The Influence of the Vaned Diffuser on the Turbo machinery

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ABSTRACT

This paper analyzes, through numerical methods, the influence of a vaneless diffuser on the performances of a centrifugal compressor with two stages, close to surge line, to see how this is affecting the stable operating range. Two cases were studied in this paper: a) both stages having a vaned diffuser, b) first stage with a vaned diffuser and the second with a vaneless diffuser. The results obtained showed that a vaned diffuser has overall performances better that a vaneless one. Also, in this case scroll played an important role in differentiating the two cases.

KEYWORDS: *CFD, centrifugal compressor, vaned diffuser, vaneless diffuser, stall*

NOMENCLATURE

Latin
CFD – Computational Fluid Dynamics
EV – Valve through which the compressed air leaves the the discharge pipe
DV – Proportional discharge valve
VM – Manual valve
H - Total enthaply
M_a – Mass flow
n - Rotational speed
n_w– Working speed
$n_{protection}$ – Protection pressure ratio
p - Pressure
p_s - Selected pressure (required)
w – Molar mass

Greek
π_s– Selected pressure ratio
τ_{ij} - Viscosity tensor
ρ - Density

1 INTRODUCTION

In recent decades, the compressor design technique has been constantly improved, especially due to the development of CFD technology. Significant improvements in the performance of centrifugal compressors have been achieved by solving the three-dimensional Navier - Stokes equations by means of CFD. Experimental and numerical studies have shown that the diffuser has a significant importance on the stability limit of a centrifugal compressor, depending on the impeller design and on the interface between the impeller and the diffuser [2].
The mass flow, efficiency and pressure rise in a compressor are the three important parameters used in defining the performances of a compressor and in its selection. The efficiency and the pressure

ratio of centrifugal compressors depend on the design of the impeller and on the fluid flow inside the work channel, as well as on the losses in the diffuser [11].
Two-thirds of the total losses in a centrifugal compressor occur in the diffuser [2]. There are two types of diffusers, depending on their application. Vaneless diffusers have a wider operating range, lower efficiency and lower pressure recovery than vaned diffusers [1]. The space between the impeller tip and the diffuser vanes is critical to ensure an effective diffusion process and helps the uniformization of the impeller flow [2].
In the case of the vaneless diffuser the throat is absent; which is leading to a wide operating range. Performances of a vaneless diffuser are affected by the channel width and the radius ratio between the diffuser inlet and outlet [5].
In the case of the vaned diffuser, the positive incidence increases the likelihood of surge, while the negative incidence leads to blockage [6]. Surge can also occur in the vaneless diffuser [7]. Due to the influence of the negative pressure gradient, the boundary layers on the wall are deviated in a tangential direction rather than in the axial direction [8]. Generally, for vaneless diffusers, the pressure recovery is less than in the vaned diffuser case, up to 20%, and 10% less on a stage [17].
The vaned diffusers provides a better or similar discharge pressure compared to vaneless stators at a smaller diameter. A vaneless diffuser leads to a larger compressor dimension. A vaned stator allows the output angle of the diffuser to be controlled downstream of a volute [9, 16], which is important. There are 5 important design parameters that affect the overall performance of a diffuser: diffuser width, radius ratio, chord length, blades number and angle at diffuser inlet [10].
An increasing in fluid diffusion with decreasing of the flow rate will lead to compressor surge. Shaaban [12] showed that in a small turbocharger with vaneless radial diffuser the aerodynamic losses are 33-45% from the overall compressor aerodynamic losses at surge.
A numerical study regarding the rotating stall phenomena in a vaneless diffuser, using an incompressible viscous flow solver, was realized by Ljevar et al. [13]. It was found that the geometry design and the outlet and inlet flow conditions influence the stability limit.
In the case of vaned diffusers it was found that with the increasing of the number of vanes, the operating range narrows [14]. For a radial vaned diffuser, the most critical area, according to P. Delbert et al. [15], is the inlet triangle, or the “semi-vaneless” part, where the strongest backflow phenomena and the highest pressure gradients occur.
Vaned diffusers are used in many applications. Their impact on the operating range of a single stage compressor depends on parameters such as the Mach number at the inlet to the stator, the angle of inlet flow, the performance of the rotor and on the rotor - stator coupling [18].
In this paper, the influence of the vaned diffuser on the performances of a centrifugal two-staged compressor close to surge line is studied. The goal is to see how the range of stable operating regimes is affected. Two cases are studied: a) the centrifugal compressor with both stages having vaned diffusers, b) the centrifugal compressor with the first stage having a vaned diffuser, and the second one using a vaneless diffuser. Since the flow patterns on a diffuser are influenced by the velocity and flow angle distribution at the impeller outlet [15], those are the main characteristics to be surveyed.

2 PROBLEM FORMULATION

The novelty of the presented approach resides is the direct coupling between the two stages, allowing an assessment of the direct interaction between them. Also, the use of the vaneless contact section was taken into account in order to increase the range of stable regimes at which the compressor can work.

2.1 Mathematical Model

For this simulation, a compressible flow has been considered. The equations that govern the flow are the Reynolds-averaged Navies–Stokes equations, as presented below:

- Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial \rho u_j}{\partial x_j} = 0 \tag{1}$$

- Momentum equation:

$$\frac{\partial \rho u_i}{\partial t} + \sum \frac{\partial \rho (u_i u_j)}{\partial x_j} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i}\left(\tau_{ij} - \rho u_i' u_j'\right) = 0 \tag{2}$$

where $\tau_{ij} = \mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)$.

- Energy equation:

$$\frac{\partial}{\partial t}(\rho H) + \sum \frac{\partial}{\partial x_j}\left(\rho u_j H + \rho u_j' H' - k\frac{\partial T}{\partial x_j}\right) = \frac{\partial p}{\partial t} + \sum \frac{\partial}{\partial x_j}\left(u_i \tau_{ij} + u_i' \tau_{ij}\right) \tag{3}$$

- Equation of state:

$$\rho = \frac{wp}{R_0 T} \tag{4}$$

2.2 Geometry definition and spatial discretization

In this study, the analysis was performed using the commercial software ANSYS CFX. The computational domain consists of all the channels of the two stages of the analyzed centrifugal compressor, each stage with an shrouded impeller, vaned/vaneless diffuser and a volute, as shown in Fig. 1. The characteristics of the centrifugal compressor are presented in Table 1.

Table 1: Parameters of the compressor

Domain		No. of Blades	Angular velocity [rot/min]
First Stage	Impeller	11 (2 splitter blades)	37550
	Diffuser	18	
Second Stage	Impeller	15 (1 splitter blades)	41500
	Diffuser	20	

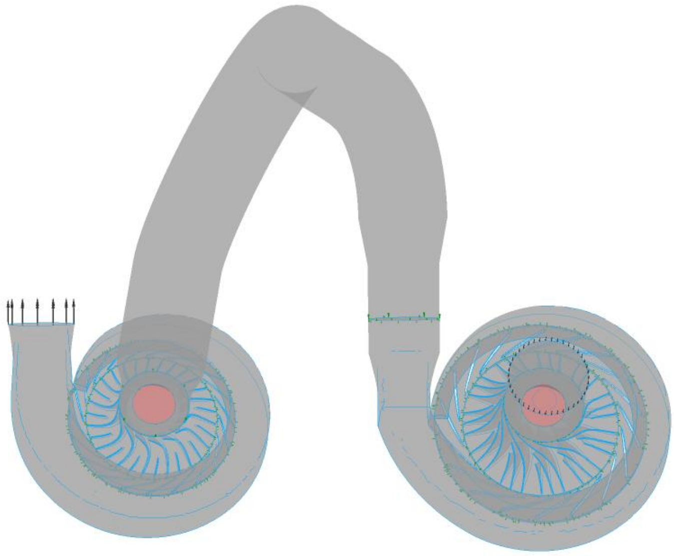

Figure 1: Geometry of the centrifugal compressor with vaned diffuser

The computational grid is structured for the impellers, diffusers and the contact section and unstructured for the volute. The structured grid for the impellers and diffusers was created using CFX – TurboGrid, while Ansys ICEM-CFD was used for the volute. The mesh obtained for the two cases has:

- Case with a vaned diffuser: 22.82 million nodes and 25.85 million elements;
- Case with a vanelles diffuser: 20.41 million nodes and 23.64 million elements;

The simulations are carried out for ideal air, stationary conditions, and the numerical scheme is of the second order in space.
At the interface between the rotational and the stationary parts, the *Frozen-Rotor* method has been used [19]. This type of interface is fixed; the relative orientation of the components is not changing, but depending on the motion defined by the domain the frame of reference is changing [19]. Between the stationary components a *None* setting is used, because these is no frame change or pitch change.

In defining the conditions of the domain, a reference pressure of 3 bar and a pressure ratio of 3.89 barg at the compressor exit have been used. In Table 2, the main boundary conditions of the computational domain are presented.

Table 2: Boundary conditions

Domain	Type of condition	Value	
First stage: entrance part	Inlet	Static Pressure	-2 [barg]
		Static Temperature	293 [K]
Second Stage: volute	Outlet	Mass Flow Rate	0.6 [kg/s]

The turbulence model used for this analysis is the k-omega SST (Shear Stress Transport) [3]. This model has been shown to yield accurate results, and is more suitable for such cases where it is important to capture both the phenomena occurring near the walls, and away from them [4].
Fig. 2 presents the work lines of centrifugal compressor (under standard conditions). In this case, the compressor characteristic was determined experimentally [18]. The surge line on which the machine protection line is set and the operating range of the machine were also established experimentally. The operating range of the compressor has been determined, resulting in a diagram illustrating the surge line - a line over which the compressor operation leads to its destruction.

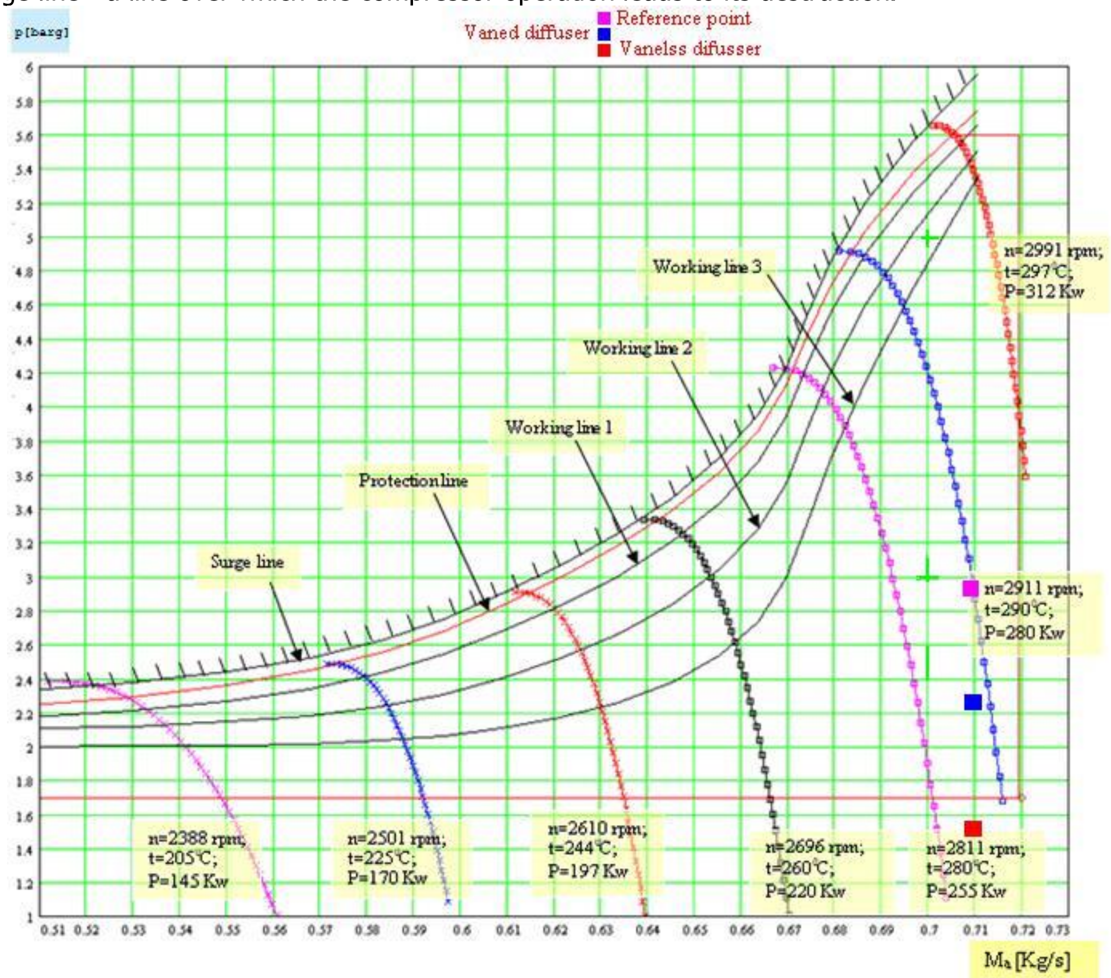

Figure 2: Work lines of the compressor (in standard conditions) [18]

In order for the operation to be safe, the compressor working range must be defined. The first measure to be taken is related to compressor surge - protection, for avoiding reaching the surge line.

So, another line below the surge line is defined at a certain distance, called the surge protection line. This line is part of compressor automatics.
The surge protection line is defined as: [18]

$$n_{protection}=3+3.35\cdot\left(\left(\frac{n}{\sqrt{t_{atm}+273}}-134\right)/44\right)^{1.7} \tag{5}$$

where: $n_{protection}$ – protection pressure ratio; n – speed [rot/min]
The compressor operating lines are [18]:

$$n1_w=\left(44\cdot((\pi_s-2.8)/3.9)^{0.625}+134\right)\cdot\sqrt{t_{atm}+273} \tag{6}$$

$$n2_w=\left(44\cdot((\pi_s-2.7)/3)^{0.67}+134\right)\cdot\sqrt{t_{atm}+273} \tag{7}$$

$$n3_w=\left(44\cdot((\pi_s-2.6)/2.3)^{0.71}+134\right)\cdot\sqrt{t_{atm}+273} \tag{8}$$

where:
π_s– selected pressure ratio
n_w – working speed [rot/min]

3 RESULTS

For the convergence of the solution, the residual history was monitored. A decrease of three orders of magnitude has been reached, obtaining a stable point where the fluid flow was in equilibrium. Also, the mass flow rate was monitored for both stages, at the first stage inlet and at the second stage outlet. For convergence, flow input must match the output within 5 x 10^{-3}.
To analyze the evolution of the fluid flow in the first compression stage, three planes were taken at 0.1, 0.5 and 0.8 of the impeller blade height, counting from the hub.

3.1 First stage:

Fig. 3 presents the total pressure variation in the impeller and diffuser of the first stage. As it can be seen, this variation is higher at the interface between these two components. This is due to boundary layer detachment from the blades and the consequent formation of recirculation zones. The pressure distribution is influenced by the rotor blade tip clearance. Differences between the two cases for the first stage, are minor, blade detachment and pressure losses appear in the same areas and between the two cases, the pressure difference is of nearly 0.12 bar.
In the diffuser, the appearance of shock waves leads to an increase in temperature, such that at the diffuser exit, the temperatures rises above 450 K. Shock waves are developing in the diffuser on the whole blade height and in each channel. Fig. 4 presents the velocity field in the rotating parts of the first stage. The appearance of shock waves limits the mass flow that can pass through flow channels of the rotating part. In the impeller, separation of the boundary layer occurs, but not as strong as in the case of diffuser.

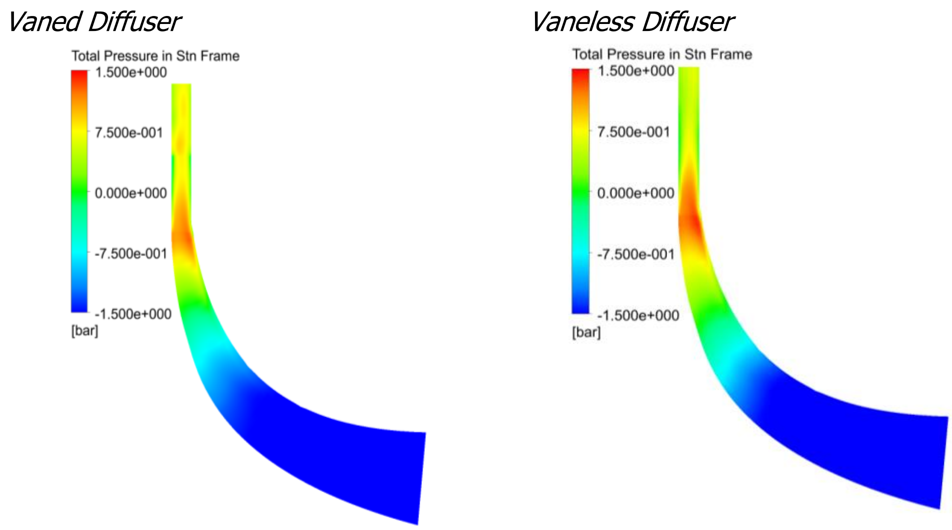

Figure 3: Total Pressure in Stn Frame

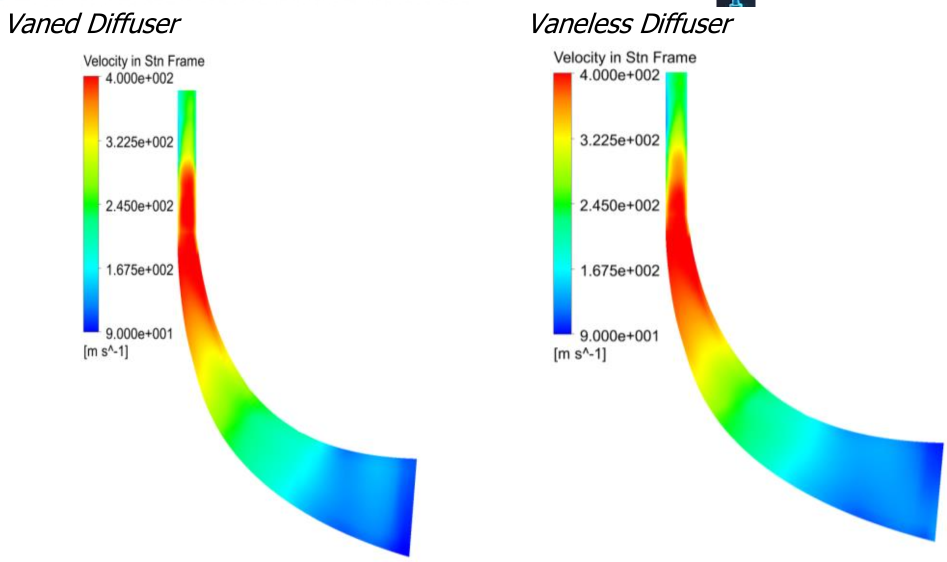

Figure 4: Velocity for the bladed parts of the first stage

Fig. 5 presents the vector velocity fields for the first stage. Here, differences in velocity between the two cases are also minor.

3.2 Second stage results:

To determine the flow properties in both diffusers, six control points in every channel of the diffuser, both on the blade suction side and on the pressure side, have been chosen, at three different blade heights: 0.1, 0.5 and 0.9 of the blade span. In the radial direction, the points were situated near the diffuser outlet, as shown in Fig. 6, at a radius of 114 mm. For a better estimation of flow properties, the data in Fig. 8 is averaged across the blade height.

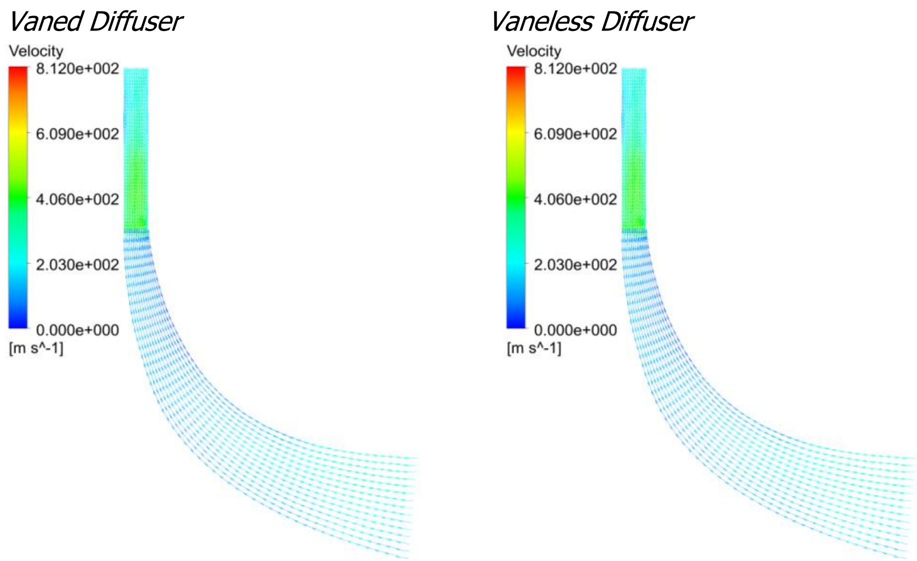

Figure 5: Vector field on the rotating parts of the first stage

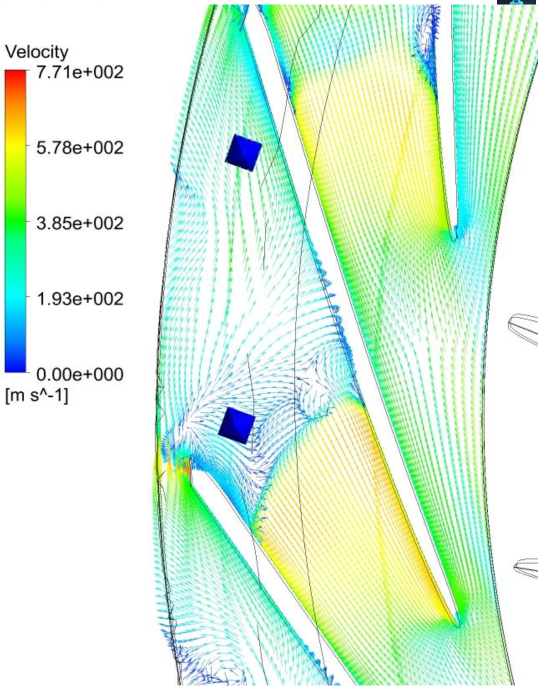

Figure 6: Vector field in vaned diffuser with two control points at 0.5 of the blade height

To determine the proper size of the first cell near the wall, Y+ was calculated to be approximately 1 near blade wall as shown in Fig. 7.

Fig. 8 presents the averaged total pressure starting from the volute tongue. Because of the small area of the volute, near the tongue, the total pressure is much higher. With the increasing of the volute area, its influence on the diffuser is decreasing, leading to a better compression. In the case of the vaned diffuser a part-span stall is developing. The closer it gets on the blade tip, the area of the stall is increasing. The pressure variation is due to the position of the reference points, one near the pressure side of the blade and the other near the suction side of the blade, and also reflects the recirculation zones that form in each channel.

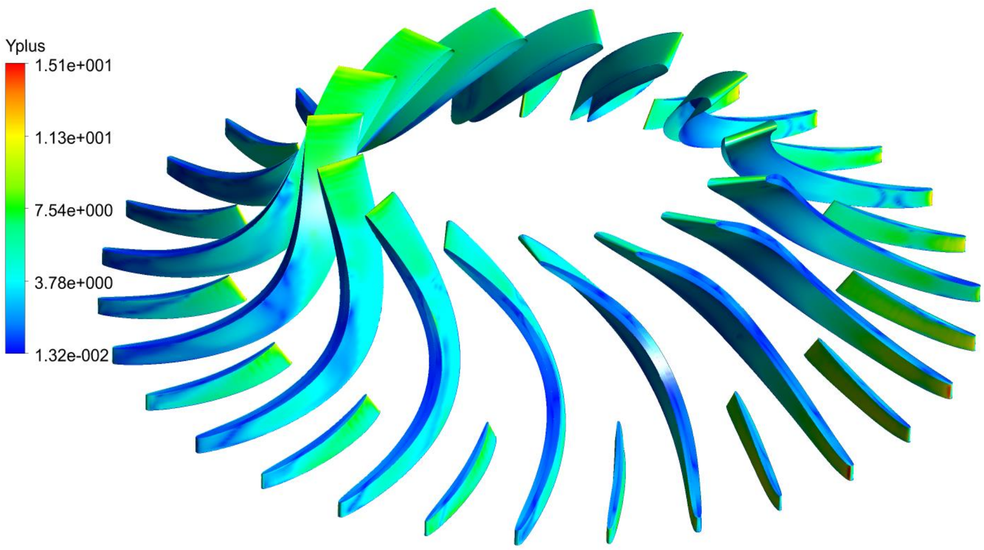

Figure 7: Distribution of dimensionless wall distance, Y+, on the surface of impeller blades, stage two

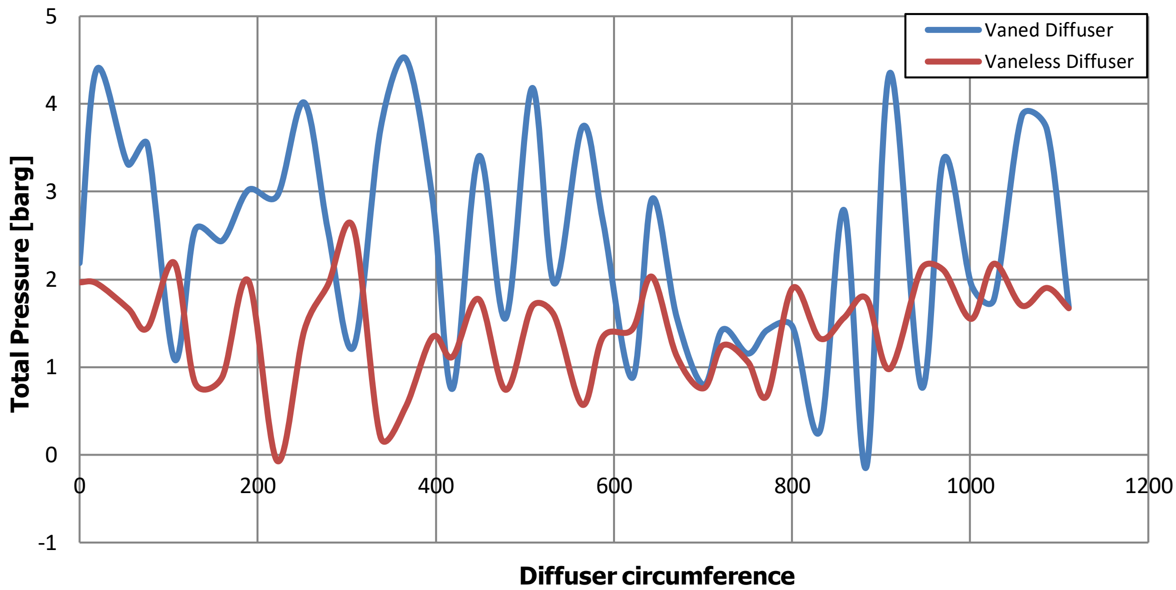

Figure 8: Average of Total Pressure on diffuser circumference

Velocity also represents an important factor in establishing compressor performances. In the case of vaneless diffuser, the velocity at compressor outlet was determined by the numerical simuloation to be of 213 m/s; with 88 m/s higher than in the case of the vaned diffuser, leading to a decrease of performances.
Polytropic efficiency for the centrifugal components is presented in Fig. 9. Analyzing each component separately, it can be seen that the efficiency for the case of the vaned diffuser is much higher than for the vaneless one. But overall, the centrifugal compressor with vaneless diffuser has a polytropic efficiency higher by 0.11 %.
The evolution of the total pressure in the impeller of the second stage is presented in Fig. 10. Impeller inlet pressure is the same for both cases, but this is changing as we approach the volute outlet. In the vaneless diffuser, a decrease of total pressure up to 4.9 bar is observed, while in the case of the vaned diffuser, the pressure reaches a value of 6.43 bar. For the vaneless diffuser, the pressure loss in the diffuser is of almost 0.6 bar, which is lower compared with the vaned one, where the pressure loss is of 1.57 bar.

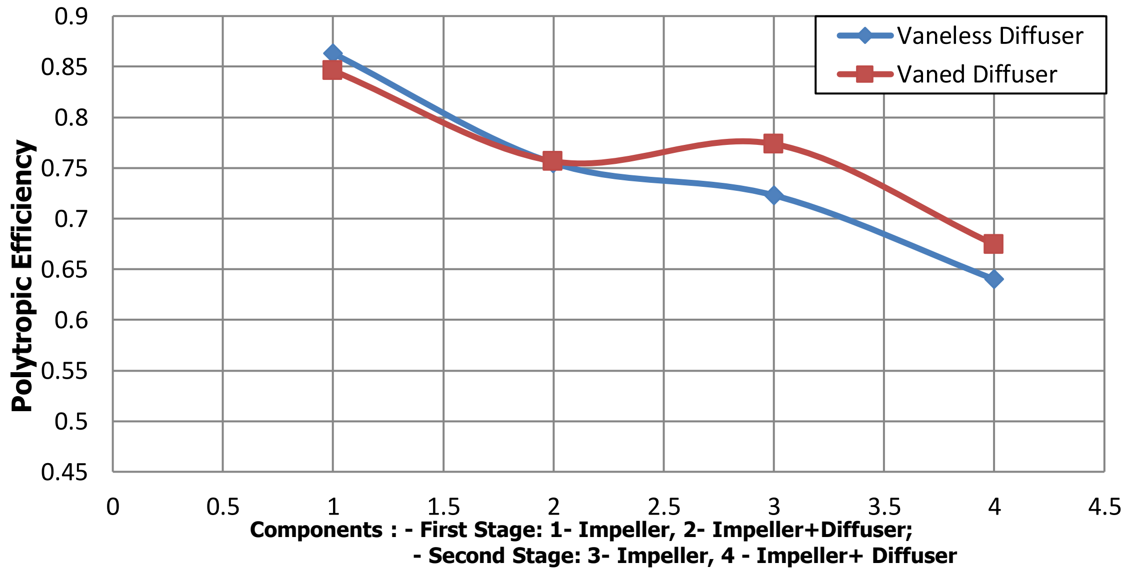

Figure 9: Polytropic efficiency for impellers and diffusers of both stages

For the studied compressor stage, the geometry of the scroll plays an important role, defining performances of the centrifugal compressor. In the case of the vaned diffuser, the pressure decreases at the exit to 4.26 bar, which is significant. For the vaneless diffuser, the pressure is decreasing to a value of 3.5 bar. It is noteworthy that in the experiments, the exit pressure was of 4.89 bar [18].

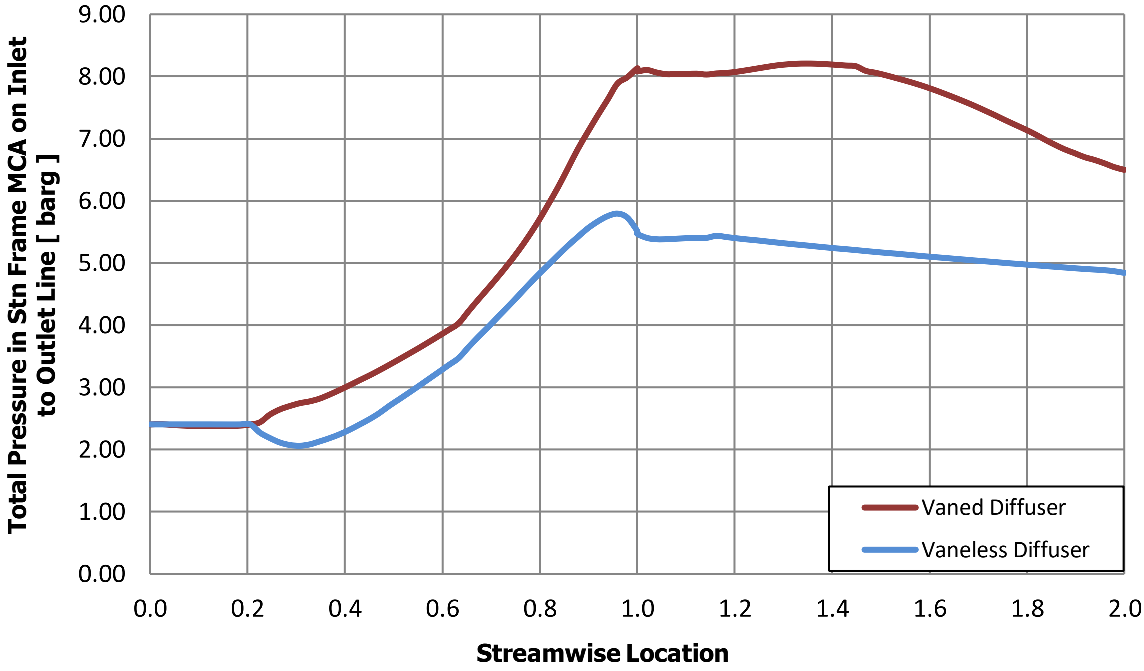

Figure 10: Total Pressure in Stn Frame for the impeller of the second stage

For streamwise locations above 1.4, the pressure drop is steeper for the vaned diffuser than for the vaneless diffuser probably due to shock waves.

4 CONCLUSION

In this paper, the influence of a vaneless diffuser on the performances of a centrifugal compressor with two stages close to surge line was studied. in order to see how the stable operating range is affected. Two cases were studied: a) both stages having a vaned diffuser, b) first stage with a vaned diffuser and the second stage with a vaneless one.
The numerical simulations carried out for the two stages of the centrifugal compressor were coupled, allowing a better comparison between the numerical and the experimental data. The vaneless diffuser not only influences the flow inside the scroll, but also the pressure ratio inside the second impeller. As proof, a difference of approximately 2 bar is observed in the simulations at the second stage stator. The agreement between the numerical and the experimental case is good, the difference in pressure between them being approximately 0.63 bar at the same location.
The scroll geometry plays an important role in establishing good performances for the compressor.
This analysis shows that using vaneless diffuser this size will affect the overall pressure ratio of the compressor in a negative way. Also, the velocity in the vaneless diffuser was seen to be too large, 213 m/s, adversely affecting compressor performances. Probably a longer diffuser, with a different volute will result in improved performances, but this has to be verified as future work.

REFERENCES

1. A.Jaatien, "Performance Improvement of Centrifugal Compressor Stage with Pinched Geometry or Vaned Diffuser", Lappeenranta University of Technology, Lappeenranta, Finland, October, 2009
2. Waleed Al-Busaidi1, Pericles Pilidis, "Techno-Economic Optimization of Diffuser Configuration Effect on Centrifugal Compressor Performance", *American Journal of Energy Research*, Vol. 3, No. 2, 37-48,2015

3. P.E. Smirnov, F.R. Menter, "Sensitization of the SST turbulence model to rotation and curvature by applying the Spalart-Shur correction term", *ASME Paper* GT 2008-50480, Berlin, Germany, 2008.
4. N.J. Georgiadis, D.A. Yoder, "Evaluation of Modified Two-Equation Turbulence Models for Jet Flow Predictions", *44th AIAA Aerospace Sciences Meeting and Exhibit*, 9 - 12 January 2006, Reno, Nevada
5. A. Jaatinen, "Performance Improvement of Centrifugal Compressor Stage with Pinched Geometry or Vaned Diffuser", Lappeenranta University of Technology, 978-952-214-820-9
6. S. Shaab, "Design optimization of a centrifugal compressor vaneless diffuser", ELSEVIER, *International Journal of Refrigeration* 60 142-154, 2015
7. Senoo Y.&Kinoshiya Y., "Limits of rotating stall and stall in vaneless diffuser of centrifugal compressors", ASME P 78 GT 19.
8. P.Frigne, R. van den Braembussche, "One Dimensional Design of Centrifugal Compressors Taking Into Account Flow Separation in the Impeller"", *Von Karman Institute for Fluid Dynamics*, Turbomachinery Department, June 1978
9. Kangsoo Im, "Development of a Design Method for Centrifugal Compressor", Dissertation, Michigan State University, Mechanical Engineering, 2012
10. David Japikse, "Centrifugal Compressor Design and Performance", Concepts ETI, Inc., ISBN 0-933283-03-2, 1996
11. Ernesto Benini, "Optimal Navier–Stokes Design of Compressor Impellers Using Evolutionary Computation ", *International Journal of Computational Fluid Dynamics*, October 2003 Vol. 17(5), pp. 357–369
12. S.Shaaban, "Experimental investigation and extended simulation of turbocharger non-adiabatic performance", PhD Thesis, University of Hanover, Germany
13. S. Ljevar, H.C.de Lange, A.A. van Steenhoven, "Rotating stall characteristics in a wide vaneless diffuser", *Turbo Expo 2005*, vol.6, Nevada, USA, pp.1335-1342, June 6-9.
14. He, N. Tourlidakis, A., "Analysis of Diffusers with Different Number of Vanes in a Centrifugal Compressor Stage", ASME TURBO EXPO, 2001-GT-0321
15. P. Delbert, G. Gyarmathy, A. Sebestyen, "Flow Phenomena in a Vaned Diffuser of a Centrifugal Stage", The American Society of Mechanical Engineers, 93-GT-53, 1993
16. Sabri Deniz, "EFFECTS OF INLET CONDITIONS ON CENTRIFUGAL DIFFUSER PERFORMANCE", Gas Turbine Laboratory, Massachusetts Institute of Technology, GTL Report #225, Cambridge, March 1997
17. P. Dalbert, M. V. Casey, E. Schurter, "Development, Testing, and Performance Prediction of Radial Compressor Stages for Multistage Industrial Compressors", *J. Turbomach* 110(3), 283-292, Jul 01, 1988
18. Gheorghe Fetea, "Theoretical and experimental research on centrifugal air turbocompressors" (Cercetări teoretice si experimentale asupra turbocompresoarelor centrifugale de aer), - to be published
19. M.Ding, C.Groth, S.Kacker, D.Roberts, "CFD Analysis of Off-design Centrifugal Compressor Operation and Performances", Thesis (M.A.Sc.), University of Toronto, 2005