iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 5 **Issue:** VII **Month of publication:** July 2017

DOI:

www.ijraset.com

Call: 08813907089 | E-mail ID: ijraset@gmail.com

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor:6.887
Volume 5 Issue VII, July 2017- Available at www.ijraset.com

Computational Analysis of a Centrifugal Compressor with Partial Shroud

S. M. Swamy[1], V. Panndurangadu[2], N. Sitaram[3] , V. Naga Prasad Naidu[4]
[1]*Assistant Professor, Department of Mechanical Engineering, GNITS, Shaikpet, Hyderabad-500104, India.*
[2]*Professor, Department of Mechanical Engineering, JNTUA, Ananthapuramu -515002, A.P, India.*
[3]*Professor, Department of Mechanical Engineering, IIT-MADRAS.*
[4]*Professor & Principal, Department of Mechanical Engineering Intell Engineering College, Anantapur, A.P.*

Abstract: Computational analysis of low speed centrifugal compressor is carried out by inexpensive partial shroud near the rotor blade tip with finite volume method using ANSYS-CFX software. Centrifugal compressor impeller with three values of clearances i.e., τ= 2.2%, 5.1% and 7.9% of blade height at trailing edge are examined at four flow coefficients φ=0.12, 0.18, 0.28 and 0.34. The effects of tip clearance with two configurations on static pressure from inlet to outlet of the compressor variations are analyzed. The drop in static pressure with increase in tip clearance is found to be high at the tip of the blade due to high pressure fluid leakage at the tip of the blade. The improvement in the compressor performance may be due to the reduction of tip leakage flows by the small extension of partial shroud (2 mm on the pressure surface side). Performance reduction with tip clearance is observed. Total pressure and velocity at outlet are analysed for four flow coefficients.
Keywords: Centrifugal Compressor, Flow Coefficient, Tip Clearance and partial shroud.

I. INTRODUCTION

The tip leakage flow thus would have dominant effect on the performance of a compressor. In turbomachines, to desensitize tip clearance effects, squealer tips / partial shrouds, tip geometry modifications, casing treatment etc. are suggested in the literature. P. Usha Sri and N. Sitaram observed that the impeller with the chamfer on suction surface of the blade tip shows small improvement in performance. Senoo and Ishida (1987) gave analytical expression to quantify the tip clearance effects in centrifugal blowers Hayami (1997) has found from his experiments that axial movement of the casing has better efficiency over the movement of casing in radial and axial directions. With an increase in the chamfer dimension on suction surface of blade tip, performance improvement is observed. S. Senthil and N. Sitaram studied the performance of a centrifugal compressor by means of squealer tips. They observed increase in energy coefficient and efficiency with squealer tips on pressure surface. Fayez M. Wassef et. al have conducted experiments on centrifugal compressor and observed improvement in limit of stability with addition of a ring and a groove in the casing in diffuser region.

II. COMPUTATIONAL DETAILS

The design details of the impeller which is used in the investigations are given below:

Inducer hub diameter, d_{1h}	= 160 mm	Inducer tip diameter, d_{1t}	= 300 mm
Impeller tip diameter, d_2	= 500 mm	Blade height at the exit, b_2	= 34.7 mm
No. of blades of impeller, N_b	= 16	Blade angle at inducer hub, β_{1h}= 53^o	
Blade angle at inducer tip, β_{1t}	= 35^o	Blade angle at exit, β_2	= 90^o
Thickness of the blade, t	= 3 mm	Rotor speed, N	= 2000 rpm

All angles are with respect to the tangential direction

A. Partial Shroud

The partial shrouds are made of stainless steel of 0.1 mm thickness. The stainless steel sheet is cut to the shape of rectangle pieces of 50 mm x 5 mm size. These rectangle pieces are attached to the tip of the blades using araldite. The configurations tested (basic configuration without partial shroud and configuration with partial shroud) are shown in Fig. 1.

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor:6.887
Volume 5 Issue VII, July 2017- Available at www.ijraset.com

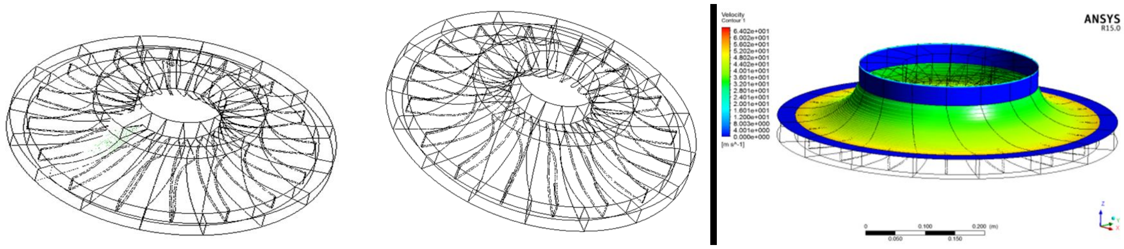


Fig. 1 Centrifugal compressor without and with partial shroud

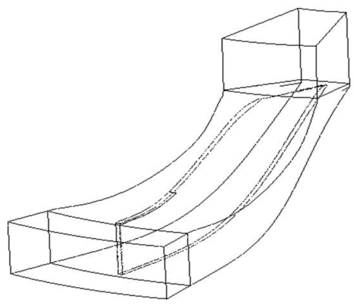


Fig. 2 Computational domain of single passage with and without partial shroud

Centrifugal impeller with above specifications with 2 mm thickness throughout the blade, 2.2% tip clearance for two configurations is shown in Fig. 1. Single passage of centrifugal impeller with and without partial shroud is analysed. A single passage of the impeller with inlet at 50 mm ahead of the impeller and outlet at a distance of 35 mm downstream of impeller is shown in Fig. 2. Casing is designed with a clearance of 0.7 mm throughout the blade height. Total pressure is used for inlet boundary condition and at outlet is mass flow rate. Rotating frame of reference is given to the domain. ANSYS-CFX 15.0 software is used for obtaining the solution and standard SSTk-ε model is used for the closure. The centrifugal compressor is analysed at four different flow coefficients (0.12, 0.18, 0.28 and 0.34), the design flow coefficient being 0.34.

III. RESULTS AND DISCUSSIONS

Low speed centrifugal compressor with three tip clearances τ=2.2%, 5.1% and 7.9% at four flow coefficients ϕ = 0.12, 0.18, 0.28 and 0.34 were analysed for with and without partial shroud. Static pressure distribution along streamwise direction from inlet to outlet of the domain and total pressure distribution along streamwise direction from inlet to outlet of the domain with and without PS with four flow coefficient for three tip clearances were plotted. Total pressure graphs, blade loading charts, pressure contours and velocity vectors are analysed.

A. Static Pressure Distribution

Static pressure distribution along streamwise direction from inlet to outlet is shown in **figure 3**. This plot shows that the pressure from inlet to the outlet of the compressor is increasing gradually along the stream wise direction due to the dynamic head developed by the rotating impeller. Static pressure drop at impeller leading edge is observed which causes the fluid to accelerate in to the compressor. In the impeller passage static pressure coefficient increase is observed due to the energy imparted by the impeller. A drop in static pressure near streamwise direction of 0.4 is observed for all cases due to the acceleration of the flow in to the eye of the impeller. Static pressure reduction is observed with PS on tip of the blade. With PS static pressure is constant before the impeller passage. Pressure is increasing steadily in the impeller passage for all tip clearances due to the energy transfer taking place the impeller. The drop in static pressure with increase in tip clearance is found to be high at the tip of the blade due to high pressure fluid leakage at the tip of the blade.

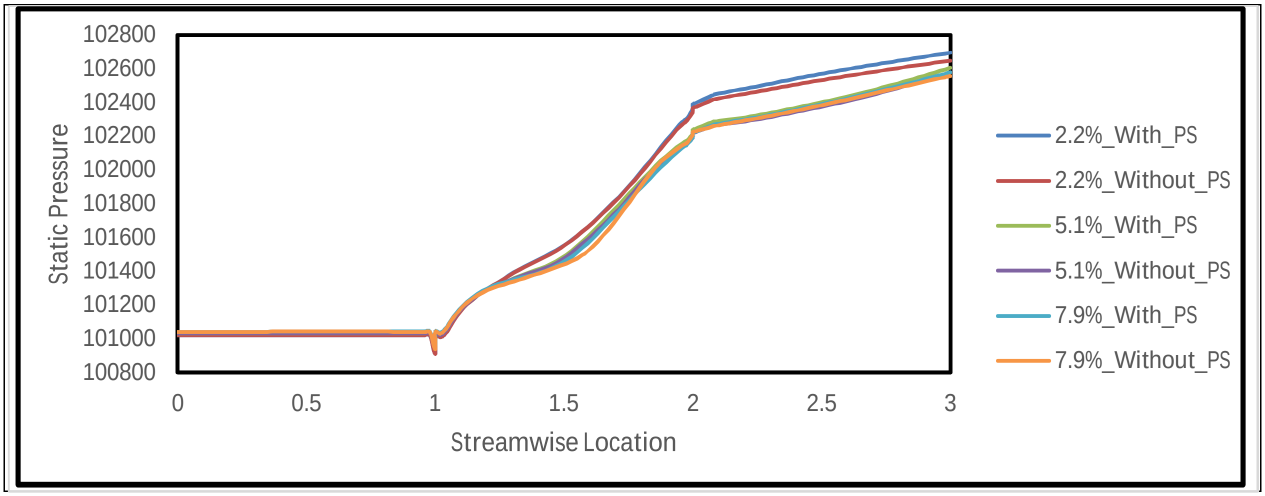


Fig. 3 Static Pressure from inlet to outlet

B. Total Pressure Distribution

Total pressure distribution along streamwise direction with and without PS is shown in **figure 4**. Total pressure coefficient is constant before the impeller passage and increasing steadily in the impeller passage for all tip clearances τ=2.2%, 5.1% and 7.9% due to the energy transfer taking place the impeller. Downstream of the impeller passage a small total pressure coefficient drop is observed for all tip clearances. Gradual increase of pressure along streamwise direction because of dynamic action of the impeller is observed. Total pressure improvement with PS is observed. With PS on tip of the blade, the fluid flow from pressure side to suction side of the blade is interacting with passage wake on the suction side of the blade. For all the flow coefficients it is seen that the total pressure decreases with increase in tip clearance. From the impeller leading edge, total pressure coefficient is decreasing with increase in tip clearance from 2.2% to 7.9% clearance. At larger flow coefficients the reduction in total pressure with tip clearance is high. At the tip of the blade, the total pressure coefficient reduction with increase in tip clearance is significant due to high pressure fluid leakage at the tip of the blade. But for 2.2% clearance at φ=0.28, the total pressure is higher than the other clearances as the mass flow rate increasing above operating range.

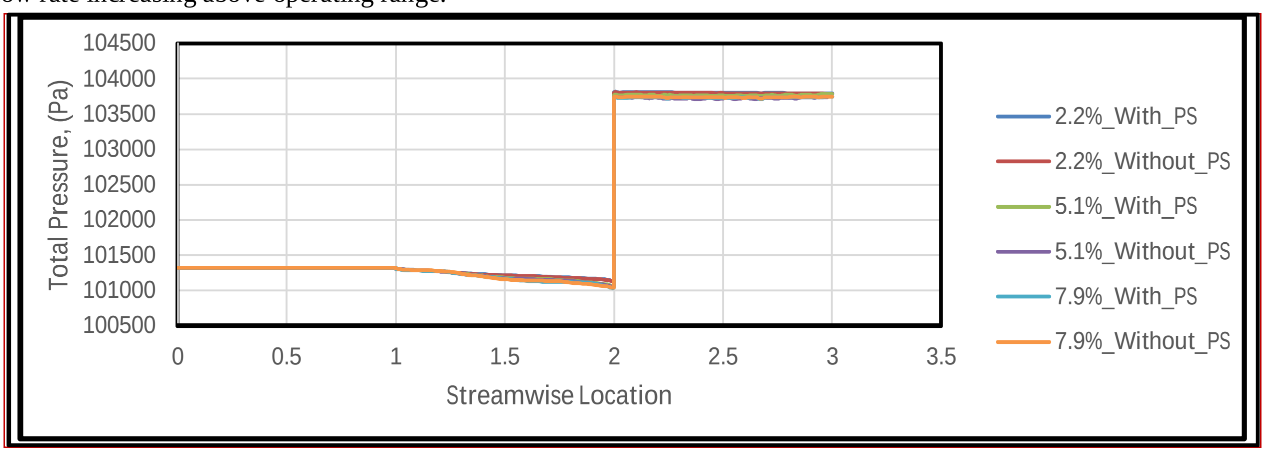


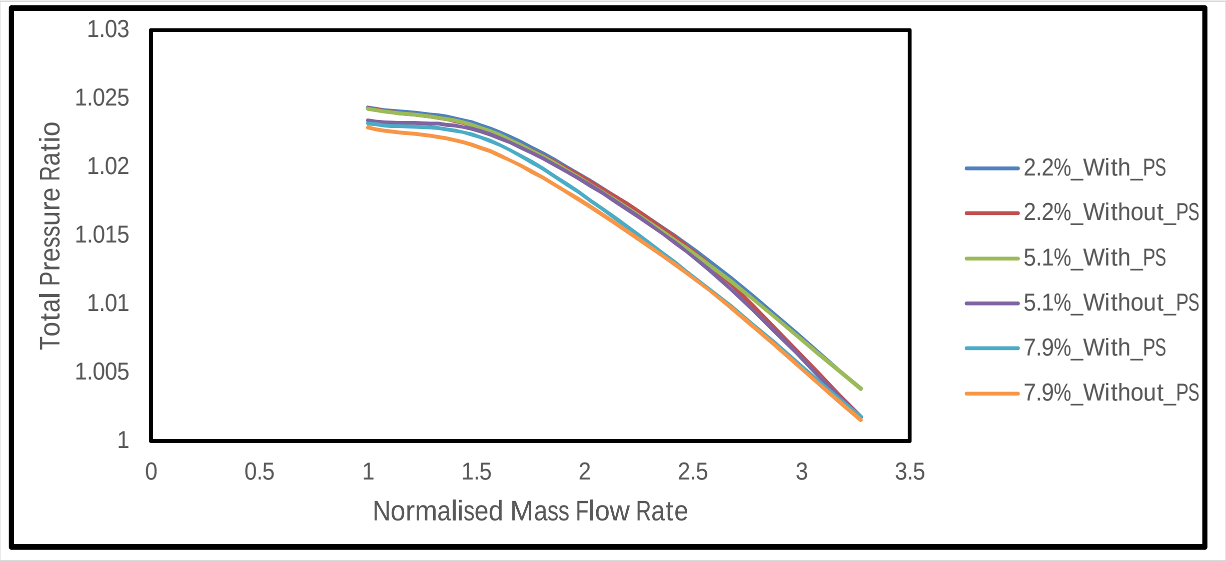
Fig. 4 Total Pressure from inlet to outlet

C. Mass Averaged Flow Performance Of Impeller

The mass averaged values of total pressure and static pressure at the rotor exit for four flow coefficients φ=0.12, 0.18, 0.28 and 0.34 and for the three values of tip clearance τ= 2.2%, 5.1% and 7.9% for both configurations is show in figure 5. The static pressures over the entire operating range for all blades with partial shroud reduced tip clearances were higher than those in the basic configuration case, indicating that reductions in the tip clearance at the tip of the blade improved the static pressure rise. The exit of the impeller total pressure also increased for the impellers with reduced tip clearances. With partial shroud had the highest static and

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor:6.887
Volume 5 Issue VII, July 2017- Available at www.ijraset.com

total pressures as compared with the basic configuration, because the tip clearance was reduced at the tip of the blade. At φ/φ_d of τ=2.2% tip clearance shows improvements in the static pressure and total pressure rise compared to 5.1% and 7.9% tip clearance respectively. Therefore, the decreased tip clearance was still effective even at the lower mass flow rate. The performance curves drawn from mass average total pressure are showing clearly that partial shrouds are beneficial in improving the pressure rise of the compressor, compared to the basic configuration near φ=0.18. For 7.9% tip clearance at φ=0.34, performance is badly affected because of the mass flow rate falling below the operating range.



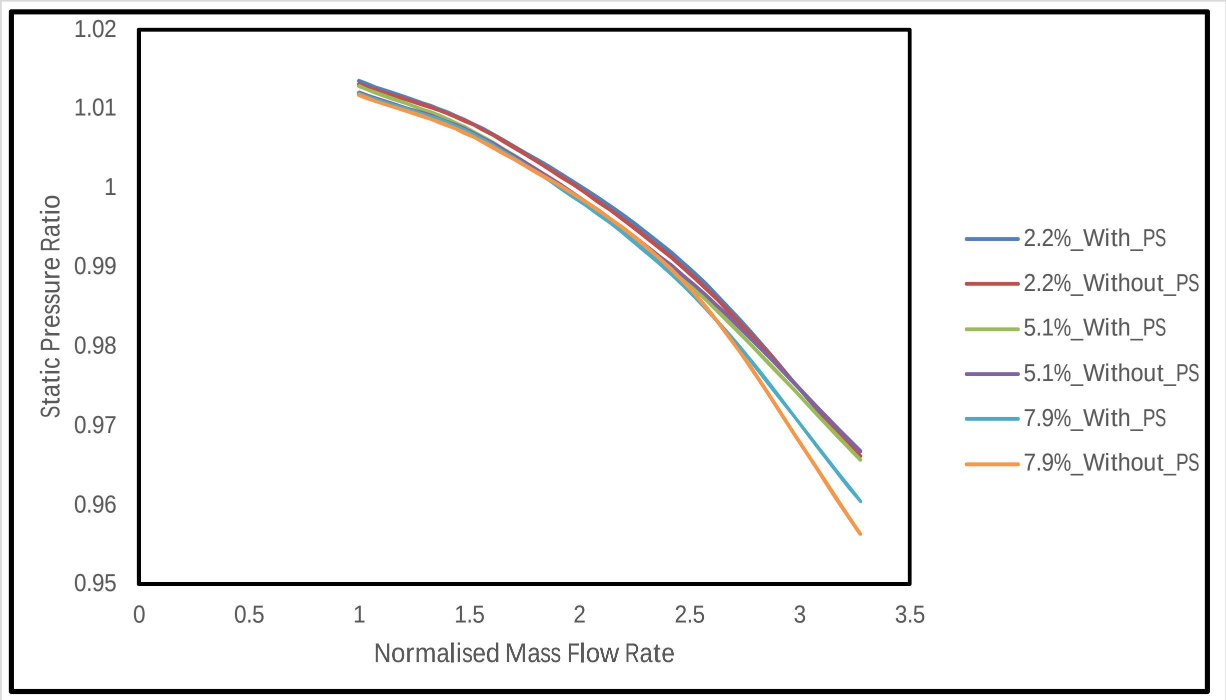


Figure 5: Comparison of the static and total pressure characteristic curves

D. Static Pressure Contours on Casing

Pressure contours on casing for two configurations, without and with PS on tip of the blade for different flow coefficients (φ =0.12, =0.18, =0.28, and =0.34) for 2.2% tip clearance is shown in figure 6. The contours show that gradual pressure rise from inlet to outlet of the compressor due to dynamic action of the rotating impeller. Pressure gradient above the blade is observed due to the high pressure on pressure side and low pressure on suction side of the blade. For with PS on tip of the blade, significant change in pressure is observed. The low pressure on both pressure and suction side of the blade is observed for without PS on tip of the blade

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor:6.887
Volume 5 Issue VII, July 2017- Available at www.ijraset.com

due to more leakage of flow.

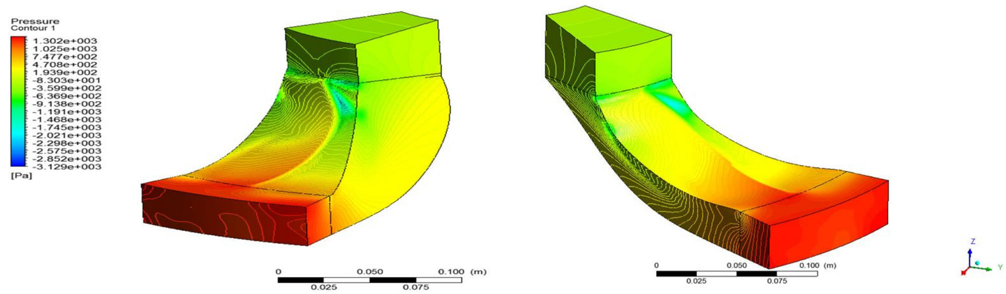


Fig. 6. Pressure contours on casing without PS and with Partial Shroud 2.2% tip clearance

E. Velocity Contours on Casing

Velocity contours on casing for two configurations, without and with PS on tip of the blade for different flow coefficients (φ =0.12, =0.18, =0.28, and =0.34) for 2.2% tip clearance is shown in figure 7. The contours show gradual increase of velocity from inlet to outlet of the impeller. The tip clearance for 2.2%, due to the boundary layer on stationary casing and rotation vertical flow is observed. Fluid flow with PS on tip of the blade, significant change in pressure is observed the velocities are high on suction surface than pressure surface because of blade curvature. For other clearances, near the suction side shroud corner leakage flow is rolling up forming a wake. With the increase in the clearance, the wake region is increasing as leakage fluid mass flow rate increases with PS as compared without PS.

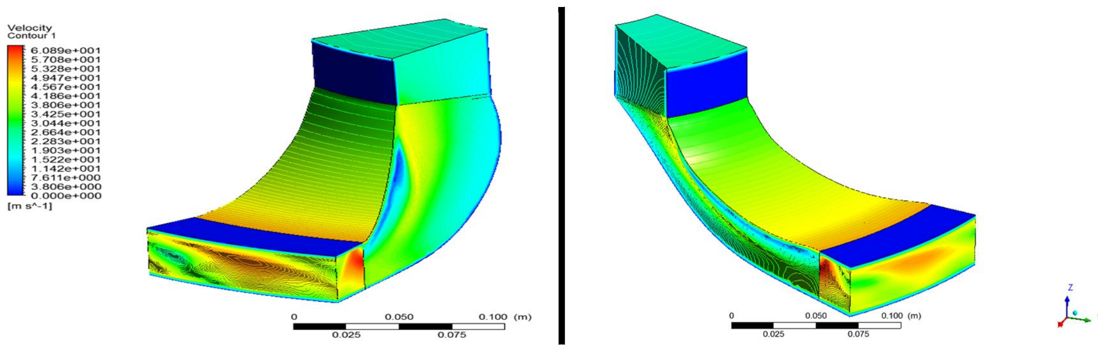


Fig. 7. Velocity contours on casing without PS and with PS

F. Total Pressure contours at Meridional Plane surface 1.8

Total pressure contours in blade to blade view, at turbo span 1.8 for two configurations, without and with PS on tip of the blade for different flow coefficients (φ =0.12, =0.18, =0.28, and =0.34) for 2.2% tip clearance is shown in figure 8.Total pressure contours on meridional plane for two cases on suction side near shroud, low total pressure area caused due to passage wake is observed. With partial shroud on tip of the blade, low total pressure area of passage wake is reduced. The leakage flow from the tip gap is interacting with passage wake. Without PS, the passage wake area is increased, but total pressure in this area is lower due to more leakage of flow from tip gap between shroud and tip of the blade is observed.

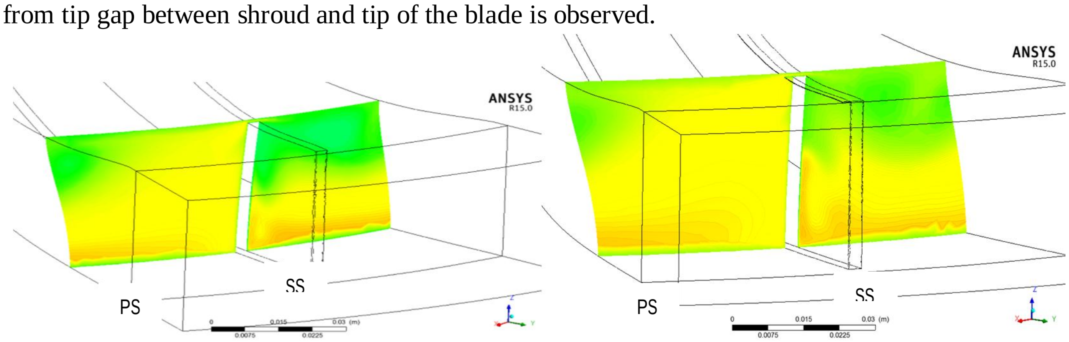


Fig. 8. Total Pressure contours at Meridional Plane surface 1.8 without PS and with PS at 2.2% tip clearance for flow coefficient 0.34.

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor:6.887
Volume 5 Issue VII, July 2017- Available at www.ijraset.com

G. Static Pressure Contours at Span 0.7

Static pressure contours in blade to blade view, at span 0.7 for two configurations, without and with PS on tip of the blade for different flow coefficients (φ =0.12, =0.18, =0.28, and =0.34) for 2.2% tip clearance is shown in figure 9&10. The contours show gradual pressure rise from inlet to outlet of the compressor due to dynamic action of the rotating impeller. Gradual increase of static pressure from inlet to outlet is clearly observed at all tip clearances. With PS on tip of the blade, low pressure change is observed. But without PS on tip of the blade, the pressure at outlet is reduced. High pressure on pressure side of the blade and low pressure on suction side of the blade are observed at all tip clearances. With increase in tip clearance, reduction in pressure on both pressure side and suction side is found.

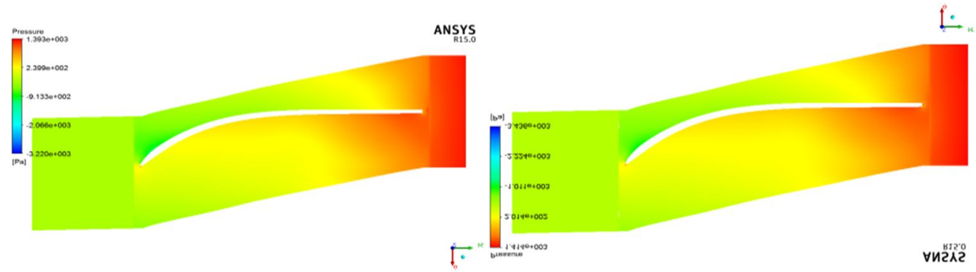


Fig. 9: Static pressure contours for 2.2% clearance at mid span without PS and with PS at span 0.7

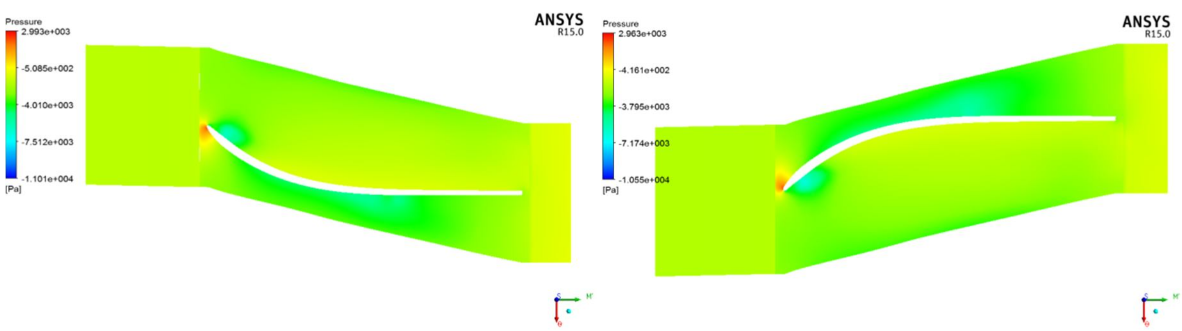


Fig. 10: Static pressure contours for 5.1% tip clearance without PS and with PS at span 0.7

H. Velocity Contours at Span 0 .7

Velocity contours in blade to blade view, at span 0.7 for two configurations, without and with PS on tip of the blade for different flow coefficients (φ =0.12, =0.18, =0.28, and =0.34) for 2.2% tip clearance is shown in **figure 11**. The contours show low velocity region on suction side of the blade. With PS on tip of the blade, the low velocity region is reducing and also velocity improvement is observed.

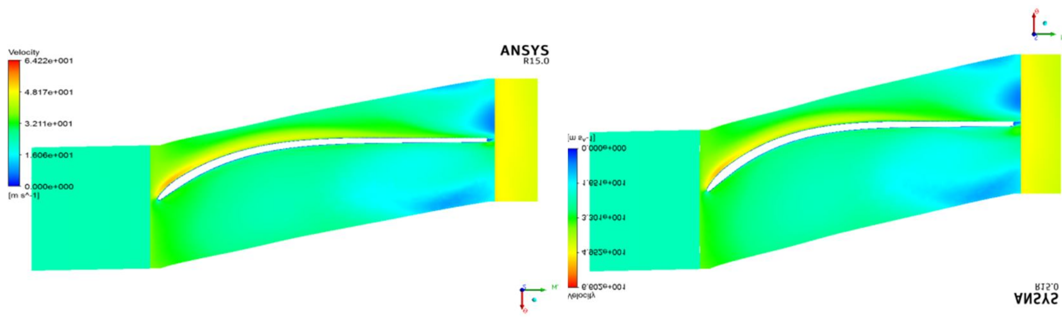
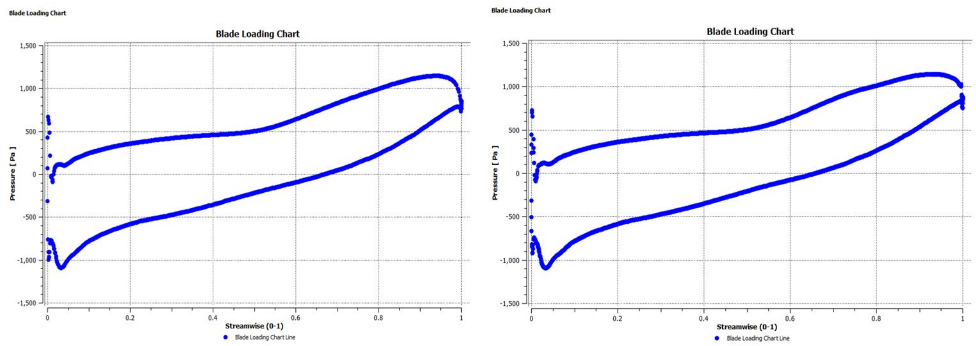


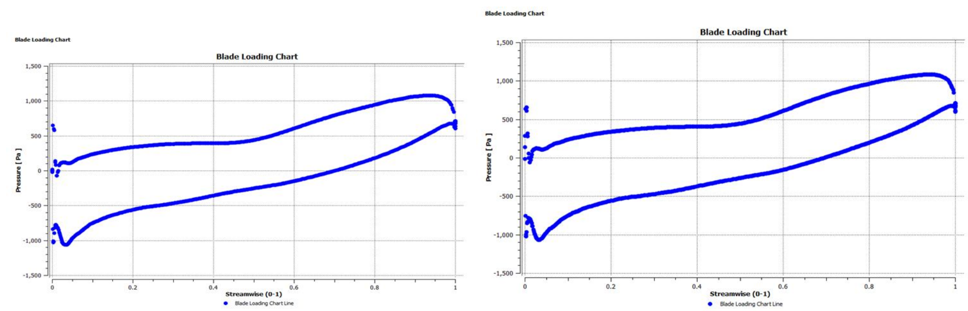
Fig. 11: Velocity contours for 2.2% clearance for without PS and with PS

I. Blade loading chart

Blade loading curve for 2.2% and 5.1% tip clearances at without PS and with PS were shown in figure 12. Fluid flow without PS and with PS on tip of the blade, significant change in pressure on suction side is observed. Low static pressure on suction and high pressure on pressure side of the blade is observed. With increase in tip clearance for 2.2% and 5.1% tip clearances, static pressure on both pressure side and suction side are reducing.



Blade loading for 2.2% clearance at without and with partial shroud



Blade loading for 5.1% clearance at without and with partial shroud

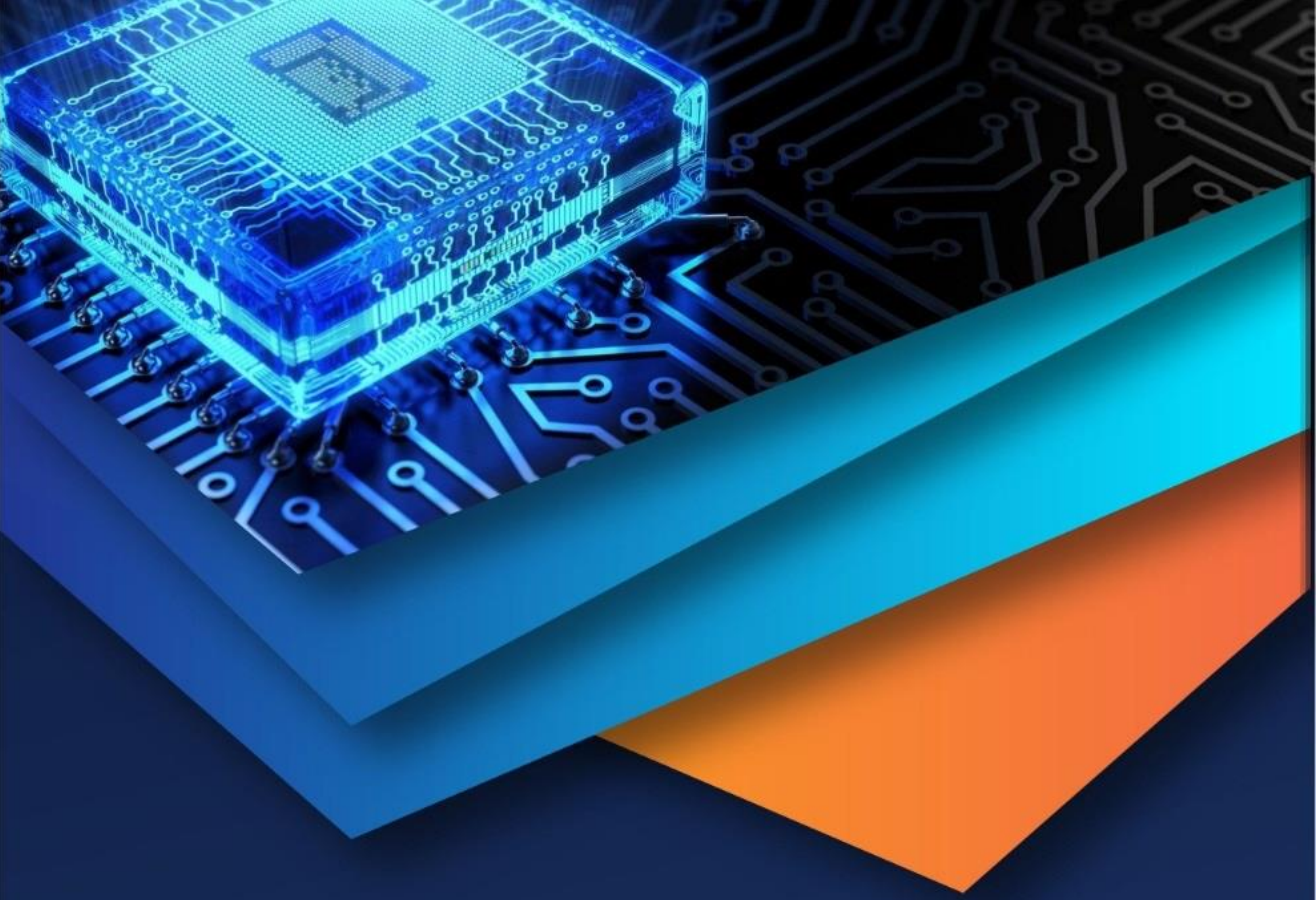
Fig.12**:** Blade loading for 2.2% and 5.1% tip clearances at without and with partial shroud

IV. CONCLUSIONS

Tip clearance effects in a low speed centrifugal compressor impeller with three different values of clearances i.e., 2.2%, 5.1% and 7.9% are examined at four flow coefficients 0.12,0.18,0.28 and 0.34. The static pressure and total pressure distribution from inlet to outlet of the compressor and total pressure at exit, of the compressor show that with increase in tip clearance the losses increase. The drop in static pressure with increase in tip clearance is observed to be high at the tip of the blade due to high pressure fluid leakage at the tip of the blade for 2.2% to 7.9% clearance. With increase in tip clearance, the velocity on both pressure side and suction side is decreasing. For 2.2% to 7.9% clearance, high velocity of the fluid above the blade from pressure side to suction side through tip clearance is clearly seen.

REFERENCES

[1] Fowler, H. S., (1971) Aerodynamics Tests on a Centrifugal Fan Impeller Model with Swept Back Blades, N.R.C. Report No. 12020.
[2] Senoo, Y. And M. Ishida, (1975) Behaviour of Severely Asymmetric Flow in a Vaneless Diffuser, Trans. ASME, Jl. of Engg. for Power, 97, 375-382
[3] Moore, J., (1973), "A Wake and an Eddy in a Rotating Radial-Flow Passage", Part 1: Experimental Observation", Part 2:"Flow Model", Trans. ASME Journal of Engineering for Power.pp.205-219.
[4] Eckardt, D., (1976) Detailed Flow Investigations within a High Speed Centrifugal Compressor Impeller, Trans. ASME, Jl. of Fluids Engg. 98, 390-402.
[5] Adler, D., (1980) Status of Centrifugal Internal Aerodynamics, Part II: Experiments and Influence of Viscosity, Trans. ASME, Jl. of Engineering for Power, 102, 738-74
[6] Shivaprasad, B. G. and B. R. Ramaprian, (1977) Turbulence Measurements in Boundary Layers along Mildly Curved Surfaces, ASME Paper 77-WA/FE-8
[7] Krain, H. and D. Eckardt, (1978) The Flow Field in a High Speed Centrifugal Impeller-A Comparison of Experimental and Theoretical Results, First International Conference on Centrifugal Compressor Technology, IIT Madra
[8] Johnson, M.W., and Moore, J., (1980), "The development of wake flow in a centrifugal compressor" ASME Journal of Engineering for Power, Vol. 102, pp. 383-390.



10.22214/IJRASET

INDEX COPERNICUS
45.98

IMPACT FACTOR:
7.129

TOGETHER WE REACH THE GOAL
IMPACT FACTOR:
7.429

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY