temperature, adiabatic stagnated pressure and temperature and density. These parameters of the vaneless diffuser are required for design of the vaned diffuser. At the second stage two-element (impeller and vaned diffuser) centrifugal compressor stage has been calculated. Geometric parameters are identical to the impeller parameters used in the first stage of the simulation. The second stage involved the determination of gas-dynamic characteristics of the investigated vaned diffusers.

#### 5. Results of numerical modelling

For all stage models simulation of the air flow was carried out on at least 6 operating conditions, which differed in the value of mass flow rate. The values of Reynolds number and Mach number at the inlet of the vane system of the diffuser are in the ranges of  $Re=(2.3-3.2) \cdot 10^5$  and  $M=(0.28-0.51)$ , respectively. Impeller tip speed Mach number for all numerical studies was  $Mu_2=0.6$ . The vane-spacing ratio, in the study of influence of the designing method the vaned diffuser, corresponded to the value  $l/t = 1.5$ . Symmetrical airfoil C-4 was used in the study of influence of a designing method of the vane diffuser, the vane-spacing ratio and the diffuser width.

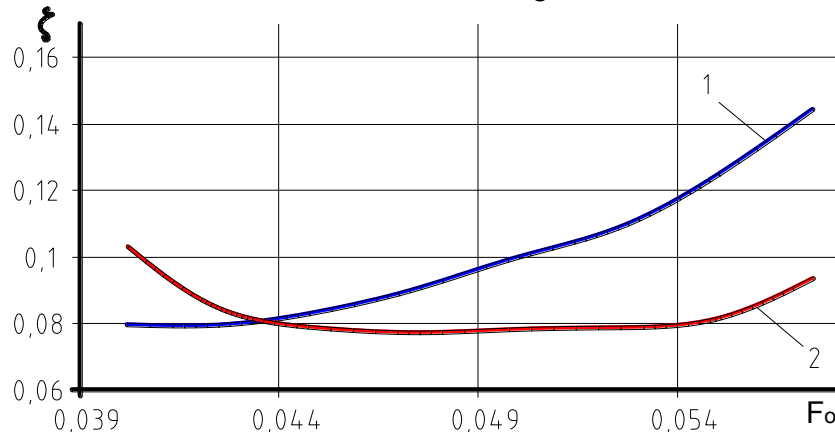
The initial data for designing of the vaned diffuser with the given velocity distribution is shown in Table 1.

**Table 1.** Initial data for designing of the vaned diffuser

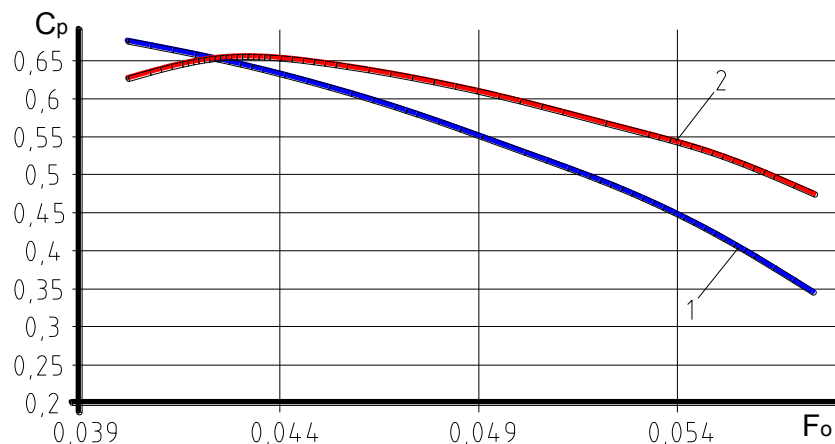
$\alpha_3, ^\circ$	$P_3, Pa$	$P_3^*, Pa$	$T_3, K$	$T_3^*, K$	$c_3, \frac{m}{s}$	$\rho_3, \frac{kg}{m^3}$
21.6	120946	138973	309.72	322.2	156	1.360

### 5.1. The comparison of the efficiency of the diffuser with the given velocity distribution and the circular-arc mean camber line vaned diffusers

Variations in the loss coefficient and coefficient of recovery of static pressure for diffusers designed with the use of different methods are shown in Figure 2 and 3.



**Figure 2.** The loss coefficient of the vaned diffuser: 1 is the circular-arc mean camber line vaned diffusers; 2 is diffuser with given velocity distribution on the vane surface.



**Figure 3.** The coefficient of recovery of static pressure of the vaned diffuser: 1 is the circular-arc mean camber line vaned diffusers; 2 is diffuser with given velocity distribution on the vane surface.

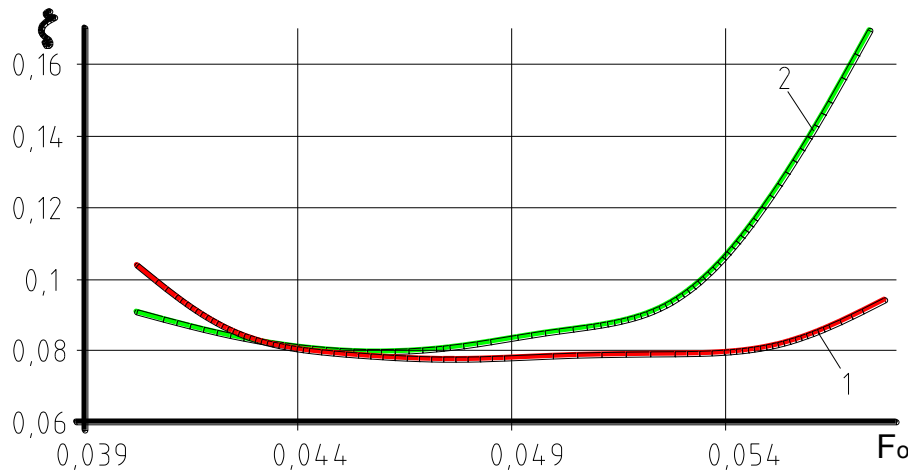
Figure 2 shows that the vaned diffusers designed with taking into account the velocity distribution on the surfaces of vanes are more effective both on a design condition (by 28%) when  $Fo = 0.0502$  and on the maximum capacity operation condition (by 54%) when  $Fo = 0.0574$ . On the near surge point of operation of the centrifugal compressor stage ( $Fo = 0.0402$ ) there is slight deterioration in values of the loss coefficient of the diffuser (by 30%), as a result of flow separation from the back surface of the vane. Separator of the air flow occurs due to impact inleakage on the diffuser vane with a positive angle of attack ( $+5^\circ$ ). The coefficient of static pressure recovery of the vaned diffuser, which was designed by the proposed procedure, reaches higher values on design mode (by 9%) and in the maximum capacity (by 21%) relatively to the circular-arc mean camber line vaned diffusers (Figure 3). Whereas, there is a 8% decrease in value of the static pressure recovery coefficient of the diffuser with given velocity distribution on near surge point of operation.

Thus, the analysis of integrated gas-dynamic characteristics of the vaned diffuser that are designed using different methods has shown a noticeable improvement in the performance of the diffuser with the given velocity distribution in the range of flow coefficient  $Fo = 0,0435 \div 0,0574$ , which is 80% of the full range of stage operation.

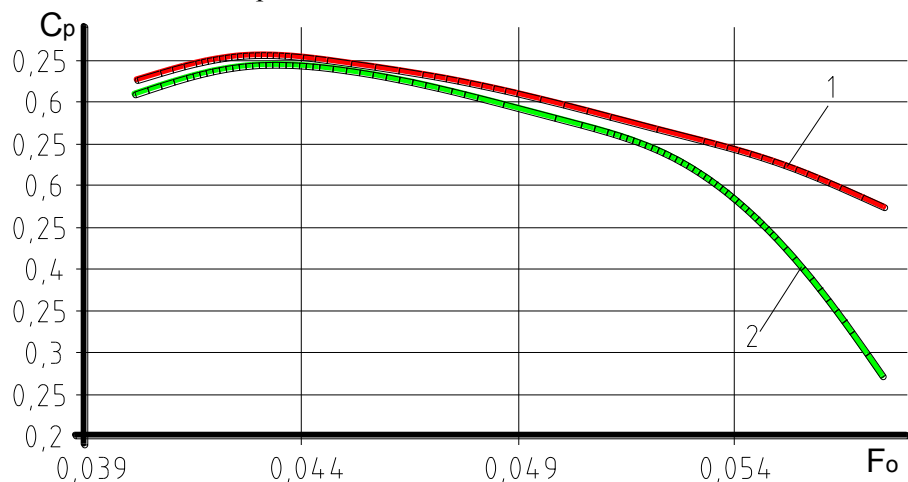
### 5.2. Investigation of influence of the vanes shape on the efficiency of the vaned diffuser

Two types of profiles were compared. The first - a symmetrical airfoil C-4, the second - the profile of constant thickness ( $\delta_v = 6 \text{ mm}$ ). Both diffusers had 19 vanes.

Figures 4 and 5 show a comparison of gas-dynamic characteristics of diffusers with different vanes profiles.



**Figure 4.** The loss coefficient of the diffusers with different vanes profiles: 1 – airfoil C-4; 2 – the profile of constant thickness



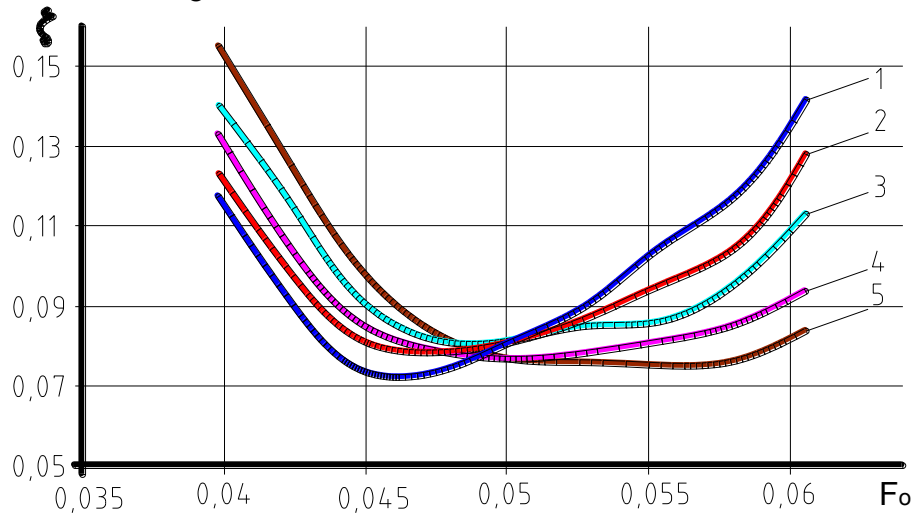
**Figure 5.** The coefficient of recovery of static pressure of the diffusers with different vanes profiles: 1 – airfoil C-4; 2 – the profile of constant thickness

From the analysis of gas-dynamic characteristics of the vaned diffuser there is an evident deterioration in diffuser's efficiency with a constant blade profile thickness in comparison with aerofoil C-4 vanes (Figure 4). Application of the airfoil C-4 in design of the vaned diffuser with a given velocity distribution allows to obtain the increase in values of the loss coefficient and the coefficient of static pressure recovery at the maximum flow coefficient ( $Fo = 0.0574$ ) by 83% and 72%, respectively. A sharp constraint of the flow on the diffuser inlet is probable reason for the deterioration of gas-dynamic characteristics of the diffuser with a constant vane thickness. Although the influence of constraint coefficient can be reduced by making the diffuser's vane thinner and in such a way improving the flow around the inlet edge of the vane. Although the influence of constraint coefficient can be reduced by making the diffuser's vane thinner and in such a way improving the flow around the inlet edge of the vane, but this approach is impossible because of the requirements for strength and stiffness of the structure of the flow path elements of the centrifugal compressor.

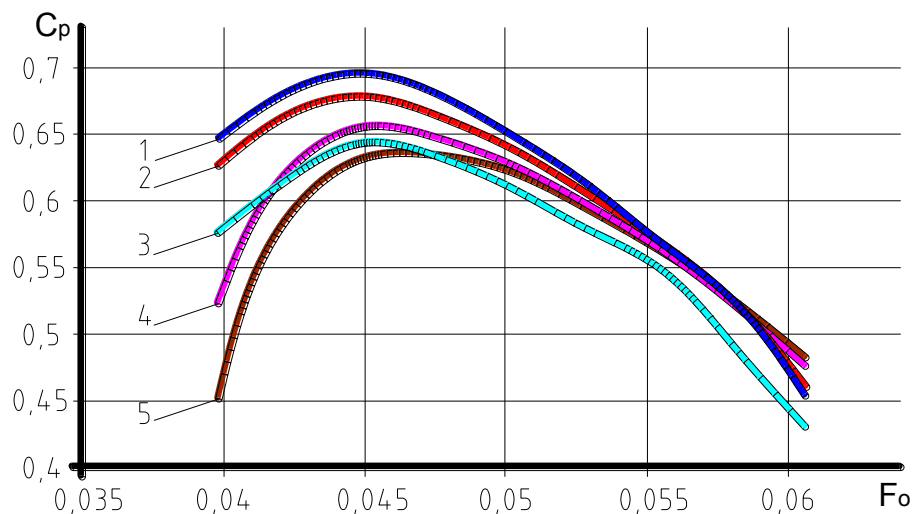
### 5.3. Investigation of influence of the vane-spacing ratio on efficiency of the diffuser

Investigation of influence of the vane-spacing ratio ( $l/t$ ) on efficiency of the diffuser designed with a given velocity distribution on the surface of the vane, was carried out for the following values:  $1.0$  ( $z_{34}=12$ );  $1.25$  ( $z_{34}=16$ );  $1.5$  ( $z_{34}=19$ );  $1.75$  ( $z_{34}=23$ );  $2.0$  ( $z_{34}=27$ ).

Results of the comparison of gas-dynamic characteristics of the studied diffusers with different values of  $l/t$  are shown in Figures 6 and 7.



**Figure 6.** The loss coefficient of the diffusers with different vane-spacing ratio: 1 –  $l/t=1.0$ ; 2 –  $l/t=1.25$ ; 3 –  $l/t=1.5$ ; 4 –  $l/t=1.75$ ; 5 –  $l/t=2$



**Figure 7.** The coefficient of recovery of static pressure of the diffusers with different vane-spacing ratio: 1 –  $l/t=1.0$ ; 2 –  $l/t=1.25$ ; 3 –  $l/t=1.5$ ; 4 –  $l/t=1.75$ ; 5 –  $l/t=2$

The analysis of the gas-dynamic characteristics of the diffuser with different vane-spacing ratio showed that diffusers with values  $l/t$  equal to  $1.75$  and  $2$  have a pronounced peak of the loss coefficient, which is shifted towards lower flow coefficient. Diffusers with  $l/t = 1.5$  have gentle changes in the values the loss coefficient and the coefficient of static pressure recovery.

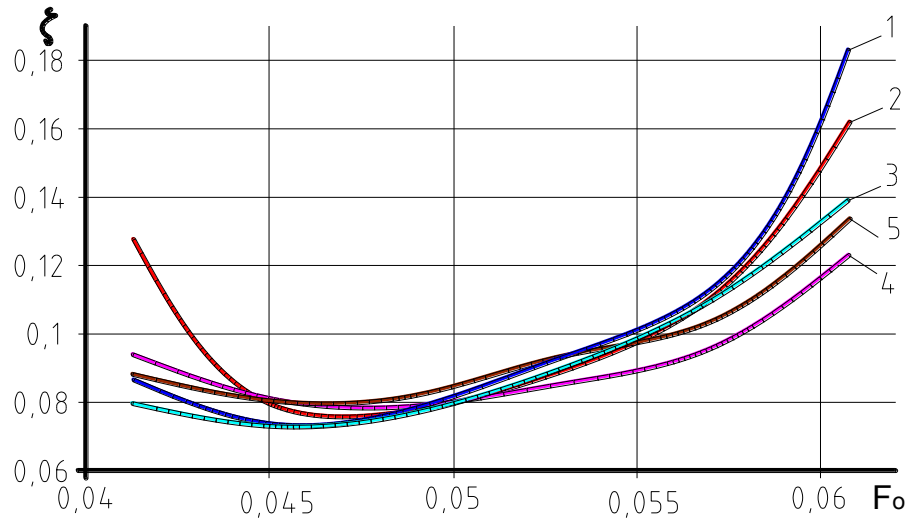
### 5.4. Investigation of influence of the width of the diffuser on its efficiency

The angle of the gas flow depends on the cross-sectional area on the diffuser inlet. While studying the influence of the width of the diffuser on its efficiency an additional piece of research was conducted. It was aimed at identifying the initial data for profiling diffusers with the velocity distribution on the surfaces of the blades. Initial data for the design of the vaned diffuser with different widths is shown in Table 2.

**Table 2.** Initial data for the design of the vaned diffuser

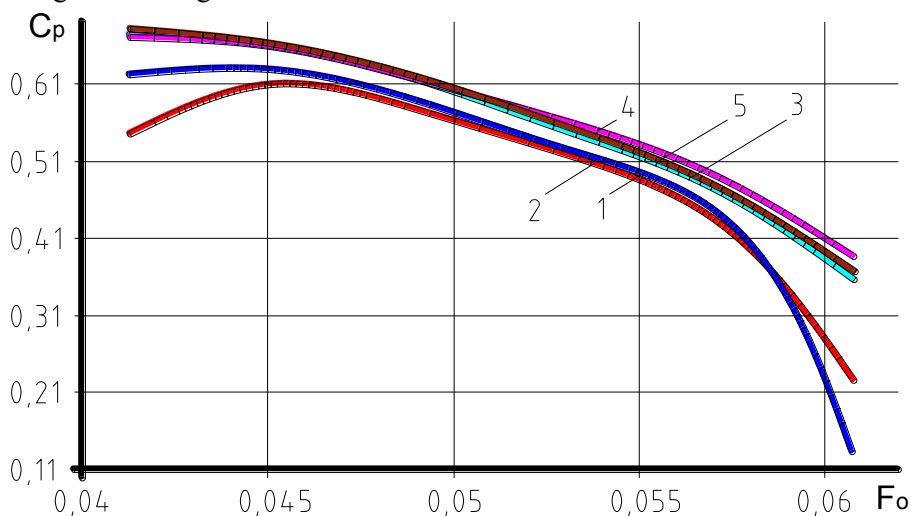
$b_3$	$\alpha_3, ^\circ$	$P_3, Pa$	$P_3^*, Pa$	$T_3, K$	$T_3^*, K$	$c_3, \frac{m}{s}$	$\rho_3, \frac{kg}{m^3}$
0.9	19.9	120417	139514	310.3	323.6	162	1.351
1.0	20.1	121503	140187	310.6	323.5	160	1.363
1.1	20.6	121085	138830	310.9	323.3	157	1.357
1.2	19.1	121311	138533	311.1	323.1	154	1.358
1.3	17.6	121299	138430	311.2	323.1	154	1.358

The Investigation of the influence of the width of the diffuser on its efficiency was carried for the following values:  $b_3 = 0.9 \cdot b_2$  (13.2 mm,  $z_{34}=20$ );  $1.0 \cdot b_2$  (14.7 mm,  $z_{34}=18$ );  $1.1 \cdot b_2$  (16.2 mm,  $z_{34}=18$ );  $1.2 \cdot b_2$  (17.6 mm,  $z_{34}=16$ );  $1.3 \cdot b_2$  (19.1 mm,  $z_{34}=16$ ). Gas-dynamic characteristics of investigated diffusers with a given distribution of velocities and different value of the  $b_3$  are presented in Figures 8 and 9.



**Figure 8.** The loss coefficient of the diffusers with different value of the  $b_3$ : 1 –  $0.9 \cdot b_2$  (13.2 mm); 2 –  $1.0 \cdot b_2$  (14.7 mm); 3 –  $1.1 \cdot b_2$  (16.2 mm); 4 –  $1.2 \cdot b_2$  (17.6 mm); 5 –  $1.3 \cdot b_2$  (19.1 mm).

Analysis of graphics presented in Figure 8 showed that the diffuser with height of the vane equal to  $1.2 b_2$  has the most gentle change in the value the loss coefficient. In the range of flow coefficient  $Fo = 0.049 \div 0.0607$  vaned diffuser with  $1.2 b_2$  has the minimal value of the loss coefficient among the investigated diffusers.



**Figure 9.** The coefficient of recovery of static pressure of the diffusers with different value of the  $b_3$ : 1 –  $0.9 \cdot b_2$  (13.2 mm); 2 –  $1.0 \cdot b_2$  (14.7 mm); 3 –  $1.1 \cdot b_2$  (16.2 mm); 4 –  $1.2 \cdot b_2$  (17.6 mm); 5 –  $1.3 \cdot b_2$  (19.1 mm).



According to Figure 9 vaned diffuser with  $b_3=1.2 b_2$  produces the most effective braking of the air flow in the entire range of values of the flow coefficient.

## 6. Comparison of the results numerical modeling and theoretical analysis

The suggested in 2nd Section method of solving the inverse task of gasdynamics allows to determine the following parameters of the flow in the vaned diffuser: pressure, density and temperature. The match in changes of the static pressure along the vane surface defined in different ways (using the proposed procedure and software package ANSYS CFX) is a confirmation of the correct choice of the mathematical model of the method of solving the inverse task of gasdynamics.

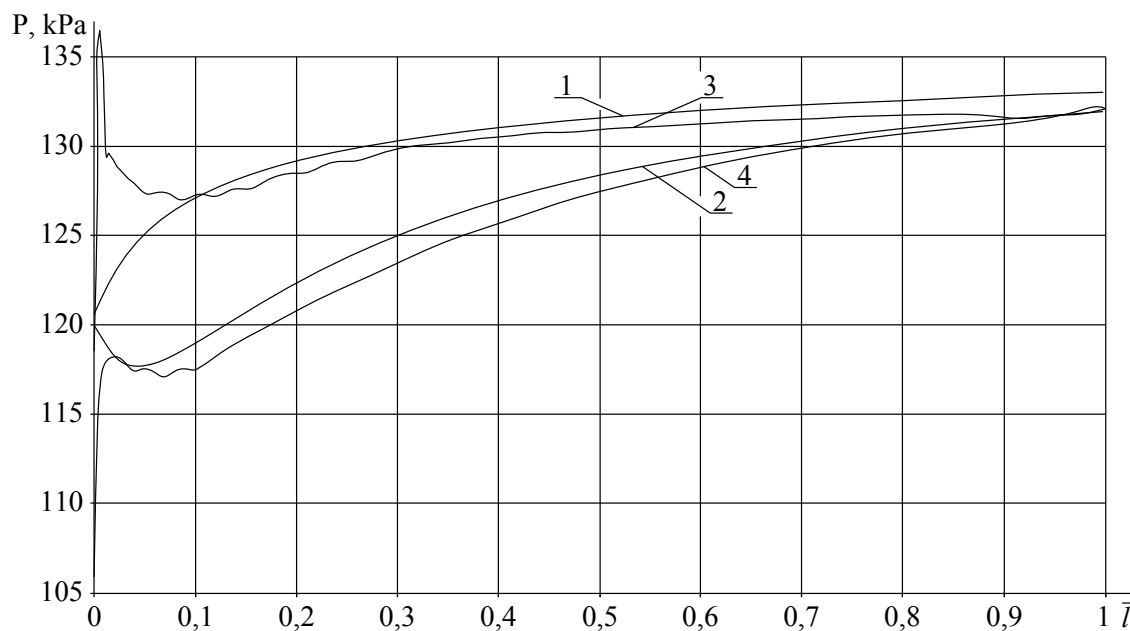
The values of static pressure along the surface of the vane in the proposed method are determined using gas-dynamic function:

$$P_i = P^* \cdot \left(1 - \frac{k-1}{k+1} \cdot \lambda_i^2\right)^{\frac{k}{k-1}}. \quad (16)$$

where the value of speed ratio for the pressure surface of the blade is determined from the equation (5), while for the suction surface - joint from the solution of equations (6) and (9).

Diagrams of the static pressure in the software package ANSYS CFX were determined in Post processor of solver along the line, located in the middle of the vane height.

Comparison of changes in static pressure along the surface of the blade (Figure 10) was carried out for the diffuser with given velocity distribution on the surface of the vanes, provided in Section 5.1. Pressure diagrams correspond to design condition of the investigated centrifugal compressor stage.



**Figure 10.** Static pressure distribution. 1, 2 – on the pressure and suction surfaces of the vane according to suggested method, respectively, 3, 4 – on the pressure and suction surfaces of the vane according to ANSYS CFX, respectively

A comparison of distribution of static pressure along the vane surfaces, obtained by solving the inverse task of gasdynamics and numerical modeling, shows qualitatively satisfactory agreement. The maximum deviation of the compared values of the pressure does not exceed 3%, which indicates on the correct choice of the mathematical model of the proposed method for designing vaned diffusers. The discrepancy between the values of the static pressure distribution along the vane surface at its inlet section is explained by taking into account the constraint of flow in the transition from vaneless space between the impeller and vaned diffuser in the numerical modeling.

## 7. Conclusions

The suggested method of designing the diffuser vane with the given velocity distribution on the vane surfaces allows to extend the range of the effective flow deceleration up to 80% in comparison with the circular-arc mean camber line vaned diffusers. The analysis of variants with different design parameters of the vaned diffuser allowed to draw the following recommendations for the design of high-performance diffuser with a given velocity distribution on the surfaces of the vanes:

- in the case of application the airfoil C-4 vaned diffuser gas-dynamic characteristics are more gently sloping in comparison with the diffusers with a constant vane thickness;
- the vaned diffuser with a vane-spacing ratio  $l/t=1.5$  has the lowest losses;
- the minimal values of the loss coefficient and the coefficient of static pressure recovery achieved for the width of the diffuser  $b_3=1.2 b_2$ ;

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