VANELESS DIFFUSER FOR LOW FLOW RATE CENTRIFUGAL COMPRESSOR STAGE

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ABSTRACT

Experiments and CFD-analysis of vaneless diffusers demonstrate that in wide range of relative width there is a risk of flow separation at design flow rate as a flow angle after an impeller is too small. To avoid it the inlet part of a diffuser is made with decreasing width along radius.

By the elementwise calculation the diffuser with relative width about 2%, narrowed initial part and conical surfaces with the divergence angle 3^0 between them is turned out to be the best variant of the considered.

The diffuser within the stage with narrowed initial part and zero divergence angle has the highest efficiency when relative flow rates are less than design regime flow rate. The stage with diffuser with walls divergence angle $0^045'$ after narrowed part is less effective on 0.5% when relative flow rates are less than design regime. The maximum efficiency of this stage is 1.5% higher and efficiency is sufficiently higher at big flow rates.

KEYWORDS

vaneless diffuser, centrifugal compressor stage, efficiency, loss coefficient, recovery coefficient, divergence angle

NOMENCLATURE

b	width of channel
$\bar{b} = b / D_2$	relative width of channel
c	velocity
D	diameter
$\bar{D} = D / D_2$	relative diameter
k	isentropic coefficient
M_u	blade Mach number
p	pressure
Re_u	impeller diameter Reynolds number
T	temperature
α	inlet flow angle
Φ	flow rate coefficient
ψ_i	work coefficient
ψ_p	polytrophic work coefficient
ψ_T	loading factor
η	efficiency
ζ	loss coefficient
υ	divergence angle
ξ	recovery coefficient

SUBSCRIPTS

des	design
0', 1, 2, 3, 4	indices of control sections
*	stagnation parameters

INTRODUCTION

Gas compression is required for technological processes in all industries. In plants with a large mass flow rate, centrifugal compressors are most widely used. Their power in some cases reaches 64 000 kW.

Gas-dynamic design firstly must determine dimension and shape of the compressor flow part, at which the specified final pressure will be provided at a specified mass flow rate. From an infinite number of possible solutions, one should choose the optimal one. Compressors operate under different conditions of the serviced network. Pressure change in the network leads to a change in flow rate, efficiency and power consumption.

The equations of a gas motion are second-order partial differential equations that cannot be integrated. Numerical solutions by the method of computational gas dynamics appear to be a suitable tool (see Bourgeois et al., 2015, Harley et al., 2015, Janke et al., 2016, Kosprdova et al., 2008, Kowalski et al., 2012, Kryłłowicz et al., 2017, S Lurie et al., 2011, Marechale et al., 2015, Meduri et al., 2015, Monjea et al., 2014, Prasad et al., 2011, Robinson et al., 2012, Sorokes, 1995).

The centrifugal compressor stages with low flow rates have flow angles at an impeller exit lower than 20^0 at the design flow rate. In vaneless diffuser (VLD) with the width of diffuser equals the height of impeller's blades the flow separation is very probable. Diffusers with narrowed initial part and parallel walls of the main part are used for stages with low flow rates (see Galerkin, 2010). In such diffusers there is no flow separation at the design regime, but the increased level of friction losses and decreased recovery coefficient cause decrease of efficiency.

The paper Galerkin et al. (2015) presents the results of the computational analysis of the vaneless diffuser with b_3 / D_2 =0.014 and conical surfaces with divergence angle υ = $0^0 28'$ - 6^0 between walls. Calculations have shown that in narrow diffuser with a small walls divergence angle shear stresses are high. It prevents the flow separation. In the main part at the some divergence angle flow separation does not occur, but the friction losses are less, and the coefficient of recovery is higher.

THE OPTIMIZATION OF THE VANELESS DIFFUSER

The problem of a vaneless diffuser is that it is most effective at the flow inlet angle $\alpha_2 = 90^0$. (see Jansen,W.) The smaller is the flow angle – the higher are friction and separation losses (see Seleznev, Galerkin, 1982). The aim on an optimization is to obtain highest possible pressure at a diffuser exit. The impeller exit relative width, VLD radial length and flow inlet angle are given. It means that the recovery coefficient $\xi = \dfrac{p_4 - p_2}{\rho \cdot 0.5\left(c_2^2 - c_4^2\right)}$ must be as high as possible.

The object of optimization is a vaneless diffuser for an impeller with relative width b_3 / D_2 =0.026. The design flow rate coefficient $\Phi_{des} = \bar{m}_{des} / \left(\rho^*_{inl} \dfrac{\pi}{4} D_3^2 u_2 \right)$ is 0.015. The relative VLD radial length is D_4 / D_2=1.743. The flow angle at the impeller exit inlet angle $\alpha_{2des} = 16^0$.

The next candidates are compared: candidate 1 is a diffuser with width which equals the height of impeller's blades; candidates 2-7 - combine narrowed initial part and main part with conical surfaces with walls' divergence angle υ = 0^0 (Var. 2), 1^0(Var. 3), 2^0 (Var. 4), 3^0 (Var. 5), $3^0 35'$ (Var. 6), 5^0 (Var. 7). Candidates ##1, 2, 5 are shown on Figure 1.

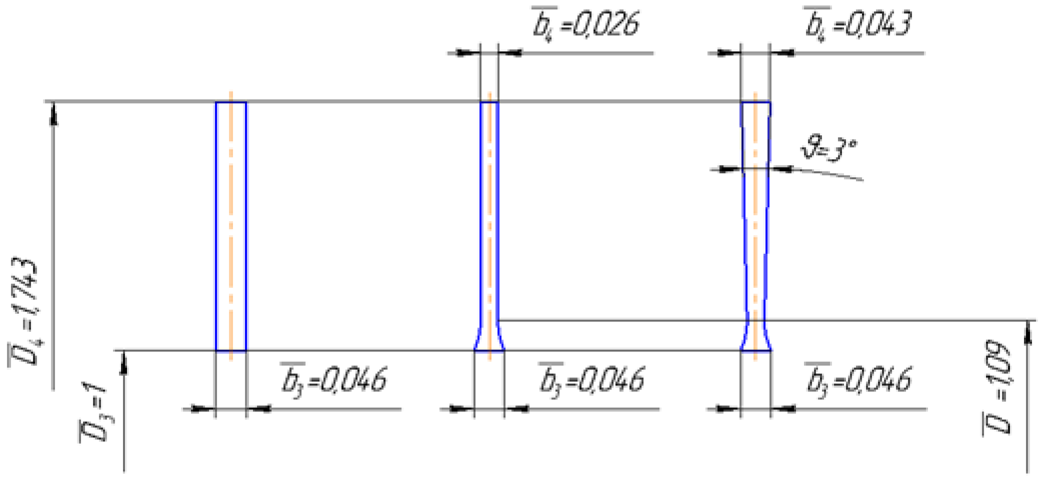


Figure 1: VLD candidates ##1, 2, 5

The program ANSYS CFX and the net generator ICEM, H-type structured nets with 90816 elements, $k-\varepsilon$ turbulence model were applied to calculations. The total pressure, total temperature and the flow angle in projections on the cylindrical coordinate axes were set at the inlet, the mass flow rate was set at the outlet. The ANSYS CFX solver uses a second order discretization scheme by default, providing the most accurate results. Other solver settings were set: CFL number = 3; the convergence criterion of the calculations was the RMS H-Energy value is below 1.0 E-3. Calculations by CFD (Computational Fluid Dynamics) method show that diffuser candidate #1 is unacceptable because of flow separation at the design flow rate α_{2des} = 16^0. Flow in vaneless diffusers with narrowed initial part and constant width (candidate #2) and VLD with the divergence angles 1- 3,35^0 (candidates ## 3 -6) does not separate in the range of flow inlet angles α_2 = 10 – 35^0.

Below are compared gas dynamic characteristics of the VLD candidates.

Polytrophic efficiency:

$$\eta = \frac{\lg \frac{p_4}{p_3}}{\frac{k}{k-1}\lg \frac{T_4}{T_3}} \qquad (1)$$

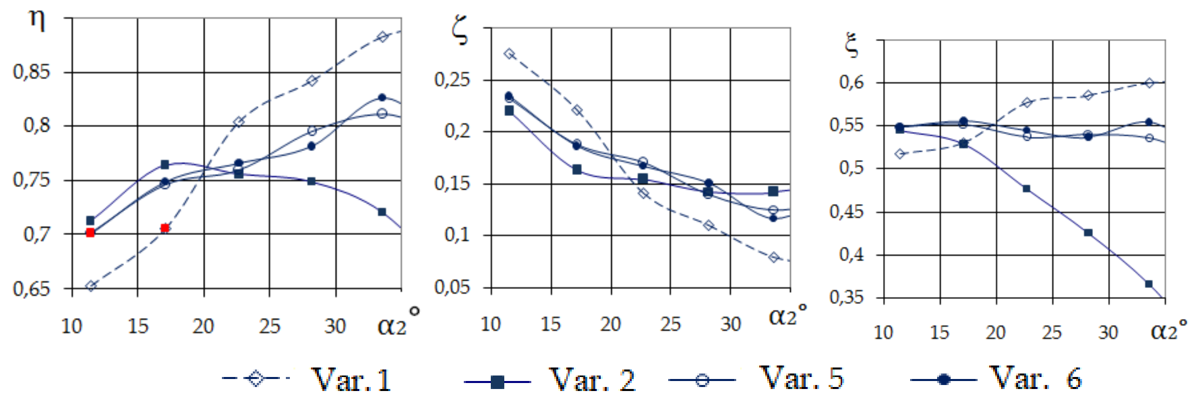
loss coefficient:

$$\zeta = (1-\eta)\left(1-\left(\frac{c_4}{c_3}\right)^2\right), \qquad (2)$$

recovery coefficient:

$$\xi = 1-\left(\frac{c_4}{c_2}\right)^2 - \zeta . \qquad (3)$$

The characteristic of candidates ##1, 2, 5, 6 are presented in Figure 2.



a) b) c)

Figure 2: VLD candidates' characteristics a) efficiency, b) loss coefficient, c) recovery coefficient

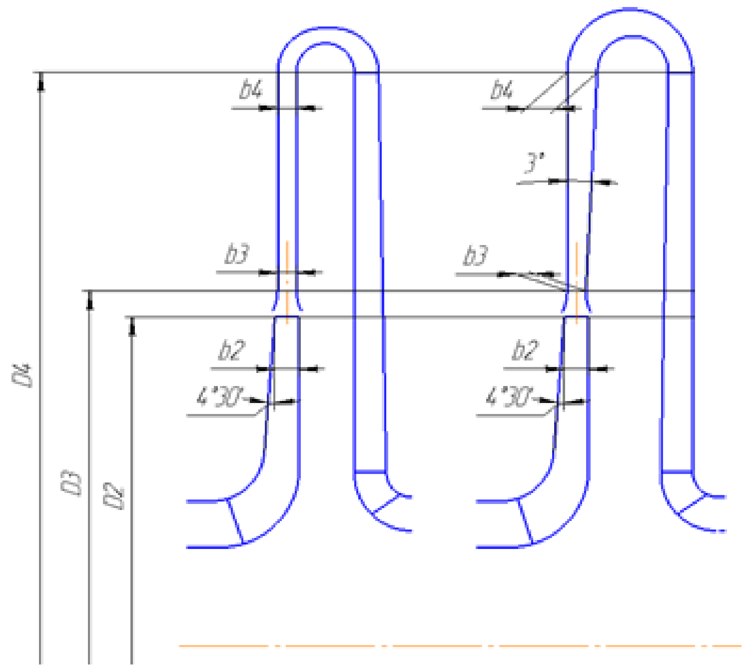
Red points indicate flow separation flow angle. VLD candidate #1 is not acceptable due to flow separation and low efficiency at $\alpha_2 \le \alpha_{2des}$ = 16^0. In the area of flow angles $\alpha_2 < \alpha_{2des}$ = 16^0, the diffuser with narrowed initial part and constant width of the main part (candidate #2) has the highest efficiency and the lowest loss coefficient. But its recovery coefficient is infertior. The diffusers with divergence angle υ =3^0 and $3^0 35'$ after the narrowed part have practically identical characteristics. They are more effective at flow angles $\alpha_2 > \alpha_{2des}$ = 16^0, and demonstrates higher recovery coefficient at $\alpha_2 \le$ = 20^0. Higher recovery coefficient means lower flow kinetic energy that leads to smaller loss of pressure in a return channel or in a scroll.

Vaneless diffuser operational conditions inside a stage flow part differ from studied above. Flow that leaves an impeller is non-uniform. A bend of return channel creates acceleration-deceleration zones in a diffuser (see Galerkin et al., 2015). It may result in an early flow separation. The objective of the contribution below is to study possible application of VLD with divergence angle in a low flow rate centrifugal compressor stage.

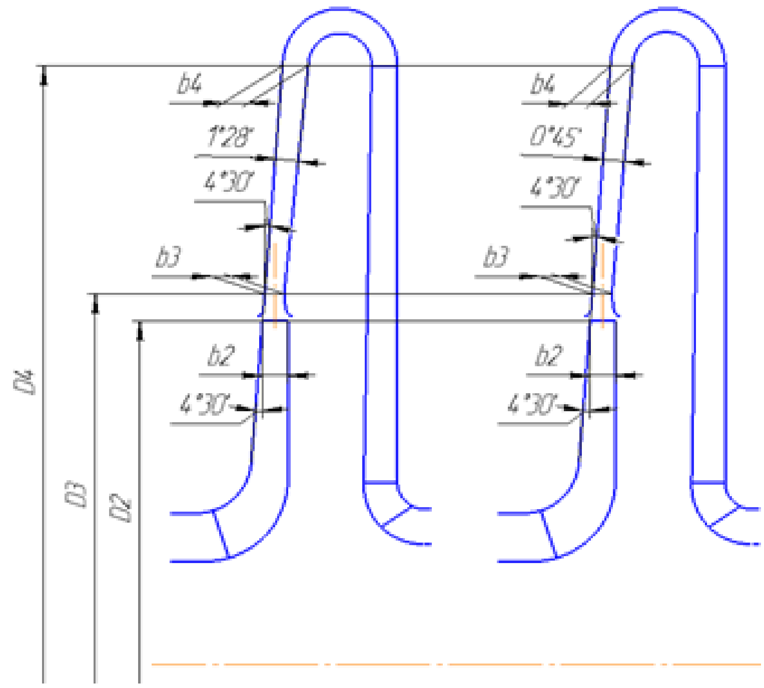
THE PARAMETERS OF LOW FLOW RATE CENTRIFUGAL COMPRESSOR STAGE

The parameters of low flow rate centrifugal compressor stage: flow rate coefficient Φ_{des} =0.015, loading factor $\psi_{T\,des} = c_{u2} / u_2$ =0.5. Similarity criteria: isentropic exponent k =1.4, Reynolds number Re_u =5 000 000, Mach number M_u =0.55.

Four considered vaneless diffusers in the stage flow part are shown in Figure 3.



Var.1 Var.2



Var.3 Var.4

Figure 3: Four VLD candidates in the stage flow part

Dimensions of vaneless diffuser candidates: relative radial length $\overline{D_4}$ =1.743, relative width at the inlet $\overline{b_2}$ =0.0371, relative width of the narrowed part $\overline{b_3}$ =0.0286. Candidate #1 $-\overline{b_3}=\overline{b_4}$, candidate #2 $-\overline{b_4}$ =0.0457, $\upsilon=3^0$; candidate #3 - $\overline{b_4}=\overline{b_2}$ =0.0371, $\upsilon=1^0 28'$; candidate #4 – $\overline{b_4}$ =0.0329, $\upsilon = 0^0 45'$.

The front wall of vaneless diffuser candidates ##3, 4 is the conical surface that is continuing the shroud of the impeller. This configuration diminishes flow turn that is always preferable.

CHARACTERISTICS OF LOW FLOW RATE CENTRIFUGAL COMPRESSOR STAGE WITH FOUR VANELESS DIFFUSER CANDIDATES

The configuration of VLD influences an impeller work coefficient characteristic. It was proven experimentally (see Seleznev, Galerkin, 1982). But this influence less important that influence on a stage efficiency. The higher stage efficiency is the goal of VLD configuration improvement.

The stage with VLD candidate #1 was designed by means of Universal modeling method (see Galerkin, 2010). Diffuser design includes narrowed initial part for increasing the flow angle at the entrance to vaneless diffuser, which is necessary for preventing the flow separation at flow angles below 15-18^0.

VLD candidate #2 with divergence angle 3^0 has demonstrated the better characteristics among studied above.

VLD candidate #3 with angle υ=1^{0}28' and equal widths at the entrance and at the exit, $\overline{b_4} = \overline{b_2}$. It was supposed that recovery coefficient of this variant would be the same as by vaneless diffuser with parallel walls. Variant 4 has similar to variant 3 profile design, with lower walls divergence angle of the main part, υ=0^{0}45'.

Calculation analysis was conducted with the usage of NUMECA Fine/Turbo program. For flow calculation the grid was formed by NUMECA Fine/AutoGrid. The nets have 2.03, 2.30, 2.17, 3.27 millions of elements for variants 1 - 4 respectively. Grid independence was reached for all variants, i.e. increasing the number of elements didn't influence the calculation results. According to NUMECA recommendations one-parameter turbulence model Spalart-Allmaras is used. The total pressure and total temperature were set at the inlet, the mass flow rate was set at the outlet. Other solver settings were set: CFL number = 3; central spatial discretization scheme; convergence criteria = -10.

The stage and vaneless diffuser candidates gas dynamic parameters are presented below as functions of Φ. The stage total polytrophic efficiency, work coefficient and polytrophic coefficient:

$$\eta^* = \frac{\ln \frac{p^*_{0'}}{p^*_1}}{\frac{k}{k-1} \ln \frac{T^*_{0'}}{T^*_1}}, \tag{4}$$

$$\psi_i = \frac{\frac{k}{k-1} R\left(T^*_{0'} - T^*_0\right)}{u_2^2}, \tag{5}$$

$$\psi^*_p = \psi_i \cdot \eta^*, \tag{6}$$

The stage characteristics with four vaneless diffuser candidates are presented in Figures 4 and 5.

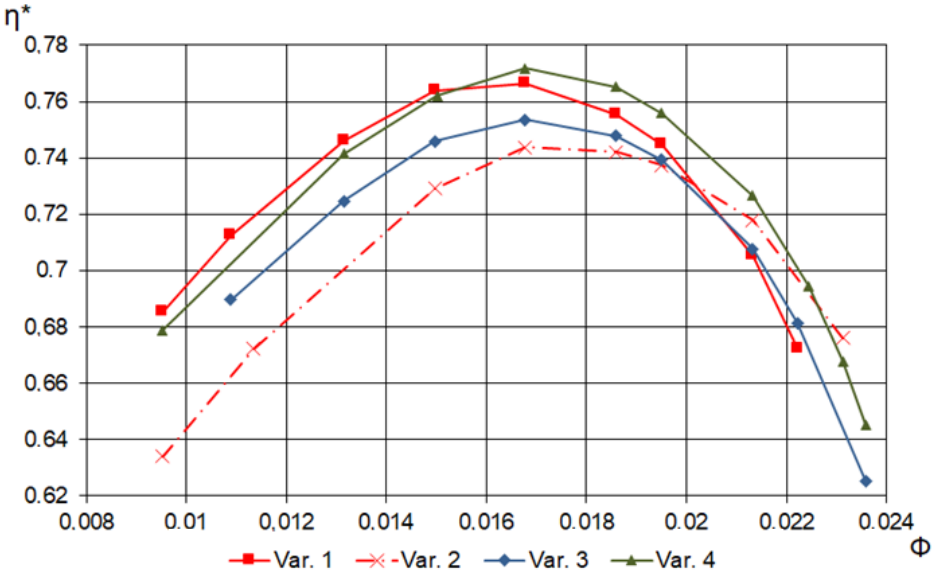


Figure 4: Efficiency of the stage with vaneless diffuser candidates 1- 4

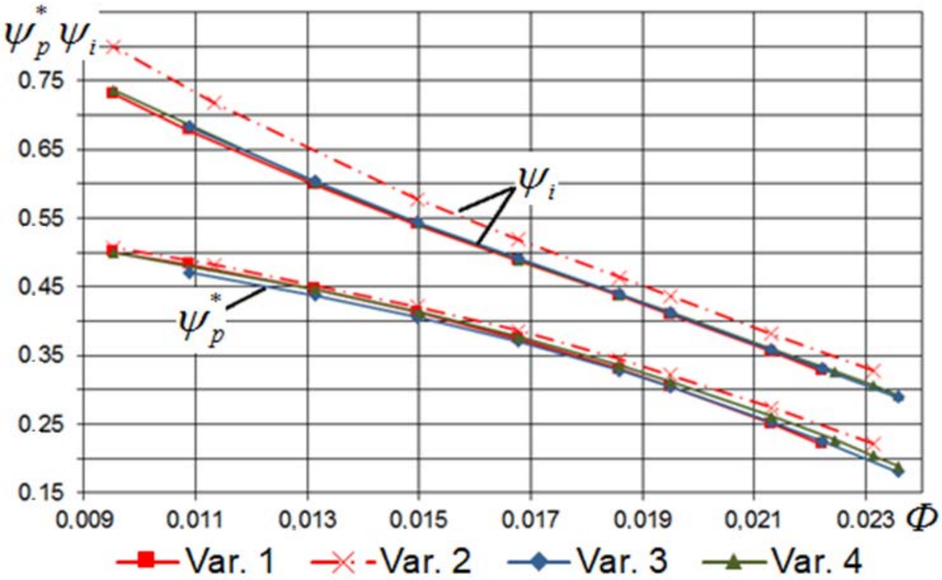


Figure 5: Polytrophic work coefficient and work coefficient of the stage with vaneless diffuser candidates 1- 4

The stage with VLD candidate #1 – common recommendation – shows predictable characteristic. The stage with VLD candidate #2 is substantially less effective. Walls' divergence angle 3^0 was optimal for the diffuser as an isolated element. It appeared to be ineffective as a part of a stage. The divergence angle $1^0$28'– candidate #3 – also appeared to be too big for a diffuser inside a stage.
The stage with vaneless diffuser candidate #4 with the divergence angle $0^0$45' has biggest efficiency, but the efficiency maximum is reached at the flow rate $\Phi = 0.0167 > \Phi_{des}$ (11% bigger).

In Figure 6 meridional structure of the flow is shown in static elements of the stage by design flow rate. Stream lines show that flow in VLD candidate # 2 is sufficiently separated.

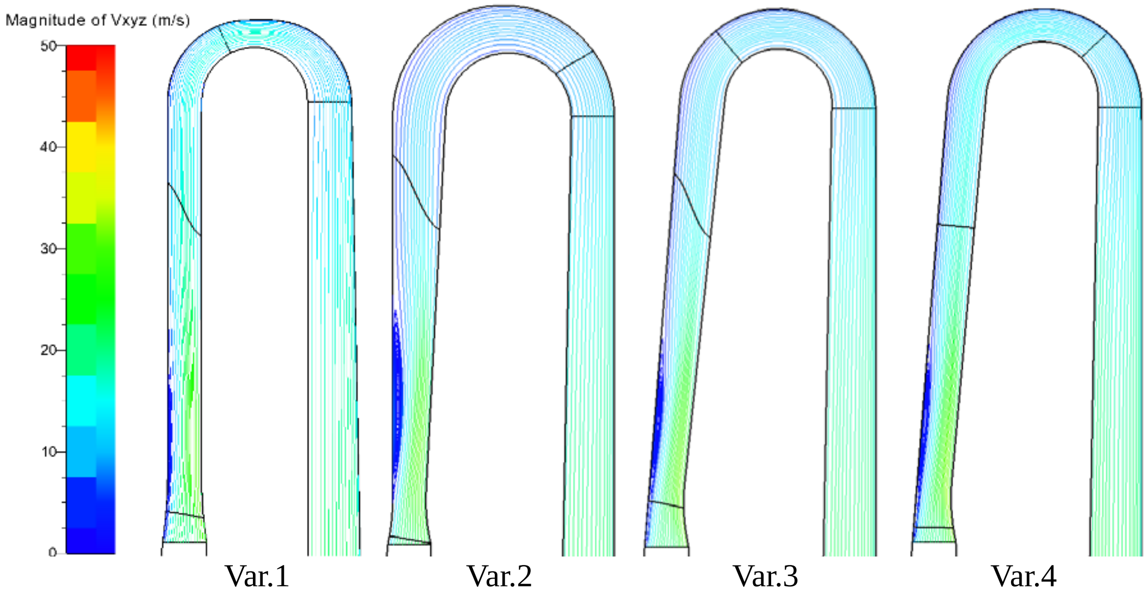


Figure 6: Stream lines in vaneless diffusers and return channel at the design flow rate

In the return channel crossover there is no flow separation for all diffusers. Local flow separation zones could be seen in all diffusers. The impeller coupled with the less effective VLD candidate #2 demonstrates maximum work coefficient. Why the highest separation zone in this VLD influenced impeller in this direction the Authors cannot explain. Commonly recommended VLD candidate #1 has the smallest local flow separation zone on the outer wall near the narrowed part. VLD candidate #2 with the biggest walls divergence angle shows substantial flow separation zone. VLD candidate #3 has smallest flow separation zone.

The characteristics of efficiency, loss coefficient, recovery coefficient, velocity ratio in vaneless diffusers are presented in Figure 7 - 10.

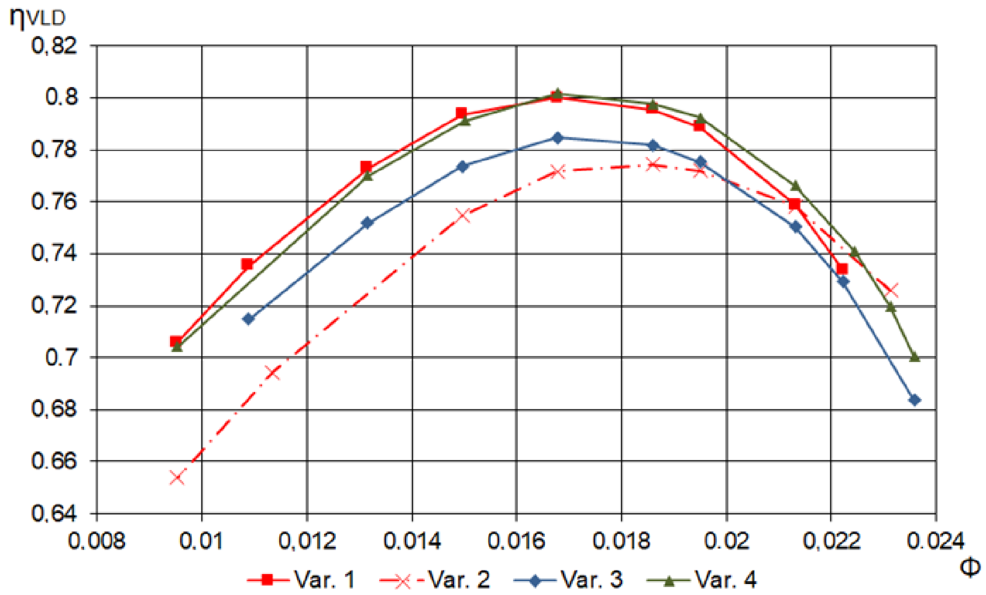


Figure 7: Efficiency of vaneless diffusers 1 – 4

The efficiency of diffuser variants is similar to efficiency of stages with these diffusers.

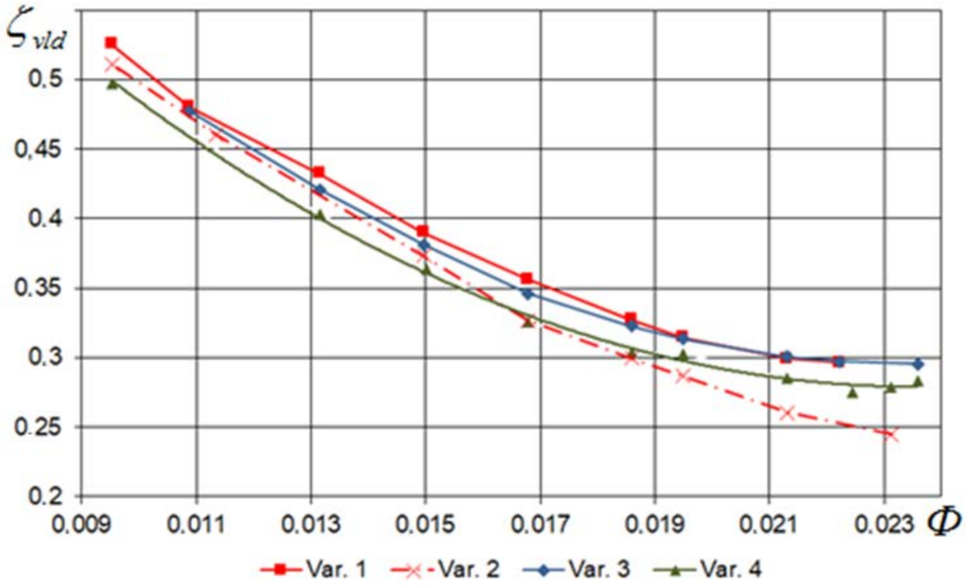


Figure 8: Loss coefficients of vaneless diffusers 1 – 4

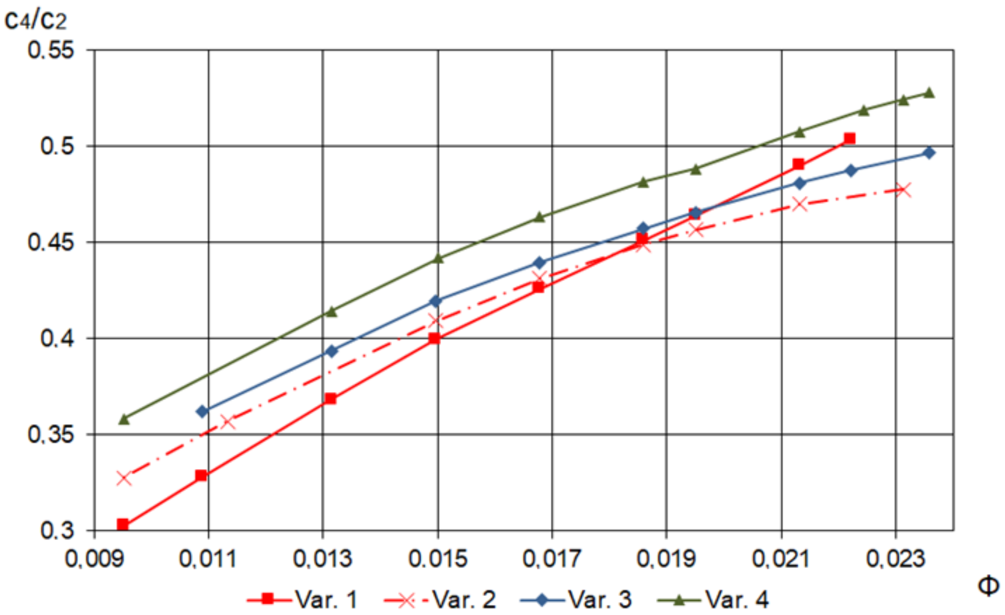


Figure 9: Velocity ratio in vaneless diffusers 1 - 4

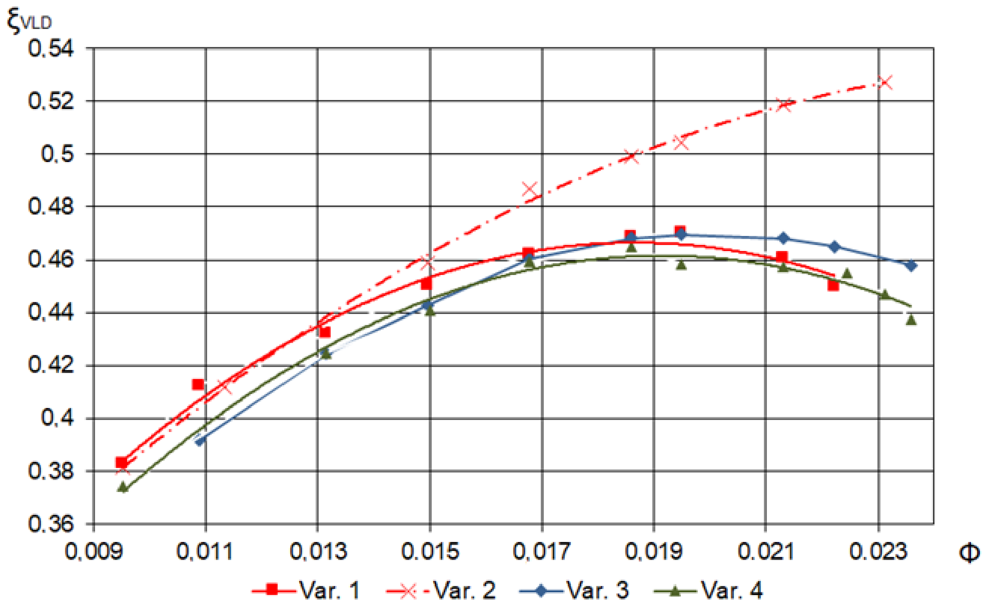


Figure 10: Recovery coefficients of vaneless diffusers 1 – 4

VLD candidate #1 with the narrowed initial part and parallel walls of the main part is the most effective at the regimes below the design point. But the loss coefficient of this diffuser is the highest at $\Phi \leq \Phi_{des}$ =0.015.

VLD candidate #2 is effective at the flow rates $\Phi \geq$ 0.020.

All parameters of VLD candidate #3 are inferior compared to VLD candidate #4. The last has little walls divergence angle υ =$0^0$45’. At $\Phi > \Phi_{des}$ VLD candidate #4 provides the highest efficiency.

CONCLUSIONS

Flow simulation has shown that in narrow vaneless diffusers the dangerous separation of the flow is suppressed by shear stresses even at low flow angles.

Modeling of narrow vaneless diffusers with divergent walls showed the possibility to increase the recovery coefficient and to diminish flow kinetic energy at the inlet to the subsequent element of the flow part. This reduces pressure losses.

For the presented low flow rate stage the vaneless diffuser candidate with limited divergence angle of walls increases the stage efficiency of 0.6% in comparison with the traditional design.

In general, CFD modeling demonstrated a great potential for solving applied gas dynamics problems.

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