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Low Solidity Vaned Diffuser study

J. Hrabovský[1], J. Klíma[2], J. Vacula[3], M. Komárek[4]

Abstract: Increase of the compressors parameters can be achieved through optimal choice of the diffuser. In case with requirements on higher efficiency the vane diffusers are used, on the other hand if the wide operating range is required the vaneless diffusers are applicable but with lower efficiency. In this study we would like to present that advantages of both diffuser types can be combined. The advantage of vaned diffuser is higher efficiency and vaneless diffuser is typical with better stability in wide operating range. Both characteristics can be projected through Low Solidity Vaned Diffuser (LSVD). Identification of optimal number and shape of vanes was carried out on radial compressor diffuser supplied by PBS Turbo Velká Bíteš. To find optimal diffuser configuration, CFD simulations in CFX software were prepared.

Klíčová slova: diffuser, low solidity vaned diffuser, efficiency, CFD simulation

1. Introduction

Compressor is device composed of several parts. The three basic components of the compressor are impeller, diffuser and volute. All these parts were and also will be investigated. It is evident that each of these parts have some limits, but the limits are continually shifted. Higher pressure on operating conditions and customer requirements lead to more detailed analysis and investigation of each part of the compressor. This paper is focused on just one part of the compressor, on the diffuser. The diffuser is static component, with high impact on the final compressor characteristic. Big advantage of this part is that appropriate choice of the diffuser type influences behavior of the compressor. The diffuser is relatively cheap part with high impact on the compressor characteristic. Wide range of diffusers is available. From vaneless with wide operating range up to vaned with high efficiency. Good compromise between both types of diffusers is LSVD. The investigation of the low solidity diffuser was carried out by many authors. The first description of the LSVD was carried out by Senoo [1], [2]. Many other authors studied and developed LSVD experimentally [3] and numerically [4].

The function of the diffuser is to convert the kinetic energy leaving the impeller into the pressure energy to minimize the pressure losses of the compressor. Generally it is known that the more number of the diffuser vanes is presented the higher static pressure and efficiency in the diffuser is achieved. On the other hand the smaller number of the diffuser vanes the better operation at low mass flow rates. Therefore a vaneless diffuser has a wider characteristic than vane diffuser because a surge line is situated at smaller mass flow rate than vane diffuser. D. Japikse in [5] states that in vaneless diffuser a logarithmic spiral flow path is forming. The flow path by

[1] L. K. Engineering, s.r.o., Vídeňská 55, 639 00 Brno, hrabovsky@lke.cz
[2] PBS Turbo s.r.o., Vlkovská 279, 595 01 Velká Bíteš, Jiri.Klima@pbsturbo.eu
[3] L. K. Engineering, s.r.o., Vídeňská 55, 639 00 Brno, vacula@lke.cz
[4] L. K. Engineering, s.r.o., Vídeňská 55, 639 00 Brno, komarek@lke.cz

comparison of vaned diffuser the logarithmic spiral implies a longer fluid path than would be encountered in a channel or cascade diffuser. This could rise to greater losses of vaneless diffuser. In this paper we studied the fundamental knowledge about diffusers and applied it on LSVD. The length of the flow path was calculated and investigated through CFD simulations. Results of studied fundamentals were used on real LSVD and were confirmed by measurement.

1. Fundamentals study

To study the compressor fundamentals the PBS Turbo provided the reference geometry of diffuser with 11 vanes and vaneless diffuser. From reference vaned diffuser the other two diffusers with 19 and 5 vanes were created. Based on the reference geometry the CFD model in Ansys Workbench was prepared. A single impeller blade segment model with full diffuser and spiral was used see Fig 1. The computational domain consists of millions cells. This approach is chosen because the full diffuser model along with spiral ensures more realistic effect of non-uniform flow in the spiral case. This results in real feedback between designed diffuser vanes and the spiral. The rotating impeller segment has a periodic symmetry applied on both sides of the domain and an interface to the diffuser domain. This interface is of type mixing plane ("stage" in CFX). This type of interface performs a circumferential averaging of the flux through the corresponding boundaries - impeller outlet and diffuser inlet. The analysis is of steady-state type and SST (shear stress transient) turbulence model was used.

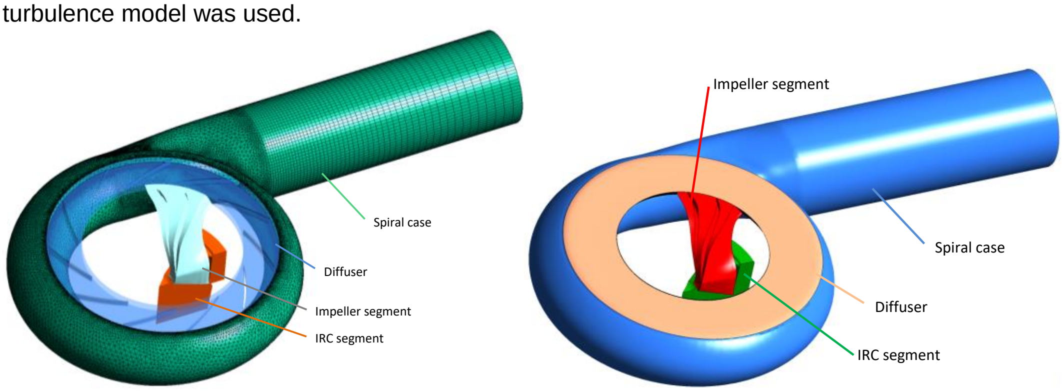

Fig 1. CFD domains with vaned diffuser (on left), vaneless diffuser (on right)

1.1 Boundary conditions

Boundary conditions respected operational conditions. Ideal compressible air was used for fluid domain. In Tab. 1. basic information about computational domain is shown. Calculations were carried out for corresponding angular velocities of the impeller. Inlet, outlet and interfaces are highlighted in green color in Fig 2. Boundary conditions were the same for vaned and vanless diffuser.

Tab. 1. Description of boundary conditions

Boundary conditions	
Inlet	Mass flow inlet, Total temperature at the inlet
Outlet	Static pressure outlet
Interfaces	
Interface 1: IRC - Impeller	Stage (mixing plane): 1/8 IRC segment to 1/8 impeller segment
Interface 2: IRC - Impeller	Stage (mixing plane): 1/8 IRC segment to 1/8 impeller segment
Interface 3: Impeller- Diffuser	Stage (mixing plane): 1/8 impeller segment to full 360° diffuser
Interface 4: Diffuser- Spiral	General connection

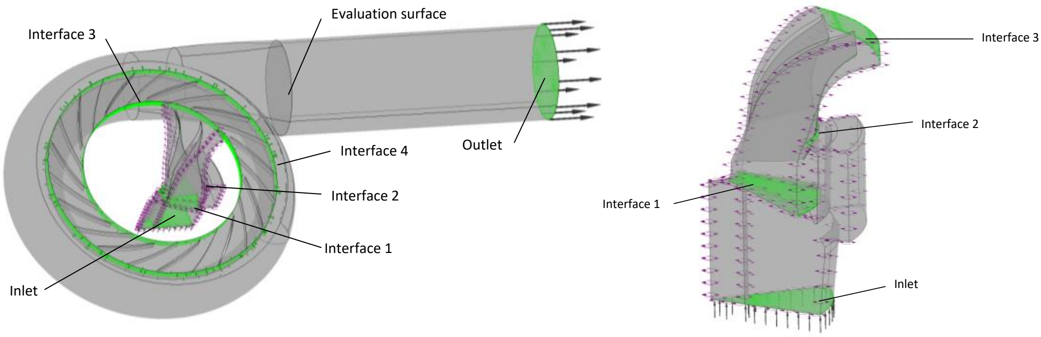

Fig 2. Boundary conditions

1.2 Results evaluation

Evaluation of the results was performed for all vaned and also vanless diffusers. Standard contours with pressure distribution and velocities were plotted (see Fig 5. and Fig 6.). The objective parameters of all calculated diffuser variants were evaluated based on the following equations:

$$\pi = \frac{P_{tot\_out}}{P_{tot\_in}} \qquad (1)$$

$$\eta = \frac{\pi^{\frac{\kappa-1}{\kappa}} - 1}{\frac{T_{out}}{T_{in}} - 1} \qquad (2)$$

$$\sigma = \frac{l}{\left(\frac{2\pi r_3}{z}\right)} \qquad (3)$$

π – Pressure coefficient

η – Efficiency

σ - Solidity

κ – Poisson number

P_{tot_in} – Total pressure at inlet

P_{tot_out} – Total pressure at outlet

T_{in} – Total thermodyna mic temperatur e at inlet

T_{out} – Total thermodyna mic temperatur e atoutlet

l - Chord length

r_3 – Radius at vane inlet

z – Number of vanes

Objective parameters were evaluated for all diffusers variants. The comparison of compressor characteristics, compression (1) and efficiency (2)) were calculated. Compressor characteristics were normalized by maximum values of corresponded variable (see Fig 3.). Next evaluated parameter is “solidity”, which defines the closeness of the vanes in diffuser. The solidity was calculated based on the equation (3) and it is presented in Fig 4. Optimum solidity of the diffuser is very important parameter, because it affects correct amount of flow rate through vanes. Solidity could be so high that in the radial direction the flow is enough deflected at low flow rates, but on the other hand must be so low that the smallest losses profile occurs at high flow profile. Effect of solidity is evident from Fig 5. and Fig 6., low and high flow rates (qm - normalized flow rate) for different level of solidity. The parameter solidity affects also length of the flow paths. The trajectory of the flow particle (flow path) in case of the vaneless diffuser (or small vane number diffuser –

LSVD) is obviously longer than the trajectory of flow the particle in case of the diffuser with higher vane number. Since the trajectory is longer the friction losses are higher and the efficiency is decreased. Increasing the diffuser vane number leads to shortening of the flow path and thereby to a compression and efficiency increase.

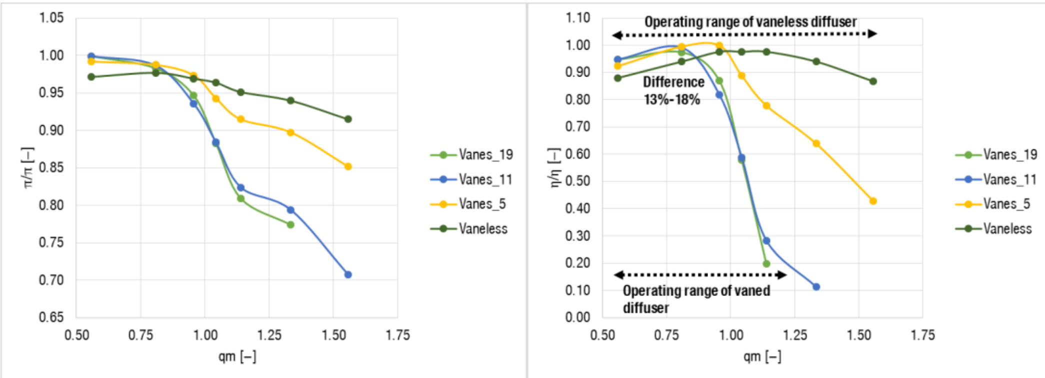

Fig 3. Comparison of compressor characteristics, Compression (on left), Efficiency (on right)

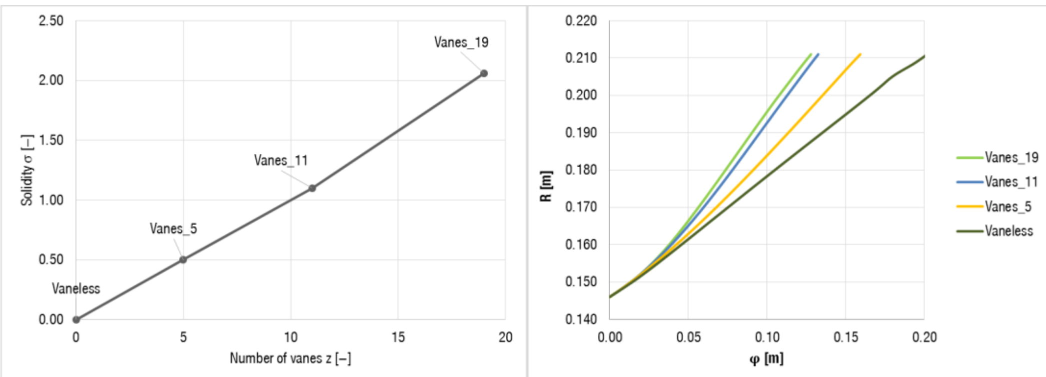

Fig 4. Solidity evaluation (on left), Flow path evaluation (on right)

A flow path can be described by the flow angle α. It is angle between velocity vector and radial direction. The flow angle is influenced by the diffuser vane shape and it varies in dependency on the radius R of the diffuser. Flow angle at radius R2 (diffuser inlet flow angle) is denoted as α_2. According to Japikse [5] in case of the vaneless diffuser a critical flow angle α_{c2} entering the diffuser can be evaluated. If the flow angle α_2 of the stream entering the diffuser inlet is higher than the critical flow angle α_{c2} a backflow in the diffuser occurs. In that case the entering flow is too much inclined towards tangential direction. In case the inlet flow angle α_2 is higher than critical flow angle α_{c2} and backflow in the vaneless diffuser tends to occur. Using the vanes into the vaneless diffuser forces the flow path to incline to the radial direction. Therefore shortening the flow path is achieved as well as efficiency increase. Calculated flow paths were circumferentially averaged and one representative flow path was obtained. The length of the flow paths φ for all calculated diffusers was evaluated (see Fig. 4 and Fig. 7). Based on Fig 4. it is evident that the more number of diffuser vanes is presented the shorter an average flow path is and vice versa. Moreover it is obvious that increasing the number of the diffuser vanes (>11) does not decrease the streamline length significantly.

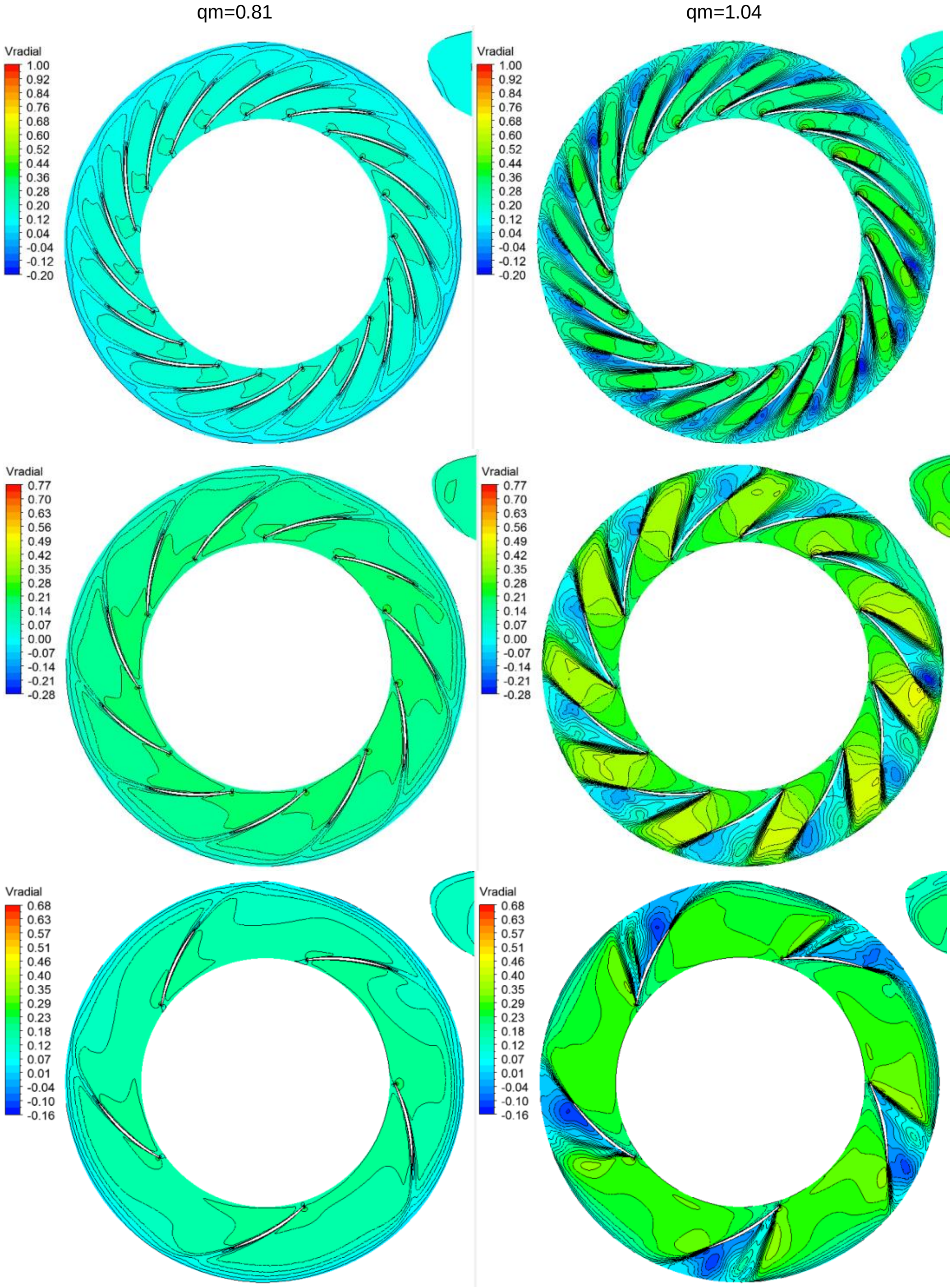

Fig 5. Normalized radial velocity of vaned diffusers

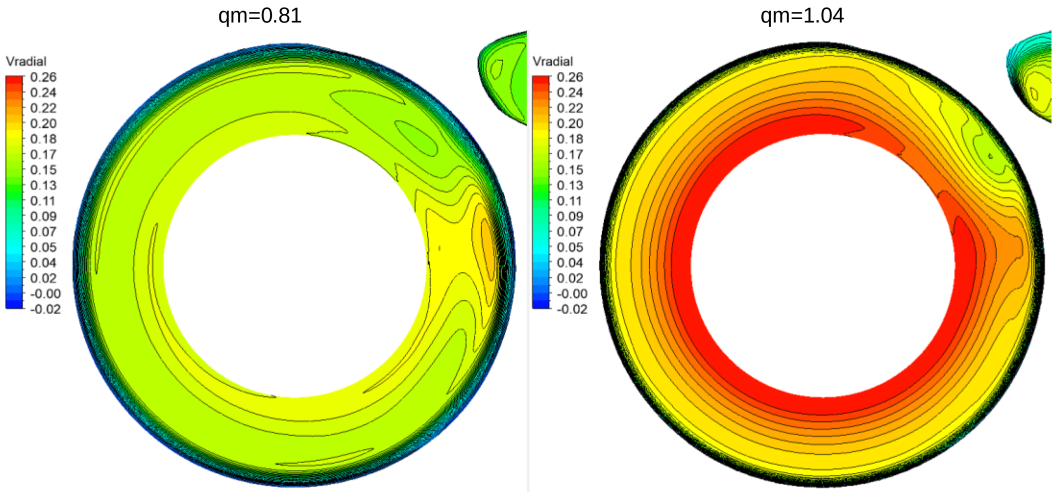

Fig 6. Normalized radial velocity of vanless diffuser

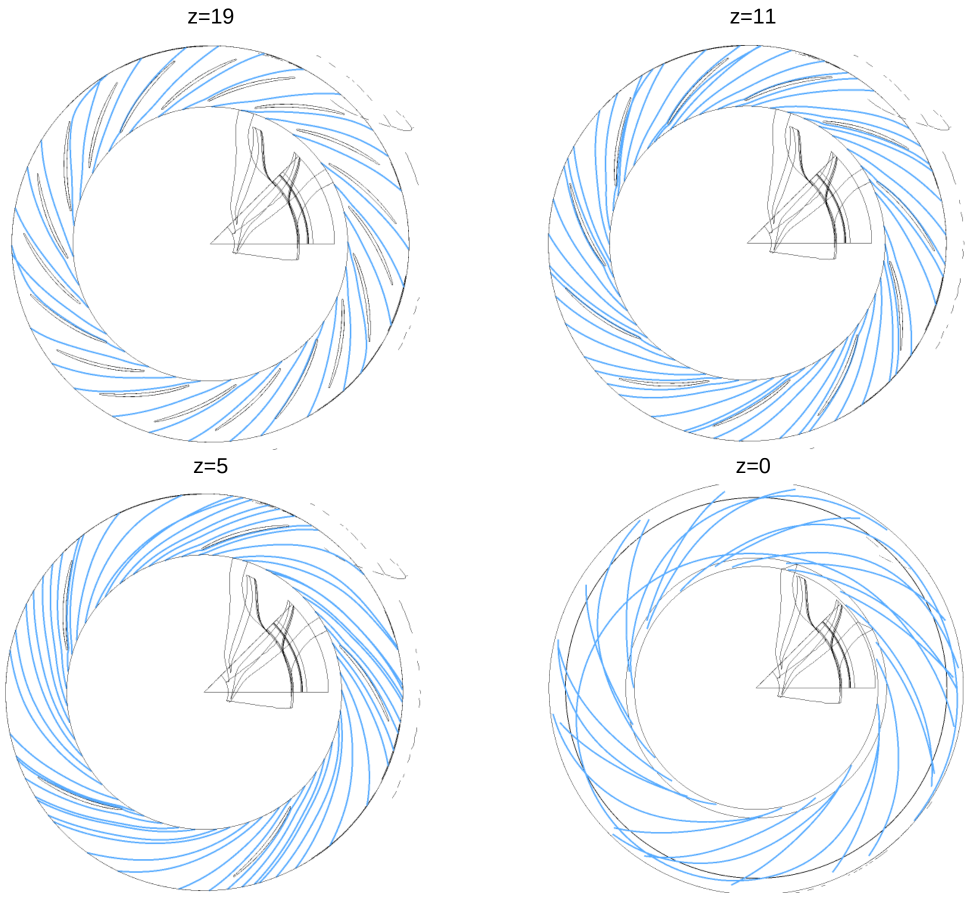

Fig 7. Flow path evaluation

2. Industrial Application

Results presented in previous chapter could be possibly applied for different industrial application. Common application is to find diffuser with appropriate characteristics. It was shown, that it is possible to optimize diffuser with wide operating range and with relatively high efficiency (LSVD). This application is described on real diffuser used in PBS Turbo. The requirement was to find diffuser with the same operating range and efficiency as vaneless diffuser characteristic, but stabilize the flow conditions and stabilize due to the structural reasons. The LSVD was chosen to meet defined requirements.

2.1 CFD model

The same methodology for CFD model creation as was described in chapter 1 was applied. From reference geometry of the compressor assembly the CFD domains were prepared (see Fig. 8). Also the same turbulence model and boundary conditions as in previous chapter were applied. The reference geometry of the diffuser contains five vanes with solidity value 0.19. This diffuser was defined through parameters and the parametric model for optimization process was created.

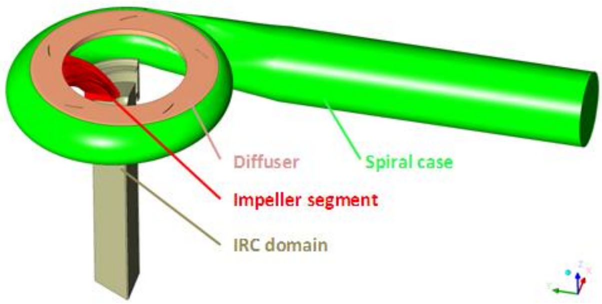

Fig 8. CFD domains

2.2 Parametric optimization

Parametric optimization of the diffuser was carried out in several steps. The first step was preparation of model of the diffuser whose shape was changed by several parameters. In this case two parameters were selected (φ-circumferential position, β3-vane angle) see Fig. 9. The range of the parameter β3 was limited by the size of the diffuser vane surface at hub and shroud. The nominal values of selected parameters correspond with values of reference geometry. In the second step the optimization method was applied. Parametric optimization methods can be used for optimization of structures of geometries based on defined input and output parameters.
The output parameters were obtained from CFD analysis. As the target function of the optimization the minimizing size of the backflow area in the diffuser was chosen. Minimizing this parameter as a target function was performed. Efficiency and compression were evaluated too.
On the basis of the optimization final variant was selected. The comparison of the reference and optimized variant is shown in Fig 9. Solidity of the optimized diffuser slightly decreased to vale 0.18. The same variables as in previous chapter were evaluated. The radial velocity of the reference and optimized diffuser was evaluated see Fig 10. The flow path of both diffusers is presented in Fig 11, the length of the flow path for optimized diffuser was shorter about 1% than for reference diffuser. From radial velocity evaluation and also from flow path evaluation it is evident that the vortex regions (position 1, 2, 3 in Fig 11.) were reduced and flow condition were improved for optimized diffuser.

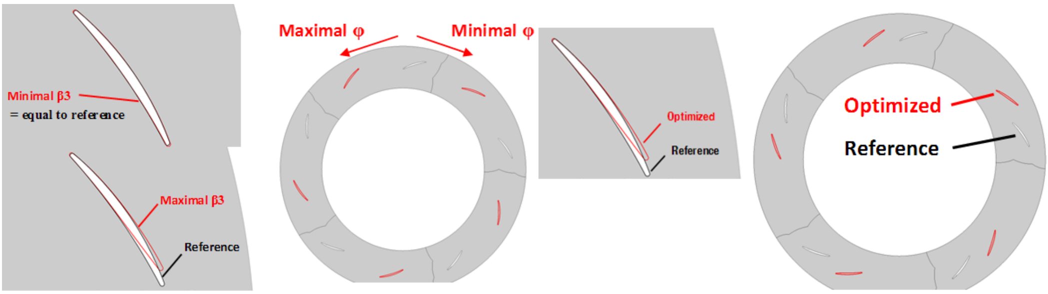

Fig 9. Illustration of parameters range (on left), optimized variant (on right)

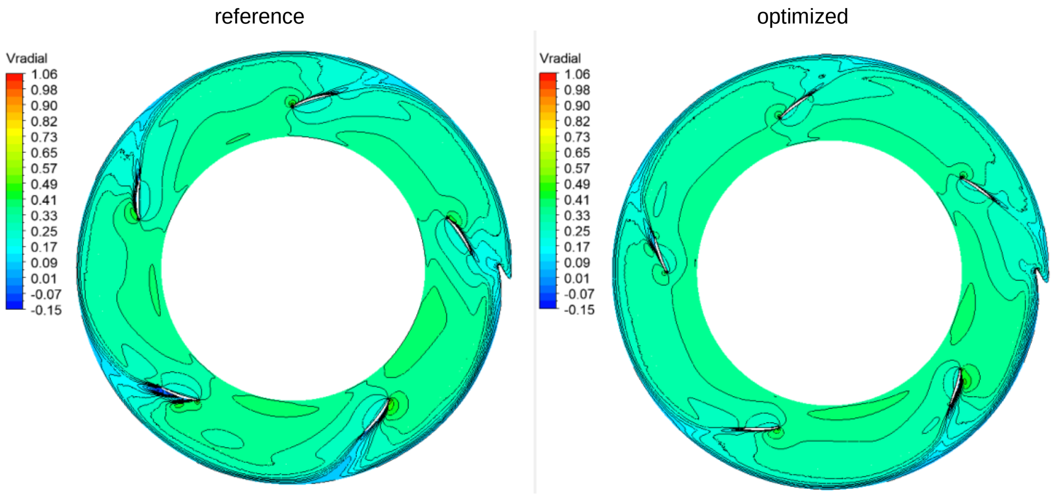

Fig 10. Normalized radial velocity of reference and optimized diffuser

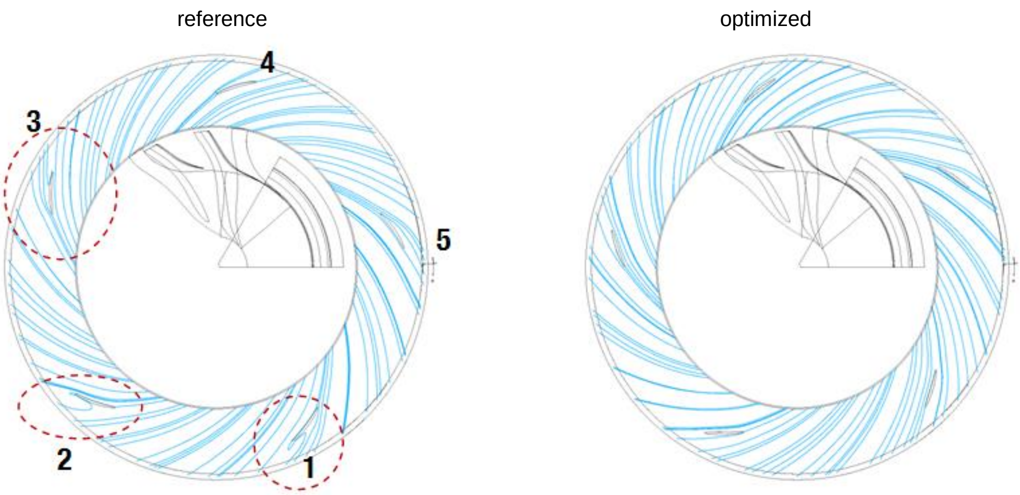

Fig 11. Flow path evaluation of reference and optimized diffuser

2.3 Measurement

To confirm improvements reached through optimization process the measurement of vaneless diffuser and optimized LSVD was carried out. The measurement was performed by standard measurement process in PBS Turbo. Measured characteristics of vaneless diffuser and LSVD are presented in Fig 12. respectively Fig 13. From measured characteristics it is evident, that requirements on wide operating range and efficiency were met. The optimized LSVD preserved required parameters and improved stability on surge line, moreover the efficiency was significantly increased about 2%.

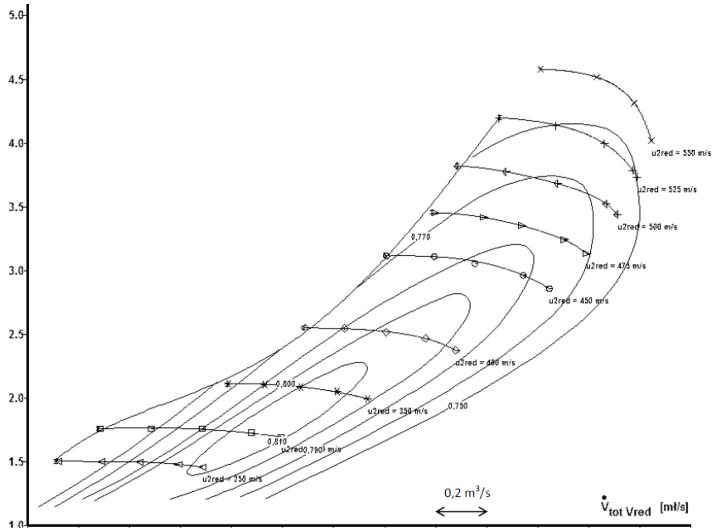

Fig 12. Vaneless diffuser measured characteristic

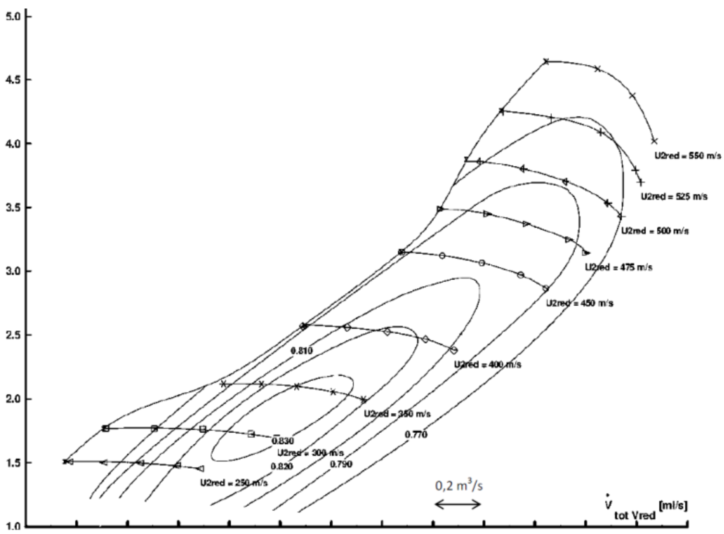

Fig 12. LSVD measured characteristic

3. Conclusions

In this paper the fundamental of the centrifugal compressors diffusers and LSVD were studied. The study of vanes impact on characteristics was carried out on real diffuser configuration provided by PBS Turbo. Main goal of the fundamental study was to confirm that the advantages of the vaned (high efficiency) and vanless diffusers (wide operating range) can be combined appropriately. As a suitable solution for this purpose the LSVD was found. Results obtained from CFD calculations clarified the LSVD benefits, which led to advantages combination. Shorter flow path has positive impact on the efficiency and stability of the operating range due to low vane number. The presented results from study are applicable for each diffuser, but for optimal solution it is necessary to find the best solution through optimization calculations. It was shown in chapter 2 (Industrial application). In this chapter the study results were applied on the different vanless diffuser and LSVD was optimized. Due to optimization process the appropriate position of vanes and solidity was found. Comparison of the reference vanless diffuser and optimized LSVD confirmed the positive impact on the compressor characteristics. The operating range of LSVD was nearly the same as vanless diffuser and efficiency increased about 2%. This relatively small changes in diffuser caused significant improvements of the compressor characteristics.

4. References

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