Losses and blade tip clearance for a centrifugal compressor

Oana DUMITRESCU*[,1], Ramona STANCIUC[1], George Bogdan GHERMAN[1],
Valeriu DRAGAN[2]

*Corresponding author
[1]COMOTI – National Research and Development Institute for Gas Turbines,
B-dul Iuliu Maniu 220D, Bucharest 061136, Romania,
oana.dumitrescu@comoti.ro*, ramona.stanciuc@comoti.ro,
bogdan.gherman@comoti.ro
[2]KTH Royal Institute of Technology,
Kungl Tekniska Högskolan, SE-100 44 Stockholm,
valeriu@kth.se

DOI: 10.13111/2066-8201.2018.10.2.4

Received: 15 November 2017/ Accepted: 17 January 2018/ Published: June 2018

***Aerospace Europe CEAS 2017 Conference**,*
16th-20th October 2017, Palace of the Parliament, Bucharest, Romania
Technical session & Workshop Aerothermodynamics & Thermal Science

Abstract: *The present paper presents the numerical analysis for a transonic centrifugal compressor using steady state CFD. The blade tip clearance effect over the position of shock waves, tip losses and the performances of the impeller are studied. Numerical simulations have been performed using RANS modelling, with the k-omega SST turbulence model (Shear Stress Transport). Eight cases were taken into consideration for the impeller with the following blade tip clearances values: 0 mm, 0.1 mm, 0.3 mm, 0.4 mm, 0.5mm, 0.7 mm, 1 mm, 2 mm, at the same operating conditions. For the entire stage only seven cases were studied, without the value for 0.1 mm because of its abnormal behaviour, as can be seen in the case of the impeller simulations. Results showed that the position of the shock wave does not change with the increase of the tip clearance. Aerodynamic losses due to shock wave, secondary flow and turbulence can be seen in the polytropic efficiency of the centrifugal impeller and the difference between the two extreme cases is about 3.2 %.*

Key Words: *CFD, centrifugal compressor, tip clearance, losses*

1. INTRODUCTION

In recent years, research started focusing on high capacity centrifugal compressors with high pressure ratio for turbo shaft engines that are used for combined heat and power applications [13]. The trend in developing these new turbo shaft engines is to increase specific power with decreasing weight and volume. Another highly studied issue in recent years regarding the centrifugal compressor impeller is the influence of the tip clearance on the performance of the compressor [14]. It has been noticed that aerodynamics of the tip clearance plays an important role in the development of a centrifugal compressor and will have an important significance for the new engine models. This is particularly important for off-design point operation, where any additional loss to the stage may result in unstable operation and hence limit its range. The greatest problem with tip clearance is the secondary flow between the casting and the tip of the blade due to the pressure differences at the blade tip [1]. The boundary layer separation on the blade leads to greater tip leakage, therefore optimization of both blade angles and end wall geometry is required [2], [3]. For centrifugal impellers with a subsonic flow reduction, tip clearance positively influences the performance of the compressor. Numerical studies have been made for the NASA centrifugal compressor (LSCC), with a vanelles diffuser, by Gao et Al. [4] regarding the influence of the blade tip clearance. Their calculations were performed in several operating conditions, with four different tip clearances (0%, 50%, 100% and 200% of the tip clearance design size). The numerical results show that the secondary flow area from the trailing edge of the blade is considerably influenced by the tip clearance. A volume of reduced kinetic energy fluid accumulates near the casing of the unshrouded impeller.

Ishida, Ueki and Senoo [5] also made a synthesis of the secondary flow at the tip clearance. According to the theory presented by them, the losses caused by the tip clearance in the case of centrifugal rotors consist mainly in two categories:

- First is related to the frictions occurring in the flow field at the tip of the blades:
- Second is caused by the loss of pressure to support the fluid in the space between the casing and the tip of the blades, against the gradient of pressure in the meridional plane (in the vaneless space between the rotor blades and the casing).

Another study regarding the influence of the tip gap, showed that performances of a stage can be correlated with the gap size while rotor performances appear to be subjected to more complex design [6].

In all cases, for a highly charged transonic centrifugal compressor, the presence of shock waves affects the secondary flow rate at the tip [7] and also because of the potentially unstable effect of a blade diffuser on the rotor flow losses. Shum [8] has shown that there is an optimal radial clearance that will provide optimum compression.

In ref [9] Large-eddy simulations were carried out to understand the mechanism for low pressure fluctuations downstream of the tip-gap. They revealed that tip-leakage vortex and tip separation vortices are influenced by blade wake and moving wall.

I. Sashina et al. [10] have developed a parameterization method for altering the tip geometry of a blade impeller. They have tested different blade tips for improving the performances of a high speed centrifugal compressor. Their study showed that using a winglet tip improves the pressure ratio and the grooved tip leads to a decrease of the pressure ratio because of the leakage flow at the tip which is increasing [10]. Using a Gurney flap on the tip of the blades was tested with success by CFD methods [11], increasing pressure ratio and efficiency at design point. Following these ideas in this paper is studied the possibility of finding an optimum where the effect and the intensity of the shock wave, the mass flow rate at the tip and the tip clearance will be minimal for the performances of the compressor.

2. SPATIAL DISCRETISATION

The present paper analyses a centrifugal compressor from an industrial application turbine. The present geometry has an impeller with 15 blades and two diffusers with 19 blades each. Numerical simulation were carried out using the ANSYS CFX solver. All numerical simulations have been performed using a RANS model that compensates for the rotation and curvature of streamlines as well as the transport of shear stresses.

The computational domain used is presented in Fig. 1, depicting the impeller and both diffusers. A structured mesh was created with near wall refinement, to resolve these region efficiently and capture the phenomena that appear in the boundary layer flow. For this case

Menter turbulence model SST (Shear Stress Transport) [12] was used. The model combines two turbulence models k-omega and k- epsilon. It uses k – omega in the region from the interior of the domain and k – epsilon in free shear flow.

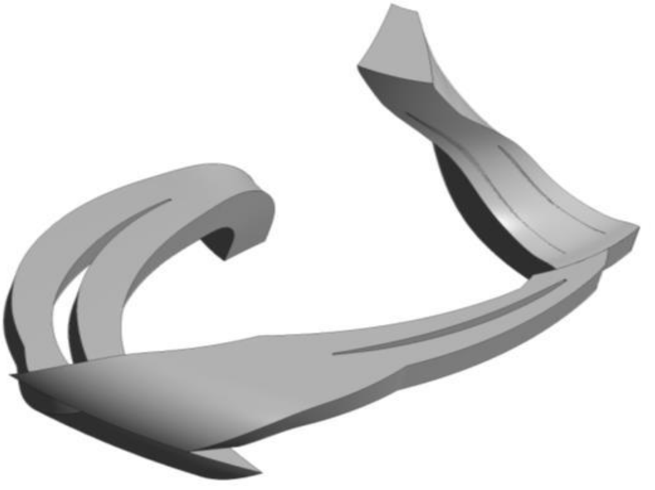


Fig. 1 – Computational domain: impeller, diffuser

3. RESULTS

In this paper a centrifugal impeller with a medium pressure ratio of about 4:1 is considered. It has 15 blades and the flow rate is 8.1 kg / s. Due to this pressure ratio and inherent high relative Mach number, shock waves form between the blades. In order to estimate the effect of the tip clearance on the impeller, the calculation of the steady state using the turbulent k-ω SST model is used to simulate the nominal point. Different values were tested for tip clearance, keeping conditions at constant limits. All simulations were performed at 8.1 kg/s and 22000 rpm. The value of the non-dimensional distance to the wall, y+ was maintained under one unit to take advantage of the capabilities of the SST model.

Before analyzing the influence of the tip clearance on the entire stage, the impeller alone was studied in order to provide guidelines and reveal possible bottle-necks in this study.

On the impeller a shock wave is formed between two blades - it is obvious that the position of the shock wave does not change with the tip clearance.

The mass flow and pressure loss at tip, which increases with tip clearance, does not affect the intensity of the shock wave. As we move away from the blade, the intensity of the shock wave decreases showing that the secondary flow rate at the tip clearance has a major influence on it.

In order to estimate the effect of curvature of the streamlines on the impeller, it is seen in Fig. 3 a recirculation zone near the end of blade section which increases as it approaches the end of the section. Also this area increases with the rise of the tip clearance.

Analyzing the flow through the tip clearance, Fig. 4, it is possible to note that the mass flow from the secondary flow increases with the tip clearance, affecting the angle of flow at the exit of the impeller, approximately 10 degrees difference between the clearance of 0.1mm and 2mm.

Also, close to the impeller exit, the vortex that is formed due to the secondary flow through the tip clearance and the boundary layer separation are different depending on the tip clearance, as can be seen in Fig. 4.

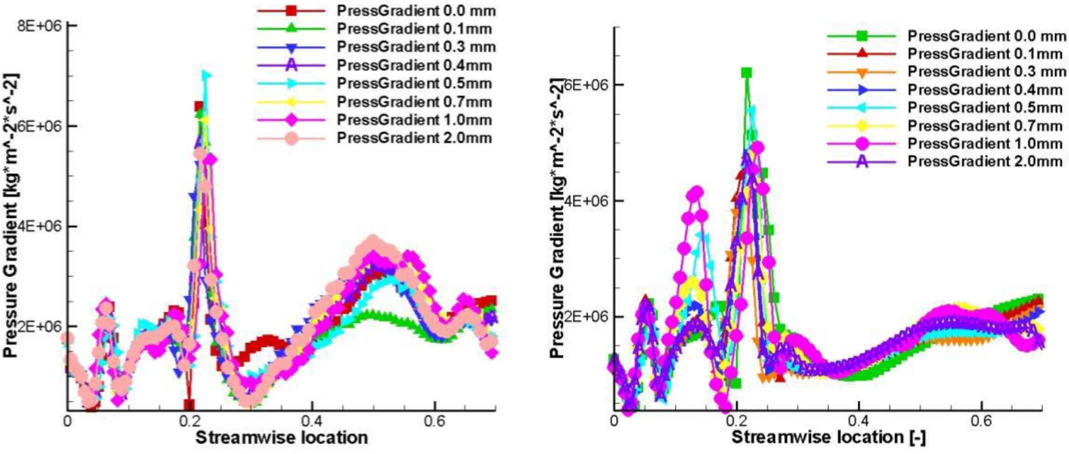


Fig. 2 – Pressure gradient depending on the longitudinal position, line of 11% (left) and 27% (right) for the main blade

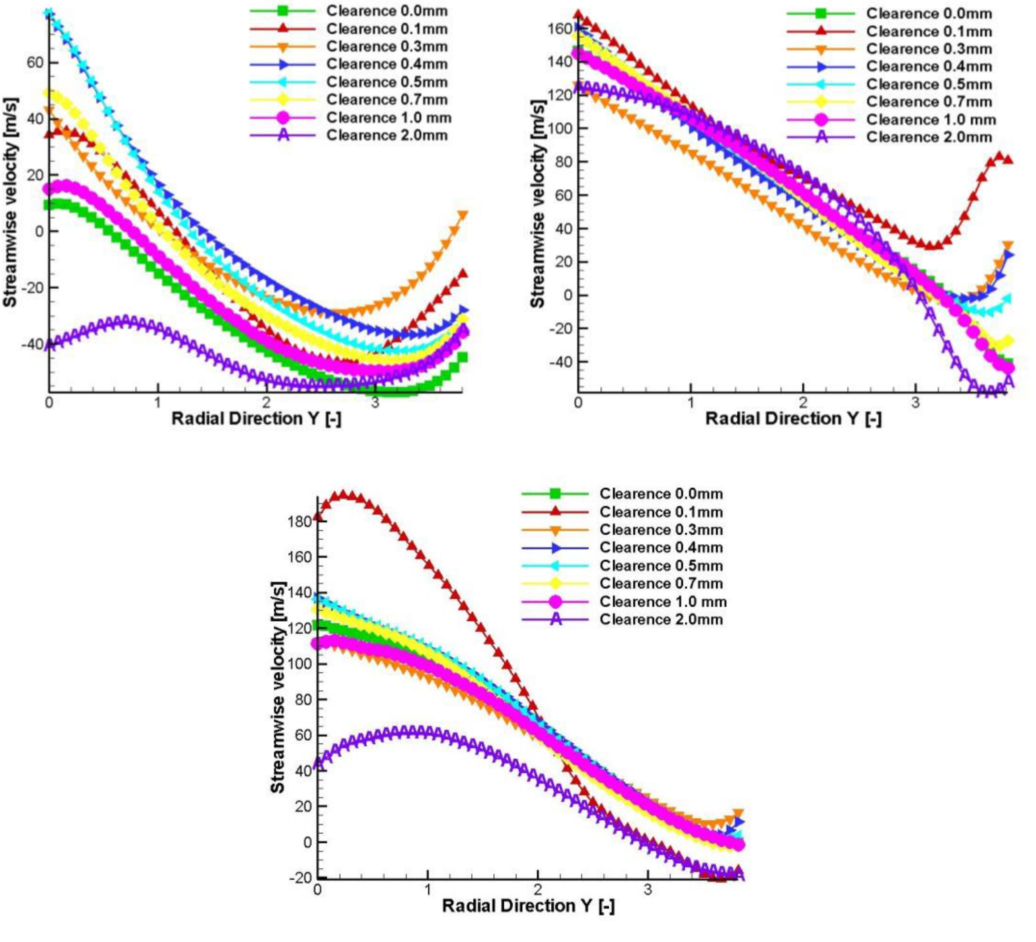


Fig. 3 – Velocity distribution at various radial positions 0% left, 10% center, and 20% right of the main blade, on the blade trailing edge, at 95% of the blade height

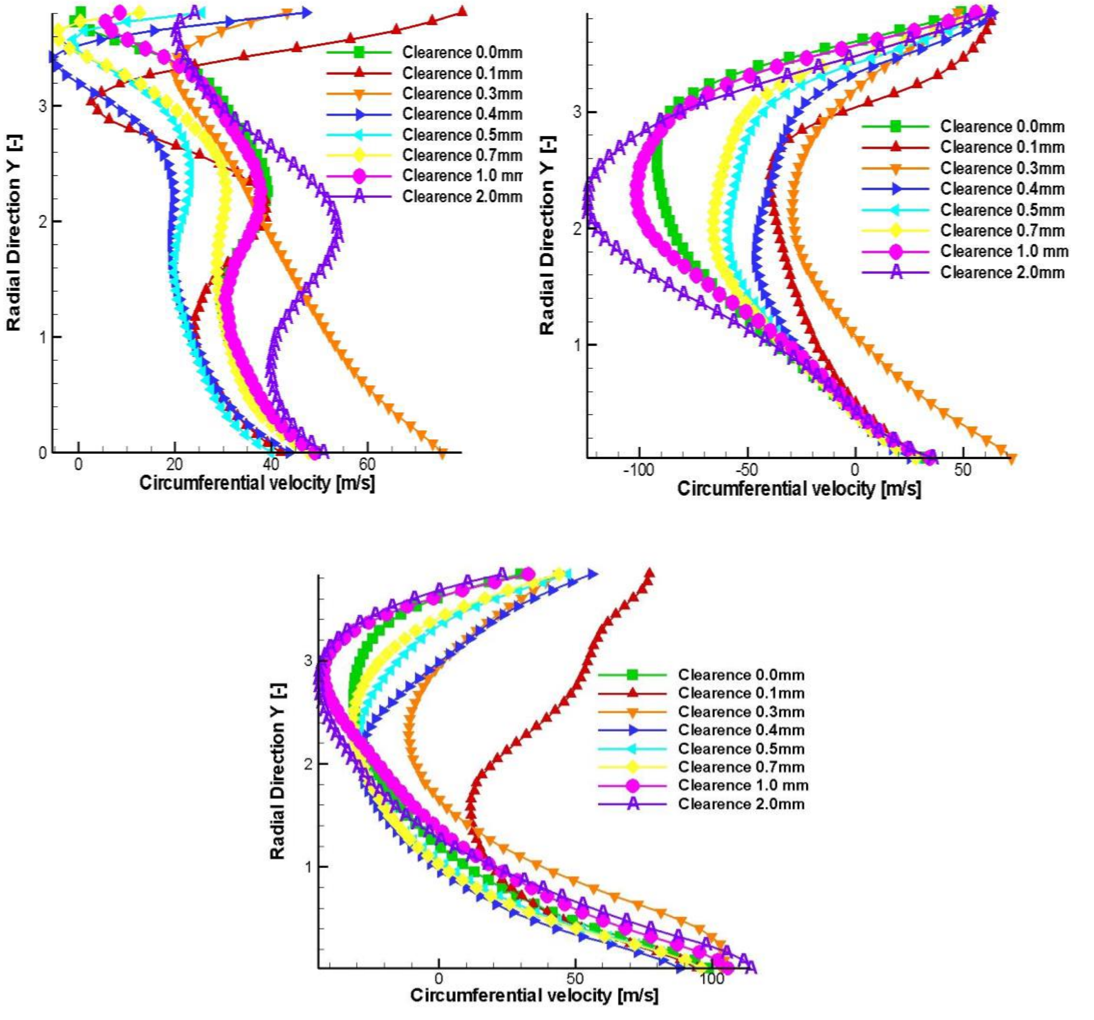


Fig. 4 – Velocity profile in circumferential direction at various radial locations 0% left, 10% center, and 20% right of the main pallet, on the blade trailing edge, at 75% of the blade height

Following the numerical analysis, it was notice that change of the tip clearance has a major impact on the flow between impeller and first diffuser, thus the results that follow will be focused on this space.

Figure 5 presents the evolution of the velocity on the entire compressor stage. From this figure, the regions affected by the value of the tip clearance are between impeller and first diffuser and the exit from the deswirler (the second diffuser). In the case of the impeller-diffuser interaction, a vortex is formed caused by the same secondary flow. Near the shroud of the impeller a high velocity area is formed, its magnitude is decreasing with the increase of the tip gap.

Figure 6 depicts the distribution of static pressure on this area. Increasing value of tip clearance leads to a decreasing of static pressure. At the interaction between impeller – diffuser, the boundary layer detachment from the impeller blades causes significant secondary flow resulting in the change in flow angle.

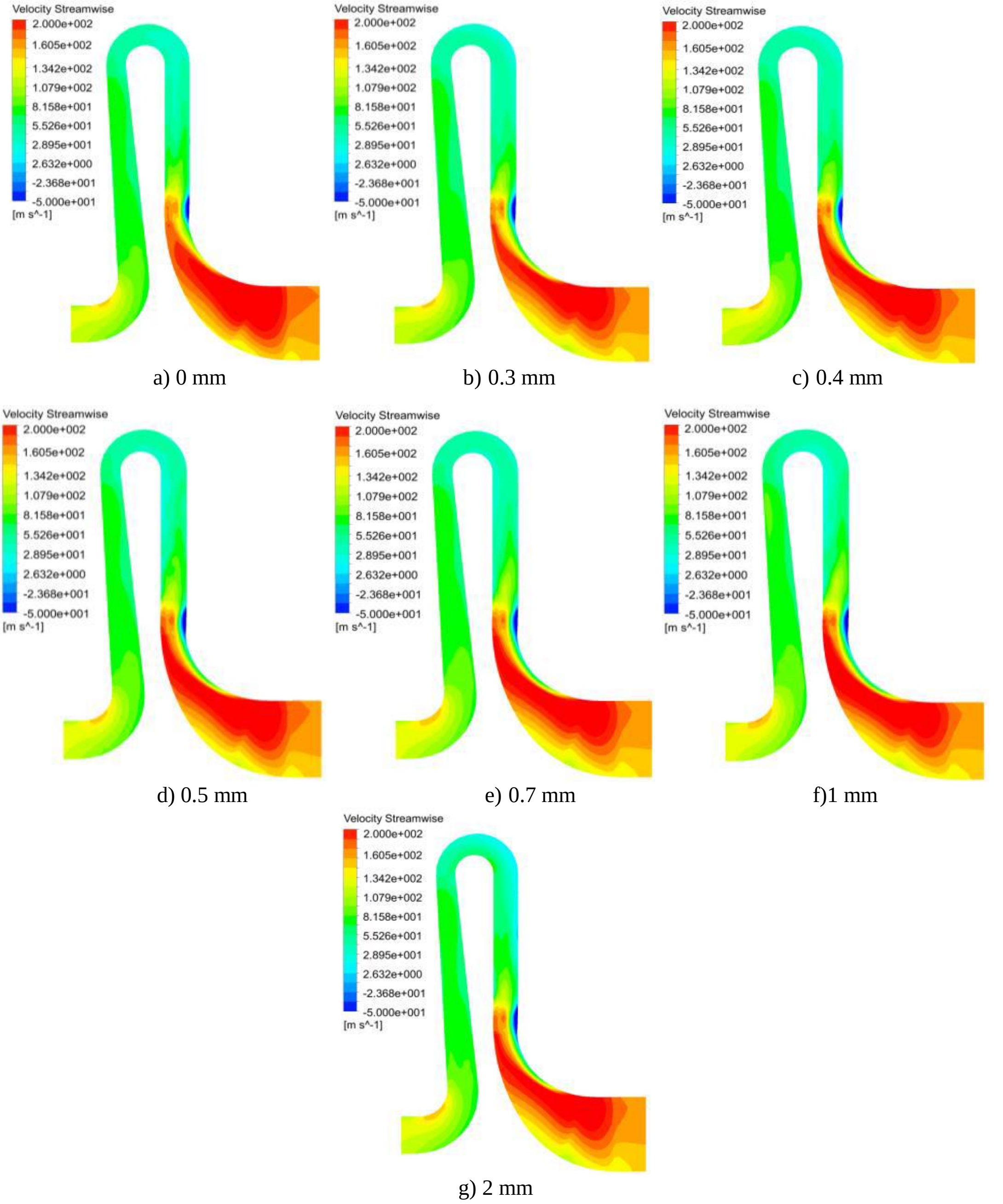
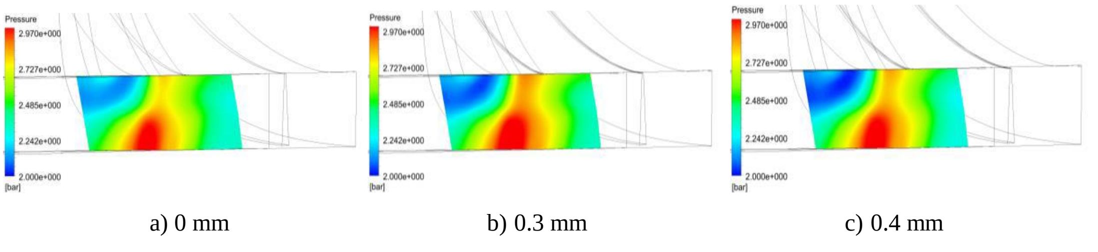
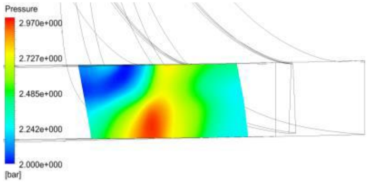
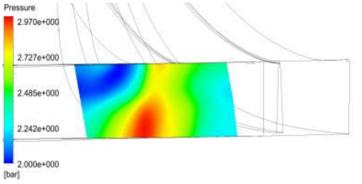


Fig. 5 – Velocity Streamwise for different tip clearances: a) 0 mm, b) 0.3 mm, c)0.4 mm, d) 0.5 mm, e) 0.7 mm, f) 1mm, g) 2 mm

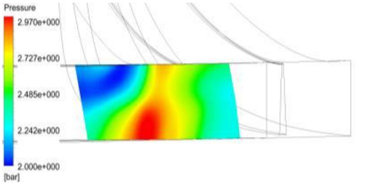




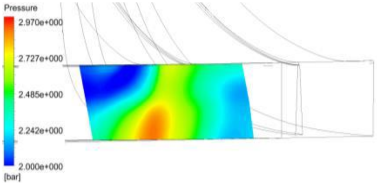
d) 0.5 mm



e) 0.7 mm



f) 1 mm



g) 2 mm

Fig. 6 – Pressure in Stn Frame for different tip clearances: a) 0 mm, b) 0.1 mm, c) 0.3 mm, d) 0.4 mm, e) 0.5 mm, f) 0.7 mm, g) 1mm, h) 2 mm

The two graphic in Fig. 7, presents the evolution of pressure on two streamlines positioned in the Z direction at first diffuser inlet, one in the right side and the other in the left side of the diffuser.

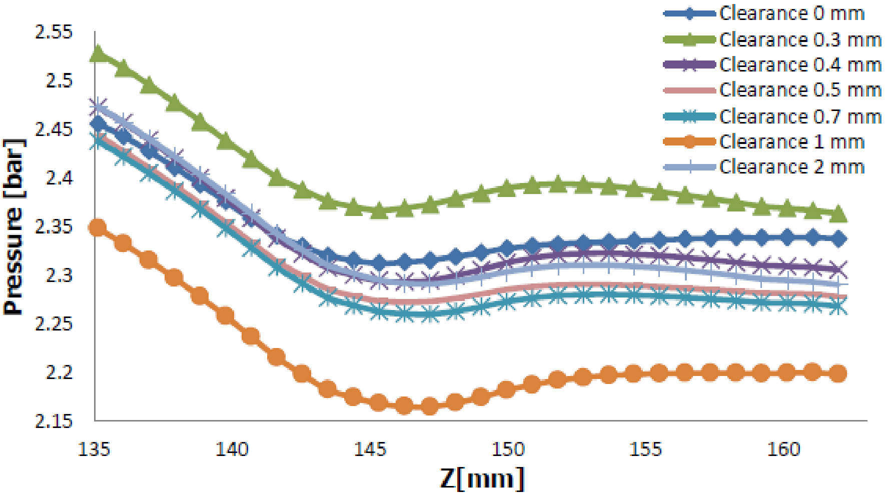
Pressure is presented from shroud to hub, a drop of pressure near the shroud can be seen, it is caused by the recirculation area that forms there.

The two extreme cases are the one with the tip gap of 0 mm (upper bound) and 1 mm (lower bound).

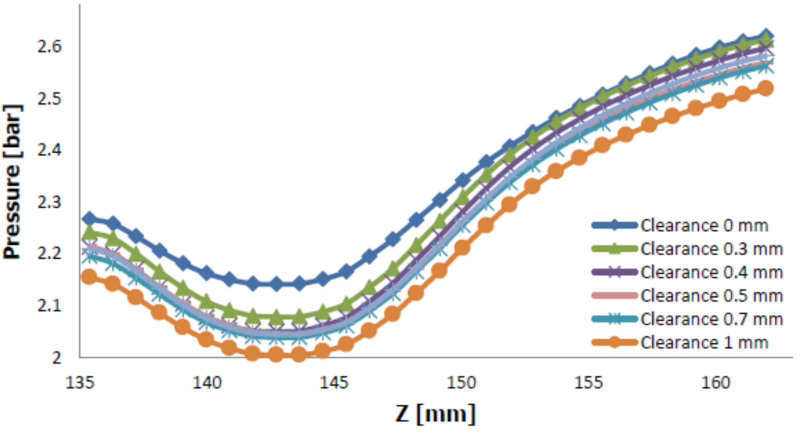
Figure 8 presents the evolution of the velocity in the Z direction, from shroud to hub. Differences regarding velocity variation are not so great between the seven cases, following the same path for all.

In the first part of the graphic, near the shroud, velocity is negative caused by formation of a recirculation volume.

In the proximity of the shroud values of velocity are increasing, indicating that the recirculation area appears only near the blade leading edge, and is caused by the secondary flow.

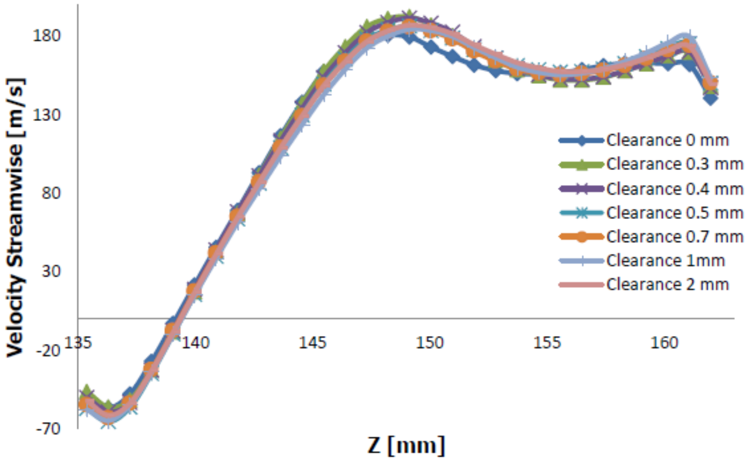


a) left

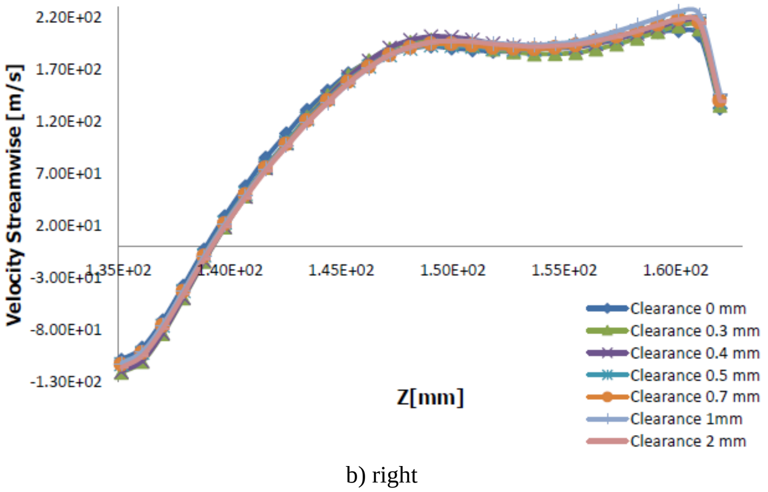


b) right

Fig. 7 – Pressure near the first diffuser inlet



a) left



b) right

Fig. 8 – Velocity Streamwise near the first diffuser inlet

Figure 9 presents relative error for the pressure loss between the components of the centrifugal compressor.
The lowest pressure loss is on the two diffusers, for a tip gap of 0.5 mm. On the entire stage, pressure loss is increasing with the tip clearance.
A gap of 0.7 mm leads to the major pressure losses, between each component and on the entire compressor stage.

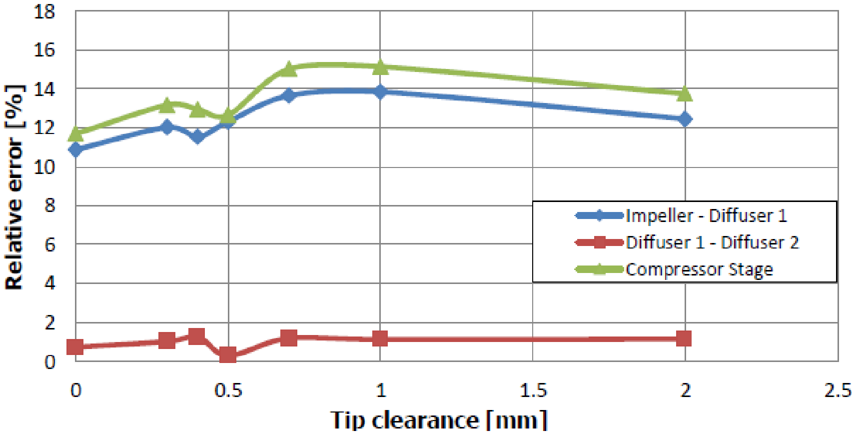


Fig. 9 – Relative error for the pressure loss

Figure 10 presents the evolution of polytropic efficiency for all seven cases. With the increase of tip clearance a non-uniform evolution of the efficiency can be seen.
The best result corresponding for this centrifugal compressor is at 0.3 mm gap, where for all components the polytropic efficiency and aerodynamic power has the highest value.
Worst results in terms of compressor performances are obtained when tip clearance is 1mm.

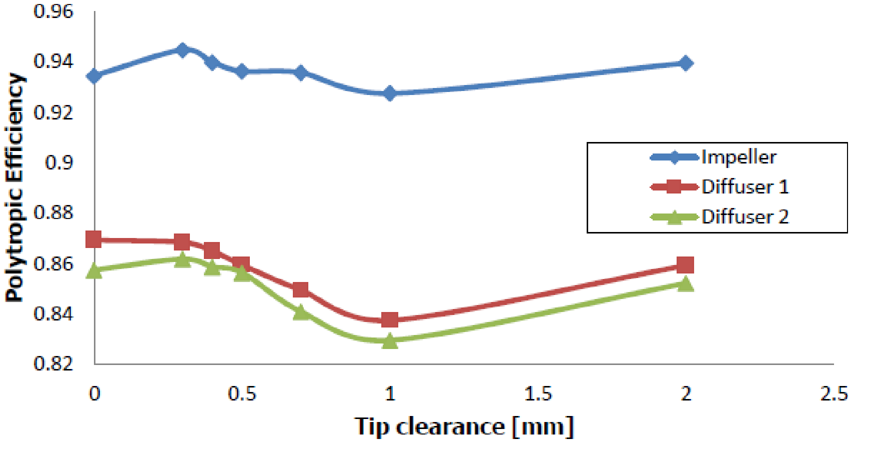


Fig. 10 – Evolution of Polytropic Efficiency with the increase of tip clearance

Figure 11 present polytropic efficiency vs. total to total pressure ratio. It has been shown that a change in tip clearance changes the performance of the centrifugal impeller.
For example, polytropic efficiency decreases by 3.4% for a 1 mm tip clearance (Fig. 11) and increases with 0.5% for a clearance of 0.3 mm.

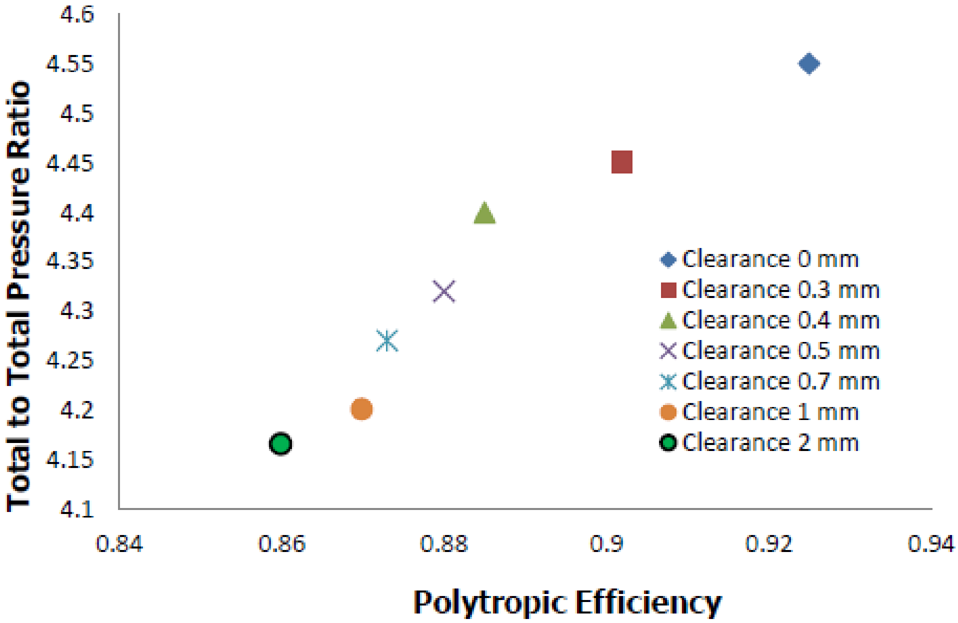


Fig. 11 – Polytropic Efficiency vs. Total to Total Pressure Ratio

4. CONCLUSIONS

The present paper analysis a centrifugal compressor from an industrial application power plant. All numerical simulations were performed using a RANS model that compensates for the rotation and curvature of the streamlines as well as the transport of frictional stresses.

The results showed that the position of the shock wave does not change with the increase of the tip clearance. In this particular case, the appearance of a small recirculation zone due to the rapid change from an axial flow to a radial one in the impeller which can be attributed to the detachment of the casing boundary layer is observed. The vortex attached to the tip of the rotor is proportionally correlated with the clearance between the impeller and the casing; increasing with it. The impact of this is shown by the value of the output flow angle, which is reduced by approximately 7 degrees for a 2 mm tip clearance. Aerodynamic losses due to shock wave, secondary flow and turbulence can be seen in the polytropic efficiency of the centrifugal impeller.

In conclusion, the optimal tip at which the rotor performance is the best is minimal, in his case 0.3 mm, but an acceptable compromise solution is around 2% of the blade height at the discharge. Continuing CFD investigation of loss tip mechanisms is critical, especially as they become more accurate and available. Understanding how total pressure or torque is lost in a compressor stage can lead to more efficient design methods, including precision of manufacturing, installation and operation.

ACKNOWLEDGEMENT

This study was carried out within the project “In-laboratory validation of a demonstrator compressor rotor with high pressure ratio, self diffusion and deswirling”, Grant no. 60 PED/2017.

REFERENCES

[1] T. Engin, M. Gur, R. Scholz, Effects of Tip Clearance and Impeller Geometry on the Performances of Semi-Open Ceramic Fan Impellers at Elevated Temperatures, *Experimental Thermal and Fluid Science*, **30**, 565–577, 2006.

[2] V. Dragan, Aerodynamic reconfiguration and multicriterial optimization of centrifugal compressors–a case study, *INCAS BULLETIN*, (online) ISSN 2247–4528, (print) ISSN 2066–8201, ISSN–L 2066–8201, Vol. **6**, issue 4, DOI: 10.13111/2066-8201.2014.6.4.4, pp. 41 – 49, 2014.

[3] V. Dragan, I. Malael, B. Gherman, A Comparative Analysis Between Optimized and Baseline High Pressure Compressor Stages Using Tridimensional Computational Fluid Dynamics, *ETASR*, Vol. **6**, No 4, 2016.

[4] G. Li-min, XI Guang, W. Shang-jin, Influence of Tip Clearance on the Flow Field and Aerodynamic Performance of the Centrifugal Impeller, *Chinese Journal of Aeronautics*, Vol. **15**, No.3, August 2002.

[5] M. Ishida, H. Ueki, Y. Senoo, Effect of Blade Tip Configuration on Tip Clearance Loss of a Centrifugal Impeller, *Journal of Turbomachinery*, **112**(1), 14-18, doi:10.1115/1.2927412 Jan 01, 1990.

[6] M. Marconcini, F. Rubechini, A. Arnone, S. Ibaraki, Numerical Investigation of a Transonic Centrifugal Compressor, *Journal of Turbomachinery*, Vol. **130**, Paper No. GT2006- 90098, pag 011010-1 -011010-8, 2008.

[7] V. Dragan, B. Gherman, I. Malael, R. Bimbasa, Global analysis of tip clearance influence on centrifugal compressor performance, *Trans Tech Publications, Applied Mechanics and Materials*, Switzerland, ISSN: 1662-7482, Vol. **811**, pp 128-132, 2015.

[8] Y. K. P. Shum, C. S. Tan, N. A. Cumptsy, Impeller – Diffuser Interaction in a Centrifugal Compressor, *ASME Journal of Turbomachinery*, **122**(4), pp. 777-786, 2000.

[9] D. You, M. Wang, R. Mittal, P. Moin, Study of Rotor Tip – Clearance Flow Using Large –Eddy Simulation, American Institute of Aeronautics and Astronautics, *AIAA2003-0838*, DOI: 10.2514/6.2003-838.

[10] I. Shahina, M. Alqaradawi, M. Gadala, O. Badr, Effect of Blade Tip Geometry on Centrifugal Compressor Performance and Stability, *Journal of Engineering and Applied Science*, **11**(12): 2769-2774, ISSN: 1816-949X, 2016.

[11] I. Malael, H. Dumitrescu, A. Dumitrache, *Methods for Improve the Performance of the Turbomachines Using the Flow Control*, AIP Conference Proceedings 1389, 1515, doi: http://dx.doi.org/10.1063/1.3637913, 2011.

[12] F. R. Menter, M. Kuntz, R. Langtry, *Ten Years of Industrial Experience with the SST Turbulence Model*, Proceedings of the 4th International Symposium on Turbulence, Heat and Mass Transfer, Begell House Inc., West Redding, pp. 625-632, 2003.

[13] E. A. Lurie, P. R. Van Slooten, G. Medic, J. M. Mulugeta, B. M. Holley, J. Feng, O. Sharma, R. Ni, *Design of a High Efficiency Compact Centrifugal Compressor for Rotorcraft Applications*, American Helicopter Society 67th Annual Forum, Virginia Beach, VA, May 3-5, 2011.

[14] H.-L. Wang, G. Xi, J.-Y. Li, M.-J. Yuan, Effect of the tip clearance variation on the performance of a centrifugal compressor with considering impeller deformation, *Proceedings of the Institution of Mechanical Engineers, Part A Journal of Power and Energy*, December 2011, DOI: 10.1177/0957650911416914.