

Tip clearance influence in CFD calculations and optimization of a centrifugal compressor stage through CFD methods

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Abstract. This paper presents a study on a centrifugal compressor stage aerodynamics. The influence of the blade tip clearance on performances of the centrifugal impeller and also on the diffuser is taken into consideration for the overall optimization process. Also a stress analysis has been carried out to investigate the stresses and strains of the blade. The impeller was designed for a pressure ratio of 4:1, a mass flow of 8.1 kg/s at 22 000 rpm. Three cases have been studied: first without a tip clearance, the second one with tip clearance and the third case which studies an optimized geometry for the vaned diffusers. Through a designer driven optimization process, the recirculation zone that appears in the diffuser was reduced by half. Following the numerical analysis some differences were noted between the three cases in terms of both global results and particular flow details. One of the main differences is given by the power consumed by the impeller. Due to the elimination of the tip clearance (shrouded impeller), the power consumed by the impeller was reduced by 72 kW and the efficiency increased by 2.2%. In the case of diffusers analysis, was noticed an improvement in stage efficiency of 8.2%.

1. Introduction

Compressed air is used in many applications, industrial purpose (energy storage, pneumatics, refrigeration, injection molding etc.) and also domestically one. In industry, compressed air is expensive, often being considered as the forth utility after electricity, natural gas and water [1]. Mass flow, efficiency and pressure rise are the three most important parameters used in defining performances of a compressor. Efficiency and pressure ratio depend on the impeller geometry and the flow through the work channel [2].

Losses in centrifugal compressors must be minimum, to minimize work input. This leads to changes in the compressor performances and operating range. There are many aerodynamic mechanisms responsible with the changes in performances, amongst them are the tip-leakage flow and low stream wise velocity region, which is responsible for flow blockage and decreased pressure rise [3].

One of the factors that affect centrifugal compressor aerodynamic performances is the tip clearance, especially the relative size of it [4, 5]. This influence on the fluid flow structure, operating range and performances has traditionally been studied through experimental methods. According to the study of S.M. Swamy et al [6], it was found that if the value of the clearance is increasing, the operating range starts to decrease; also static pressure rise is reduced in compressor which is the



reason for the stall-like behavior. A small tip clearance leads to a decrease of loss caused by interactions between the tip leakage flow and the main passage flow [7].

Neil et al [8] show in their study that the efficiency and total pressure ratio decrease with tip clearance, for stages with vaned and vaneless diffusers [9]. According to their results at a given tip clearance, an impeller with vaned diffuser has higher pressure ratio than a vaneless one. Vaneless designs offer higher operating range and lower cost, while vaned diffuser can be used for higher pressure rise and efficiency [10]. It is known that small geometric variations in the rotor cross section area can have substantial effects on performance; not only that a higher tip clearance reduces efficiency but it has also been shown that certain types of tip shapes improve the behavior of the centrifugal compressor near the steady operating limit, also improving the surge line [11].

Experimental and numerical studies have shown that diffusers have a significant influence on the stability limit of a compressor depending on the impeller design and the matching between the impeller and diffuser [12]. Currently, Computational Fluid Dynamics is used to resolve and analyze problems regarding complex engineering problems regarding turbomachinery fluid flow. Development of this tool has led to a significant progress in the design of centrifugal compressors through mitigation of undesirable flow structures while increasing the effectiveness of desired ones.

For a centrifugal compressor, the impeller is arguably the most important component. Structural stresses induced in the impeller, affect the performances of the compressor, limiting the operation speed and range. Due to unsteady operating load, the blade can break up and suffer high –cycle fatigue. To calculate deflection, vibration, stress and strength finite element analysis is used for computing the Eigen modes and their values for each individual rotating component. In this paper we will focus on the improvement of a centrifugal compressor stage.

2. Numerical methods

The geometry of the compressor was created using commercial software ANSYS VISTA CCD and verified with in-house codes. The optimization of the diffusers has been achieved using a designer driven methodology, seeking to reduce recirculation vortices. Also, CFD programs allowed us to evaluate and improve the solutions regarding the shape of the blades, angles of attack on trailing and leading edge etc. Table 2 presents the main parameters obtained for the centrifugal compressor stage.

Table 1. Main parameters of the centrifugal impeller

	Impeller	First Diffuser	Optimized diffuser	Second Diffuser	Optimized diffuser
Gas	Aer	Aer		Aer	
Rotation speed [rot/min]	22 000	-		-	
Mass flow [kg/s]	8.1	8.1		8.1	
Pressure ratio	≈ 4				
Number of blades	15	12	19	14	20
Number of splitter blades	15			-	
Hub diameter at blade leading edge [mm]	90.4	452.48	452.48	758.019	758.019
Shroud diameter at blade leading edge [mm]	258	452.48	452.48	758.019	758.019
Diameter at blade trailing edge [mm]	400	728.44	631.24	279.8	335.97
Blade height at outlet [mm]	25.2	27	27	33.7	32.262
Inlet Total pressure [atm]	1	1	1	1	1
Inlet Total temperature [K]	288.16	288.16	288.16	288.16	288.16

For the cases studied in this paper, the flow is assumed compressible and the governing equations of the flow are written in Reynolds averaged form, time and mass averaged [13]:

Figure 1 presents the computational domain of the centrifugal compressor, with the two geometries, for baseline geomerty and the optimised diffusers.

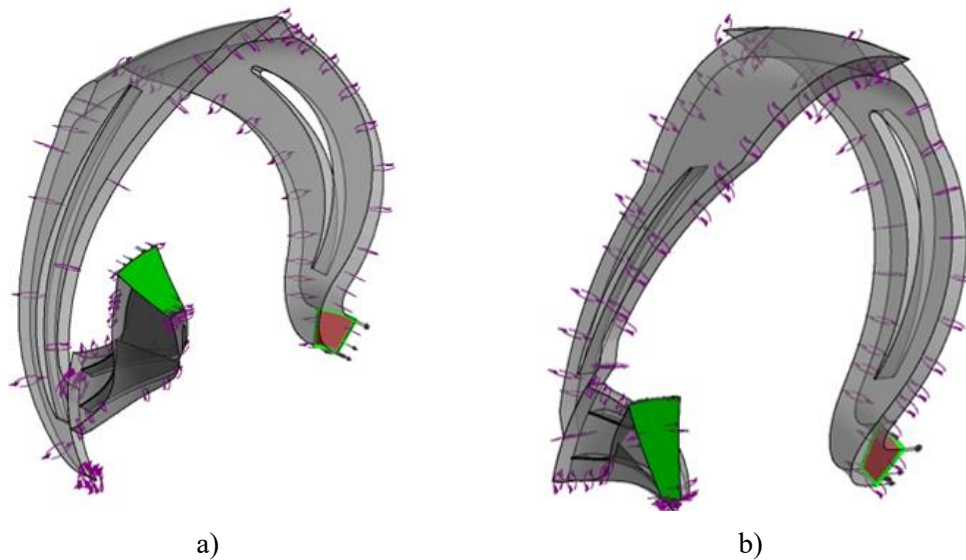


Figure 1. Computational domain, baseline geometry (a) and optimized stage (b).

The grid is structured and was realised using ANSYS ICEM CFD. Grid resolution took into account that y^+ at walls will be around 1. Walls are considered adiabatic and wall speed is zero. To improve the calculation time, a single channel for each domain was considered, thus using the periodicity function. For spatial meshing, a second-order scheme was used. Mesh statistics, for the compressor components are presented in table 2.

Table 2. Mesh statistics.

First stage components	Number of nodes
Impeller	4.1 millions
Vaned diffuser 1	0.87 milions
Vaned diffuser 2	0.32 milions

As turbulence model - SST $k-\omega$ (Shear Stress Transport) model was considered. With this turbulence model it is possible to obtain precise results, in terms of flow separation while maintaining the robustness of the k -epsilon model in the far field [14].

3. Results

In this numerical study, three cases have been considered: one with no tip clearance (shrouded impeller), second one with a tip clearance (unshrouded impeller) and the third one in which was used a new geometry for vaned diffusers, an optimized one. For studying the tip clearance, Mach number distribution at different position on the blade height was analysed. Figure 2 shows a difference at the blade leading edge for the case with no tip clearance. The difference is approximately 0.2 (figure 2b).

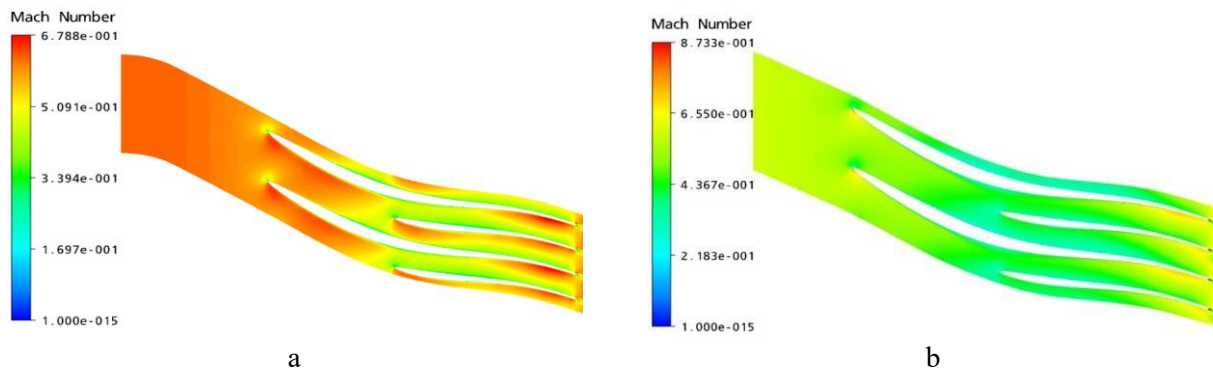


Figure 2. Relative Mach number, at 10% from blade height, measured from the hub: with tip clearance (a); without tip clearance (b).

Same difference is kept for 0.5 span, but appears another difference on the splitter leading edge where an expansion zone occurs, if the tip clearance is considered, figure 3a.

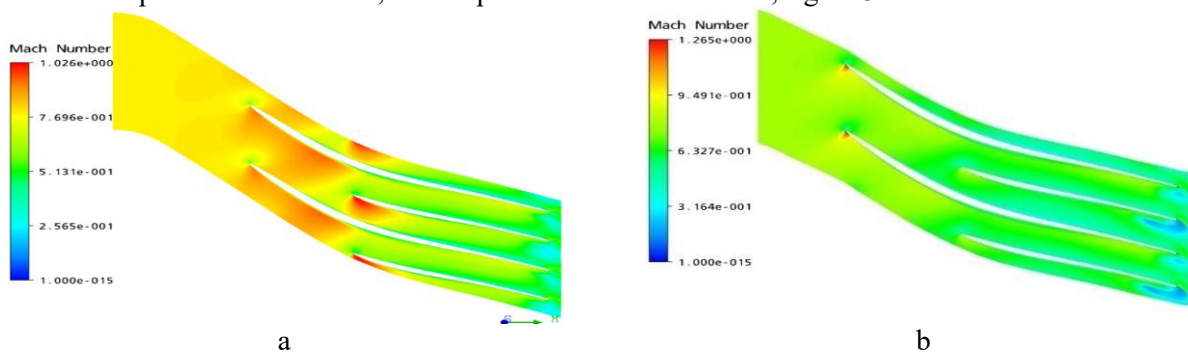


Figure 3. Relative Mach number, at 50% from blade height, measured from the hub: with tip clearance (a); without tip clearance (b).

For a blade tip, figure 4, fluid flow is different for the two cases. The difference given by the Mach number is maintained, but also other macroscopic phenomena occur, primarily the tip leakage. If in the case with tip clearance, acceleration due to streamline curvature is observed (figure 4a); at no tip clearance this is no longer visible. An explanation is that the shroud entrainment is influential in the inducer region, diminishing the relative Mach number at the compressor inlet. Near the blade tip, shock waves position changes; these shock waves forms in the front of the splitter blade (figure 4). Also, for no tip clearance the shock wave is formed also on the splitter (figure 4b). In order to be able to better observe the irregularities of the fluid flow on the trailing edge, streamlines at a distance of 90% measured from the hub are illustrated in figure 5. A pronounced recirculation area can be observed, near the blade trailing edge.

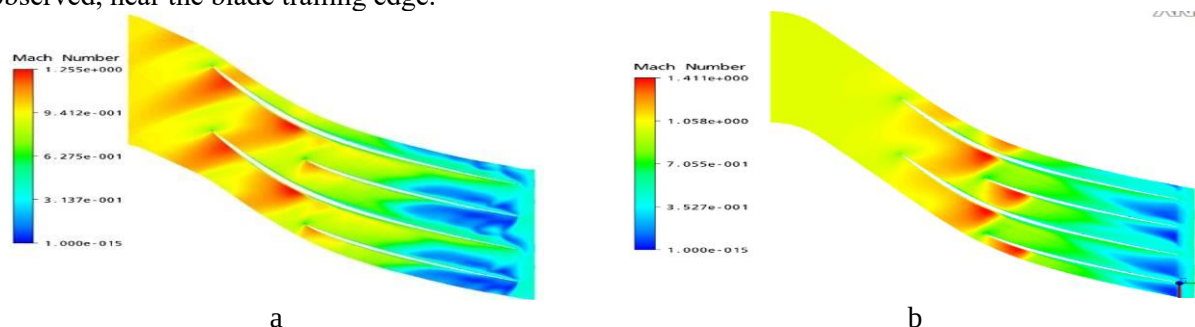


Figure 4. Streamlines at 90% from blade height, measured from the hub: with tip clearance (a); without tip clearance (b).

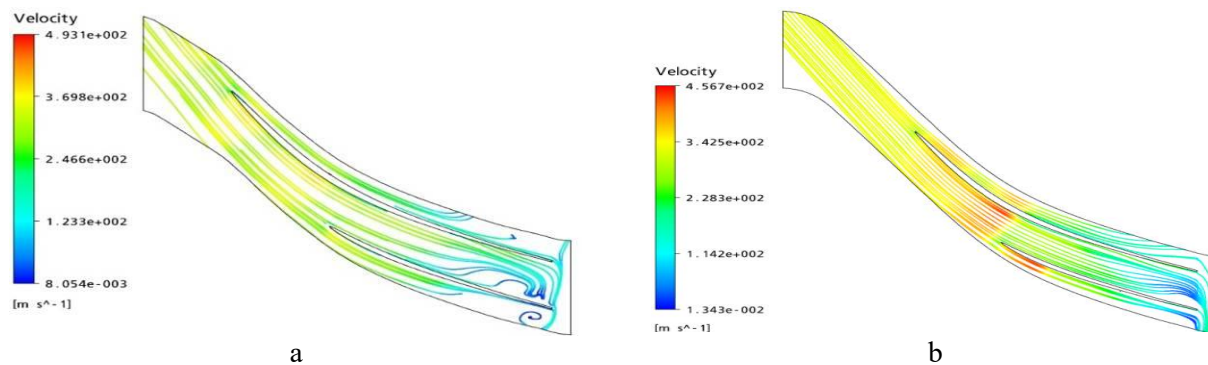


Figure 5. Streamlines at 90% from blade height, measured from the hub: tip clearance (a); without tip clearance (b).

Another difference between the two cases it can be seen in efficiency. Eliminating tip clearance, also eliminate the losses caused by it. Efficiency obtained for the case without clearance is 90.8%, which is with 2.2% higher than in the case of tip clearance. In the case of the power consumed by the impeller, a higher power is obtained in the case for the impeller with a tip clearance, 1454 kW, compared with no tip clearance, 1382 kW.

Diffuser optimization was done using its own methods and the experience gained in the field. In figure 6a it can be seen the formation of a vortex although the entire length of the blade. On the optimized diffuser it was possible the reduction of the recirculation area by half, figure 6b.

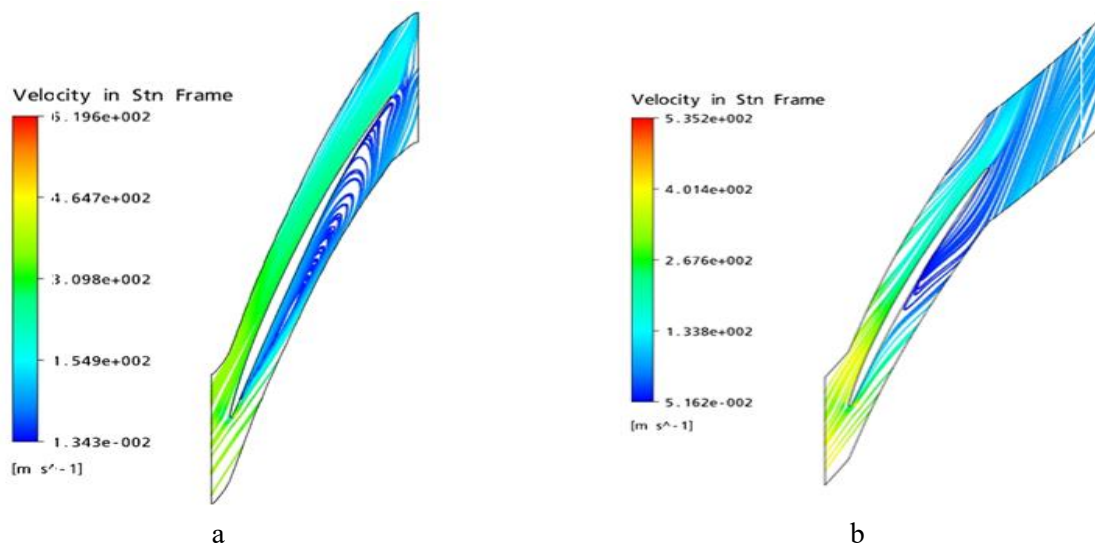


Figure 6. Streamlines at 10% from blade height, measured from the hub: basic stage (a); optimized stage (b).

Halfway distance between hub and shroud, recirculation area that forms on the case of optimized diffuser (figure 7b) was eliminated in the case. Once eliminated the recirculation area, flow angle on the stator outlet is improved.

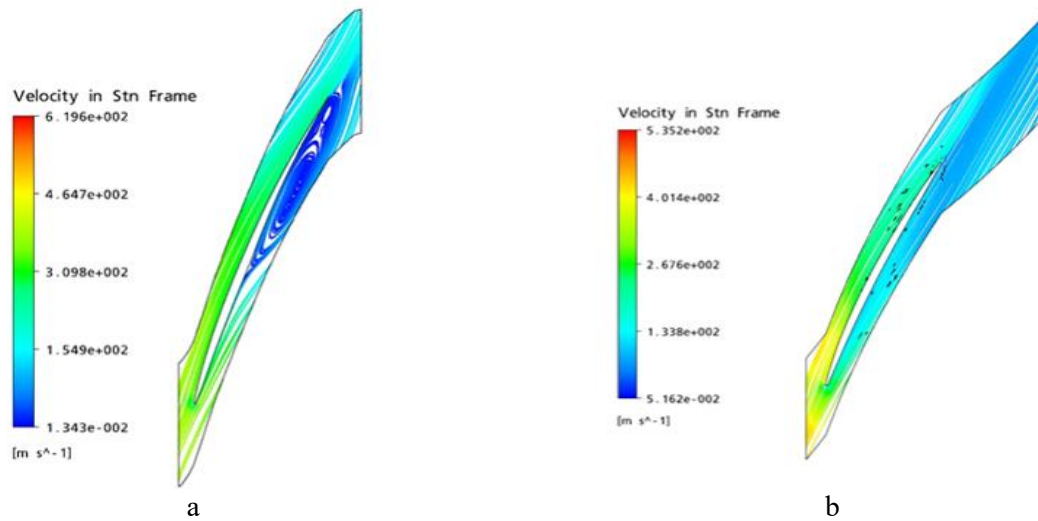


Figure 7. Streamlines at 50% from blade height, measured from the hub: basic stage (a); optimized stage (b).

In figure 8 is presented the same case, but at a distance of 90% from the blade height. The recirculation area is kept for the base diffuser. If we analyzed figures 6-8 we can conclude that this vortex is present on the whole blade height.

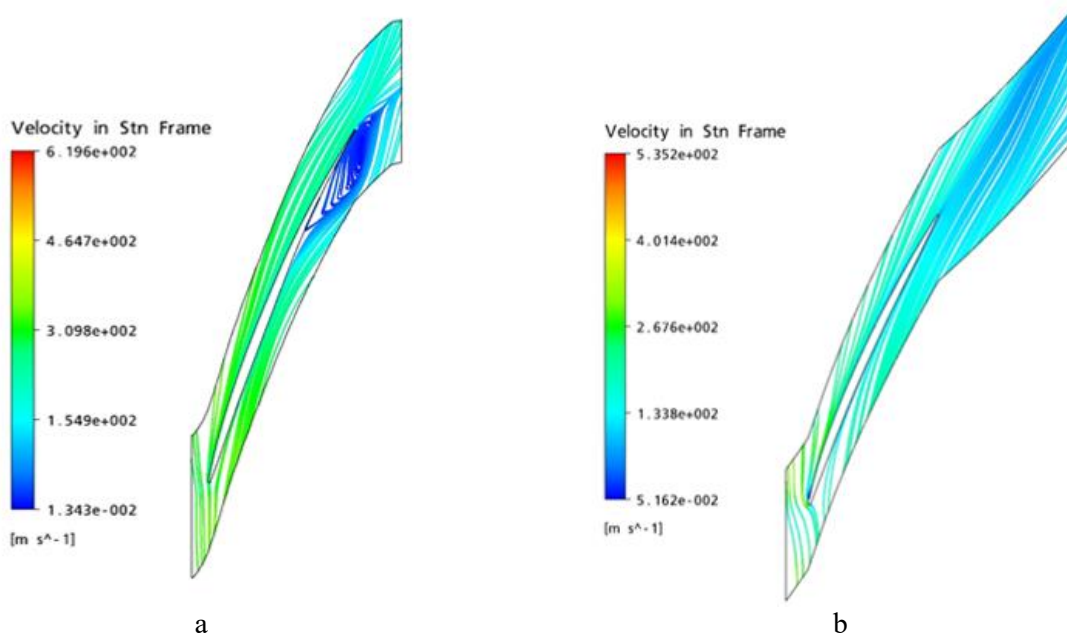


Figure 8. Streamlines at 90% from blade height, measured from the hub: basic stage (a); optimized stage (b)

In the case of the second diffuser, the recirculation zone is formed on the suction side of the blade, almost on the entire blade length, figures 9a -10a.

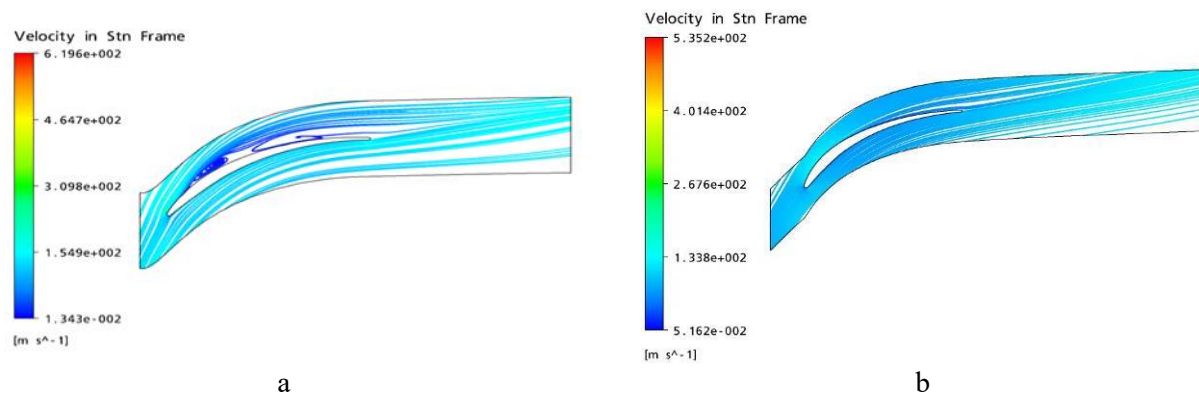


Figure 9. Streamlines at 50% from blade height, measured from the hub, for the second diffuser of the stage: basic stage (a) and optimized stage (b).

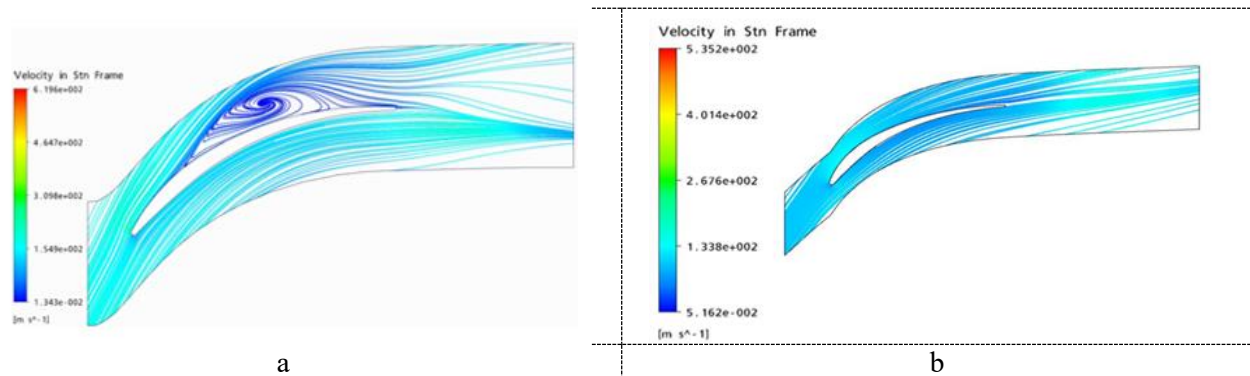


Figure 10. Streamlines at 90% from blade height, measured from the hub, for the second diffuser of the stage: basic stage (a); optimized stage (b).

Figure 11 illustrates the distribution of the total pressure in meridional view, on the entire stage of centrifugal compressor. The pressure presents some irregularities at the impeller outlet, for the baseline diffuser. For the optimized diffusers, the fluid flow is uniform in the entire stator blade system.

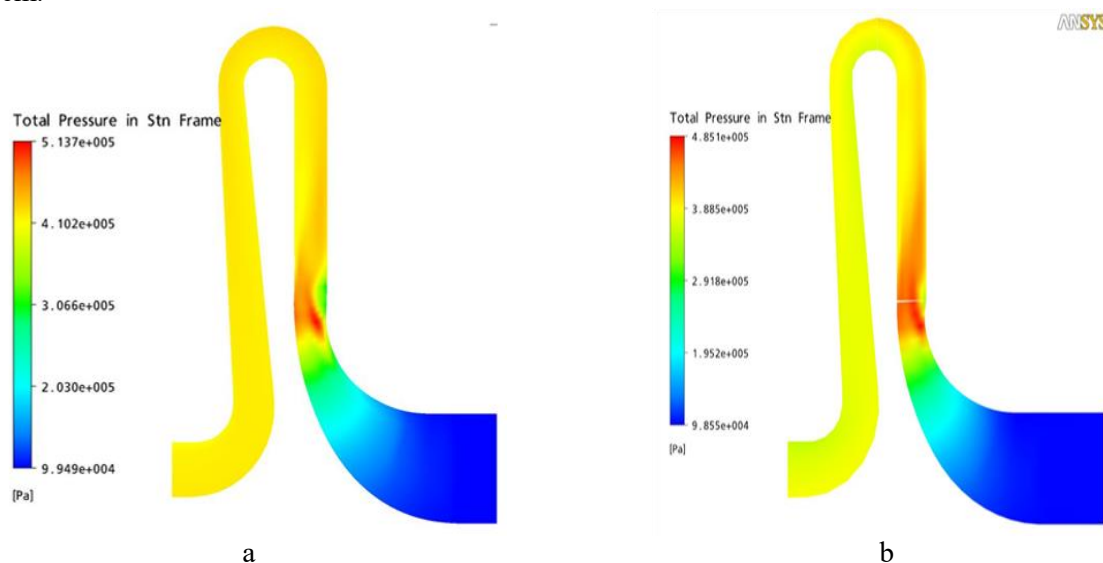


Figure 11. Distribution of averaged total pressure, in the stage channel: basic stage (a); optimized stage (b).

Pressure recovery obtained due to the optimization of the diffusers is presented in figure 12.

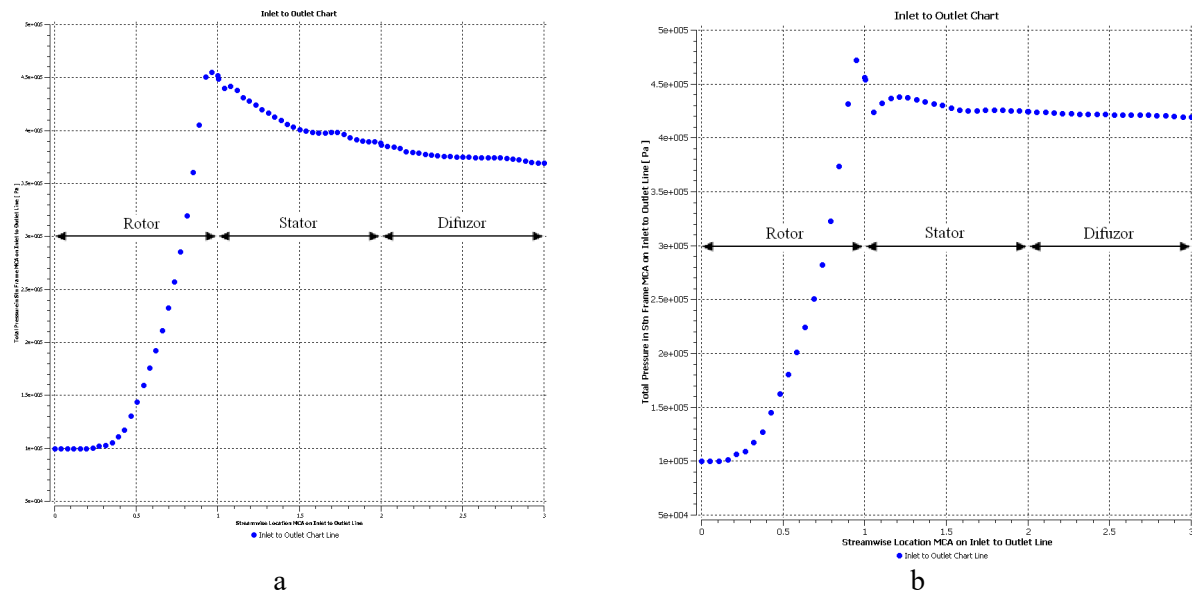


Figure 12. Total pressure distribution from the inlet in the stage to outlet: basic stage (a); optimized stage (b).

Table 3, presents the results obtained for the three cases studied.

Table 3. Geometry description and the numerical model.

	Imposed by the design conditions	Calculated with base geometry	Calculated with optimized geometry
Mass flow at nominal regime [kg/s]	8.1	8.1	8.1
Pressure ratio at nominal regime	4.1	3.696	4.198
Inlet Total pressure [bar]	1.009	1.009	1.009
Total Pressure after impeller [bar]	4.4226	4.52	4.5497
Total Pressure after first diffuser stage [bar]	4.166	3.873	4.248
Total Pressure after second diffuser stage [bar]	4.1369	3.696	4.198
Mach Number after impeller	1.228	0.957	0.94
Mach Number after first diffuser stage	0.206	0.346	0.207
Mach Number after second diffuser stage	0.09	0.342	0.280
Adiabatic efficiency impeller/stage	0.81	0.886-0.766	0.905-0.848
Total temperature at compressor inlet	15	15	15
Total temperature at compressor outlet	191	185	187
Power [kW]	1435	1454	1411

Further, results obtained for the stress analysis are presented.

Misses stress distribution caused by centrifugal inertial is presented in figure 13. The maximum Von Misses stresses is 650 MPa and the yield limit of the material is 1000 MPa. The safety factor is 1.53 ($1000/650=1.538$).

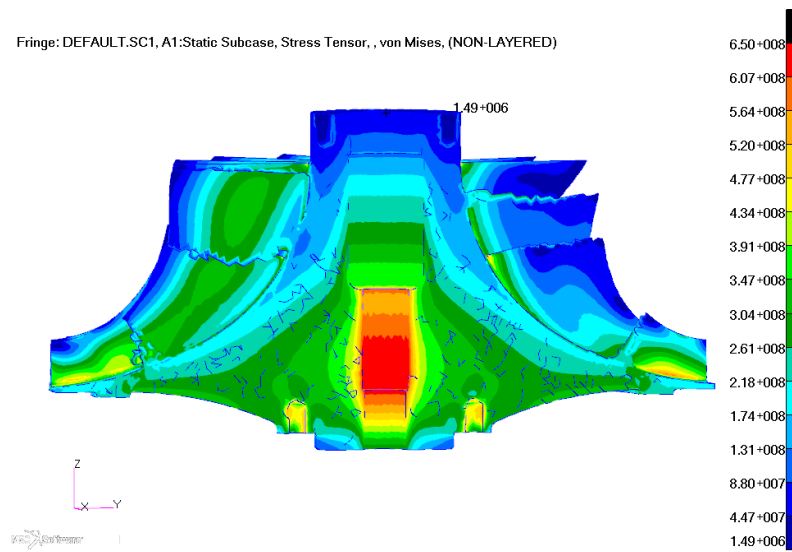


Figure 13. Von Mises stress [Pa].

Figure 14 illustrates the displacements on OZ direction.

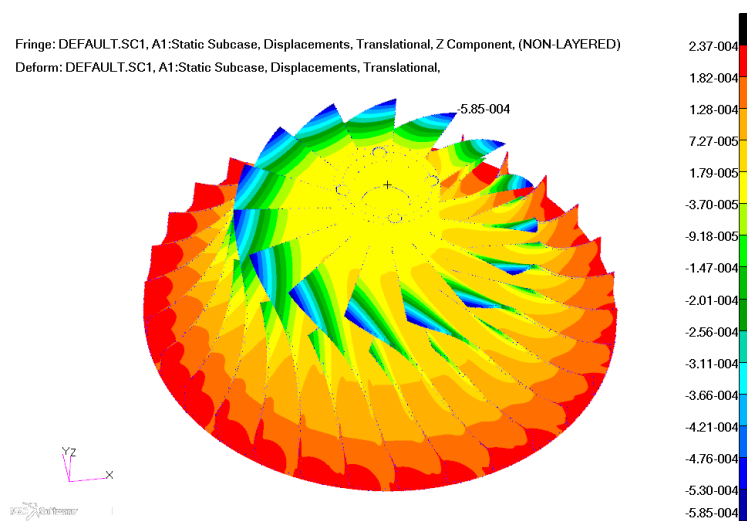


Figure 14. Displacement on OZ direction [m].

In order to design the shroud profile, the displacements of the nodes placed on the tip blade profile will be analyzed. The node used for displacements analysis is presented in figure 15.

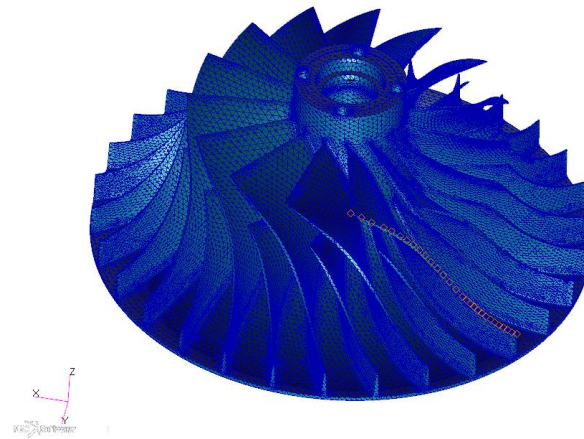


Figure 15. Displacement on OZ direction [m].

The deformed tip profile is used to generate the deformed shroud. Figure 16 illustrates the deformed rotor and the deformed shroud.

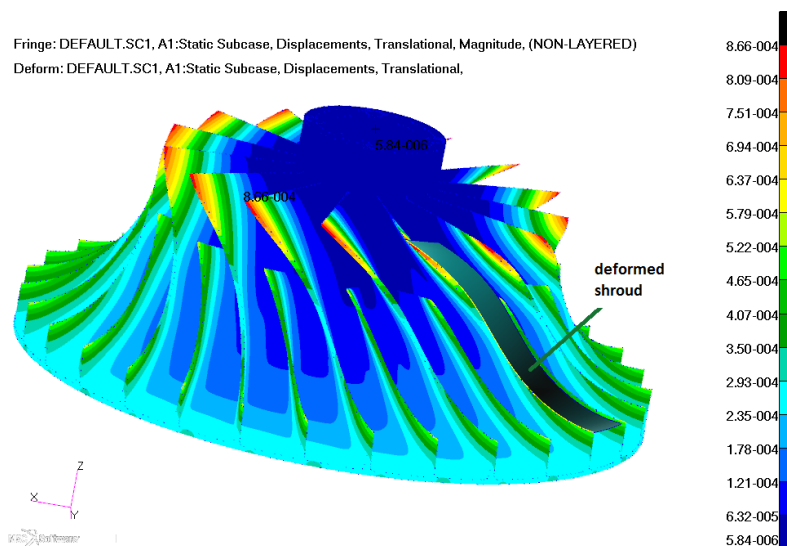


Figure 16. Displacement on OZ direction [m].

For a certain clearance the shroud will be design offsetting the deformed shroud with a normal distance equal with the clearance.

4. Conclusions

In this paper, an analysis of the influence of the blade tip clearance was made. For a shrouded impeller (no tip clearance) differences in compressor performances were obtained. A first difference is given by the increase of the Mach number with 0.2. Another difference could be seen in the power consumed by the impeller, and due to the elimination of the tip clearance the power consumed by the impeller was reduced by 72 kW and the efficiency increased by 0.022. In the case of diffusers analysis, an improvement in stage efficiency of 0.082 is obtained and the power consumption was reduced by 43 kW.

From the linear Finite Element Analysis results a maximum von Misses stress of 650 MPa. Regarding the factor of safety, this is 1.53.

It can be concluded, that for a shrouded impeller and an optimized diffuser can increase the efficiency of the centrifugal compressor. Also will lead to a decrease of the power consumed.

5. References

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