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The Influence of Tip Clearance on Centrifugal Compressor Stage of a Turbocharger

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Abstract: - To understand the effect of tip clearance on a centrifugal impeller and diffuser, detailed numerical simulations were carried out. The impeller and diffuser were connected through rotor stator boundary using frozen rotor approach. Overall performance and the flow configuration have been investigated for nine tip clearance levels from no gap to 1mm. It has been observed that at the mass flow rate of 0.35 kg/s, a loss of about 15 percent in the stage pressure ratio is observed as the tip clearance is increased from zero to 1mm whereas the peak efficiency has also been reduced by 10 percent. The impeller efficiency is found maximum at tip clearance between 0.1 to 0.2 mm however minimum diffuser effectiveness is also observed at the same clearance level. Diffuser effectiveness is found to be maximum after 0.5mm gap. Mass averaged flow parameters, entropy, blade loading diagram and relative pressure fields are presented showing the loss production within the impeller passage with tip clearance.

Key-Words: - Centrifugal Impeller, Diffuser, Stage, Tip clearance, Numerical simulation, Entropy.

1 Introduction

The flow structures within the Centrifugal Compressors are considered amongst the most complicated and convoluted in all turbomachinery. In recent past paramount advancement in the performance of the Centrifugal Compressor has been made primarily because of the computer aided design and analysis techniques that are cautious combination of empirical correlations and extensive modeling of the flow physics.

Unshrouded centrifugal compressors are mostly favored compared to shrouded compressors in order to avoid high stresses involved with the increased weight. As a result the leakage flow through the tip clearance between the blades and casing is inevitable that further complicates the flow and may depreciate the overall performance of the centrifugal compressor. There are two unique and evenly significant aspects of the tip clearance flows as suggested by Denton and Cumpsty [1]. First the reduction in the blade force and the second foremost aspect is the mixing of flow through the tip clearance gap with the flow between blades. The interaction of tip clearance flow field, blade vortex flow and leakage vortex flow generates an extremely complex flow structure.

In recent past, a number of numerical and experimental investigations have been conducted [2-6] to investigate the effect of tip clearance in unshrouded compressors. Ma et al. [2] numerically investigated the effect of tip clearance on the performance and flow characteristics of a centrifugal impeller. Entropy fields and the secondary flow development were presented showing the loss production within the impeller passage. No optimum clearance was found for all simulated results except no gap level. Usha and Sitaram [3] numerically predicted that the performance was degraded with the increase in tip clearance. Gao et al. [4] used their own CFD code to investigate the effect of tip clearance on 3D viscous flow field and performance of NASA LSCC impeller with a vaneless diffuser. The study indicated that the location of the throughflow wake was influenced by the tip clearance and there probably exist an optimal clearance at which flow loss was minimum. Their simulations indicated that the optimum clearance was about 0.9 percent of the blade height. Eum et al. [5] numerically studied six clearance levels. The effect was decomposed into inviscid and viscous components using one-dimensional model expressed in terms of the specific work reduction and the additional entropy generation. Both inviscid and viscous effects affected performance to similar extent, while efficiency drop was mainly influenced by viscous loss of the tip leakage flow. Performance reduction and efficiency drop due to tip clearance was proportional to the ratio of tip clearance to blade height. The proposed 1D model was found to be in close agreement with the experimental results. Hong et al. [6] experimentally measured the discharge flow of a centrifugal compressor at six levels of tip clearance. The study found an optimum tip clearance ratio of 0.12 in terms of surge margin, however the overall performance degradation was found with the increase in tip clearance. They also concluded that the wake region was increased with tip clearance and the deficit of the relative total pressure governed the wake region therefore the loss was magnified.

In order to study the effect of tip clearance, there are two methods to increase the tip clearance. In first method, the distance between hub and shroud is kept constant while blade height is reduced. In second method, the blade height is kept constant but the shroud radius is increased. The present study use 1st method to increase the tip clearance. The results of second method will be discussed in the next paper. The impeller and diffuser are evaluated for nine tip clearance levels (zero, 0.1mm, 0.2mm, 0.3mm, 0.4mm, 0.5mm, 0.6mm, 0.8mm and 1mm) at various mass flow rates ranging between stall to choke conditions. The simulations were executed and the results were predicted keeping in mind the existing impeller theories for secondary flow transport, jet-wake flow and internal diffusion. This paper is ordered as follows. First of all the description of studied compressor stage is provided. Computation method is then discussed in section 3 with a brief description of CFD software package. Results of CFD are then presented.

2 Description of Compressor Stage

As shown in figure 1, the impeller has 7 full blades and 7 splitter blades with features shown in table 1 and figure 2. Splitter blade leading edges are located at 30% of full blade chord. The exit diameter of the impeller is 90 mm and the nominal point tip speed is 377 m/s at 80000 rpm. A vane less diffuser (6.5mm x 26.75mm) is connected aft the impeller

Fig.1 Investigated Centrifugal Compressor

Table 1 Dimensions of Impeller

Number of Blades (Full + Splitter)	7 + 7
Leading edge hub angle	31^o
Leading edge shroud angle	46^o
Trailing edge hub angle	-27^o
Trailing edge shroud angle	-33^o

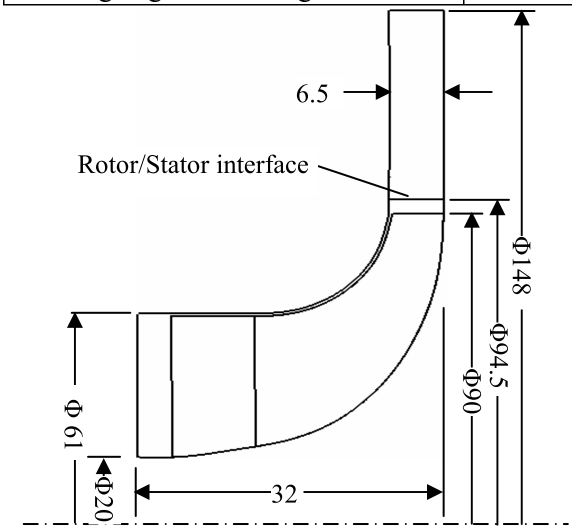

Fig. 2 Meridional View of the Blade (Dim. in mm)

3 Computational Method

The mesh was created using NUMECA's IGG/AutoGrid [10]. Briefly AutoGrid is an automatic meshing scheme for turbomachinery configurations. It provides tools to generate automatically a turbo-machinery mesh and ensure an optimal control of orthogonality and mesh point clustering for a correct depiction of viscous effects in the boundary layer.

A sixteen-block mesh was generated using Auto Grid. The four main blocks represent part of the blade-splitter-blade passage extending from inlet to outlet. These blocks are generated using H-type mesh with 9x45x85 points in each block. Two C-type skin mesh blocks (9x45x137 and 9x45x97) containing the pressure and suction surfaces of both full and splitter blades are generated in order to improve the orthogonality. Two blocks of 41x45x9 points each are generated downstream of the full blade and splitter blade. To capture the flow physics upstream of the blades, two mesh blocks of 9x45x9 and 9x45x33 points are generated before main and splitter blades respectively. Diffuser is split into two blocks containing 57x45x17 points each. Rests of the four blocks (9x13x137, 9x13x65, 9x13x97 and 9x13x45) are generated automatically with fine resolution (13 to 17 nodes from 0.1-1mm clearance) to explore the flow phenomenon inside tip clearance region. Thus, a total of mesh points are 411,768.

In order to investigate the impeller and diffuser separately, a rotor/stator boundary is set in CFD model as shown in figure 2 and frozen rotor approach is used in order to impose the continuity of velocity components and pressure
The computational simulations of the impeller physics were executed using the NUMECA's EURANUS [10] flow solver. Spalart Allmaras model is used for turbulence modeling. The spatial discretization is based on a finite volume approach allowing a fully conservative discretization. An explicit time discretization is applied through a multi-stage Runge-Kutta procedure.
The computer runs for the present study were executed on Pentium IV 2.8GHz and with 512 MB of memory. Typically, for each computation, 600 iterations were enough to reduce the mass flow residual by two orders of magnitude except for few computations where the iteration were increased up to 700.Each computation for the finest grid generally requires 3 hrs.

4 Results and Discussions

4.1 Overall Performance

The performance maps of the stage at various clearance levels are shown in Figures 3. The experimental and CFD results are found to be in close agreement. Minor deviations are found primarily because of the redesign of diffuser. The behavior of the CFD curves seems to satisfy the designer's goal. The mass flow rates and clearance sizes were chosen for two reasons. Firstly to undergo a parametric survey of the effect of tip clearance on the impeller performance and secondly to find the possible optimum size of the tip-clearance, which is not the zero tip clearance as indicated by Gao et al. [4].
Similar to the work of Ma et al. [2], the characteristic curves imply a considerable reduction in performance. At mass flow rate of 0.35 kg/s, a loss of about 15 percent in peak pressure rise is observed as the tip clearance is increased from zero to 1mm.The results follow that a linear assumption specifies a drop of peak pressure ratio by 0.036 for every 0.1mm increase in tip clearance. The peak efficiency has also been reduced by 10 percent for the same case.
Figures 3 further reflects that in contrast to Gao et al. [4] and in agreement to [2], [3] and [6], there is no clearance level indicating better performance than zero tip clearance hence the optimum tip clearance level, other than zero, doesn't exist for the compressor stage under study.

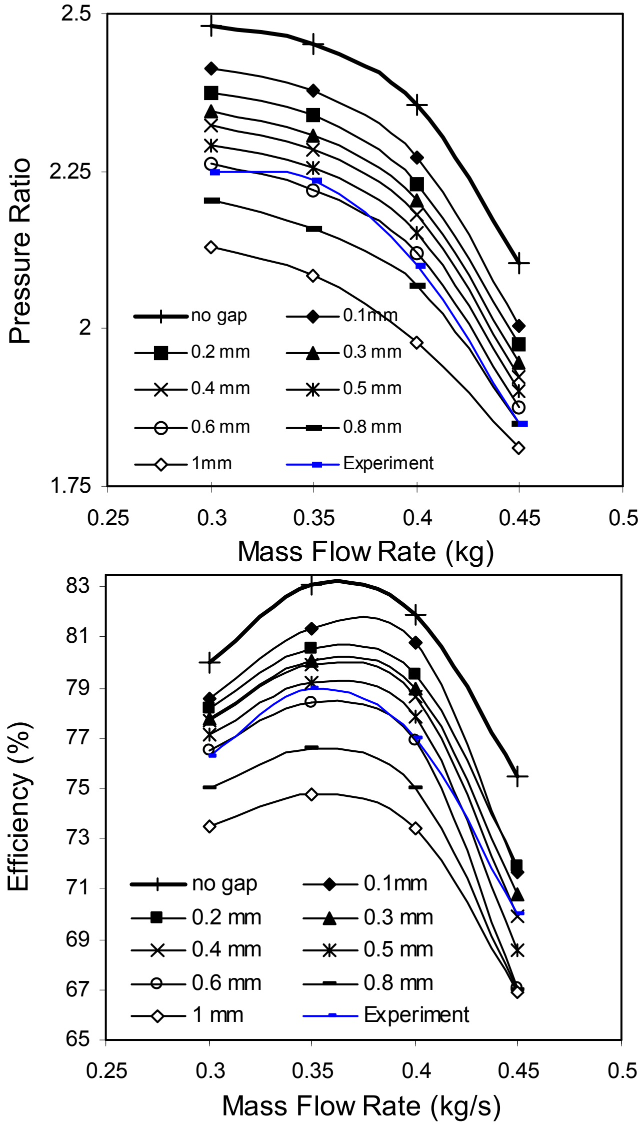

Fig. 3 Characteristic Curves of the stage

Figure 4 is the plot of total pressure ratio against the tip clearance from 0.1 to 1mm. Curve fitting of the data reveals that all the characteristic curves have almost the identical slope which can be given as:

$$\frac{dP}{d\lambda} \cong -0.295; \quad 0.1\text{mm} < \lambda \le 1\text{mm} \tag{1}$$

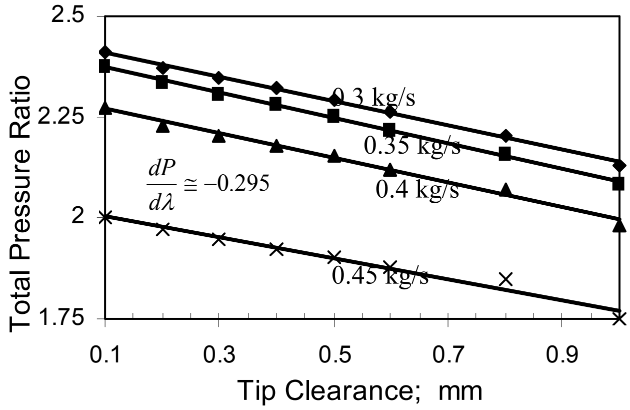

Fig. 4 Stage total pressure ratio Vs tip clearance

Separate analysis of impeller and diffuser reveals interesting results. Figure 5 shows impeller efficiency for the mass flow rate of 0.35kg/s at various tip clearance levels. The curves show that the impeller efficiency is maximum at tip clearance between 0.1 to 0.2 mm

Diffuser effectiveness at 0.35 kg/s is shown in figure 6 Vs tip clearance. It shows that the effectiveness is maximum at no gap. The minimum is reached between 0.1 to 0.2 mm and then the effectiveness is again increased steeply up to 0.5mm. From 0.5 to 1mm the variation in effectiveness is very small. As it is practically impossible to have a zero gap for unshrouded impellers so it is concluded that the optimum thickness is 0.5 mm onwards in terms of diffuser effectiveness.

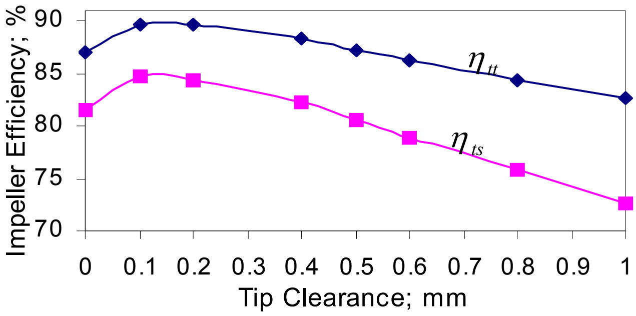

Fig. 5 Impeller η Vs tip clearance at 0.35 kg/s

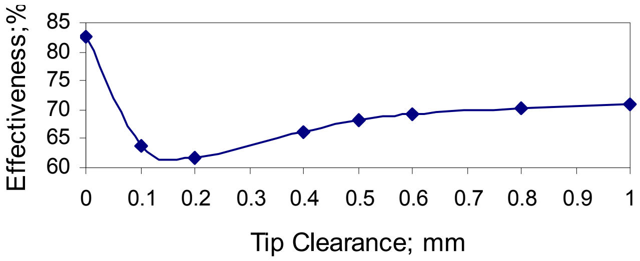

Fig. 6 Diffuser Effectiveness Vs tip clearance at 0.35 kg/s

4.2 Mass Averaged Static and Total Pressure

Mass averaged static and total pressures from inlet to the outlet of diffuser at flow rate of 0.35 kg/s are shown in figure 7 for tip clearance from 0 to 1mm.

Total pressure is constant up to the leading edge of full blade for all clearance levels. It increases continuously with meridional distance up to the blade tip because of the impeller rotation. At the outlet of impeller, the total pressure for 0.1 and 0.2 mm clearance levels is higher resulting in higher impeller efficiency compared to zero gap. Drop in total pressure is found as the fluid moves from inlet to outlet of the diffuser primarily because of the viscous effects. Effect of tip leakage flow is evident from this figure. Total pressure curves are becoming steeper with the increase in tip clearance.

Drop in static pressure is found at the inlet because of the suction effect. Energy transformation from impeller to the fluid increases the pressure up to the blade tip. Static pressure is further increased in the diffuser due to the transformation of kinetic energy.

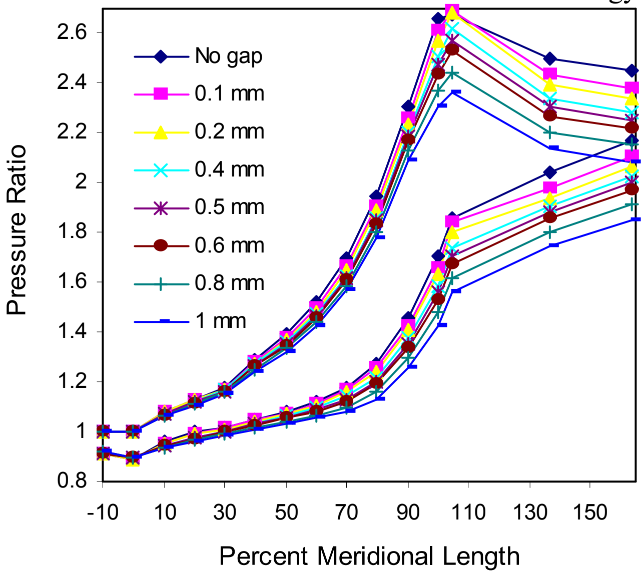

Fig. 7 Mass averaged static & total pressure ratios from inlet to outlet of stage at 0.35 kg/s

4.3 Entropy

Mass averaged entropy distribution from inlet to the outlet of diffuser at flow rate of 0.35 kg/s is shown in figure 8 for tip clearance from 0 to 1mm. The entropy is increasing steadily within the impeller for zero gap due to friction and mixing linked with the dissipation of relative kinetic energy and shear work at the shroud. Relatively higher entropy is observed for all the other clearance levels because of the additional loss of mixing due to clearance flow. Slop of the entropy curves is increasing with the increase in tip clearance. At the outlet of impeller, entropy for 0.1 to 0.5 mm clearances is lower than zero gap resulting in higher impeller efficiency and total pressure.

Figure 9 shows the pitch averaged flow field of various clearance levels at mass flow rate of 0.35 kg/s. Maximum entropy concentration is found at the exit of zero gap impeller. The strips of high entropy Regions are spreading with the increase in tip clearance.

Figure 10 shows the entropy field at the exit of impeller (radius ratio 1.05) at four clearance levels at 0.35 kg/s. Higher entropy regions generated by full and splitter blade trailing edges seems to diminish with the increase in radius ratio. Maximum entropy is found in no gap case near shroud. Rest all levels other than zero gap shows similar behavior which is also evident from mass averaged entropy generation shown in figure 8. Almost similar results were predicted by Larosiliere et al.[9].

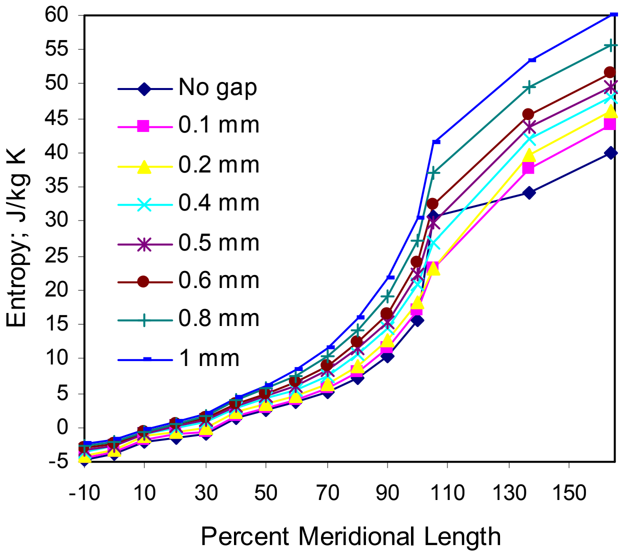

Fig. 8 Mass averaged entropy generation from inlet to outlet of stage at 0.35 kg/s

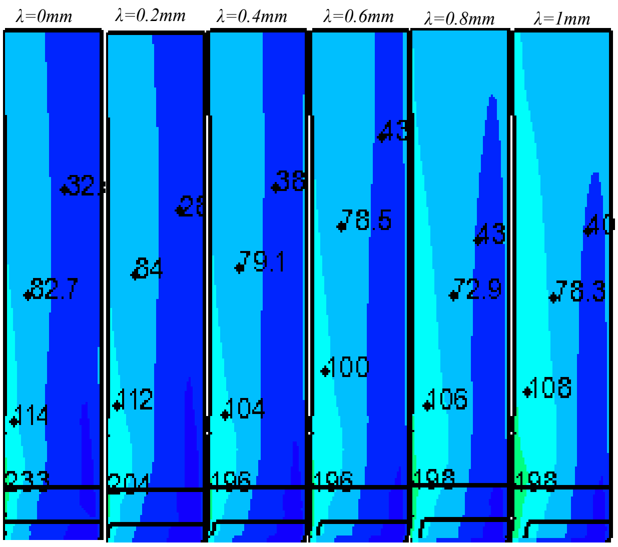

Fig. 9 Pitch averaged entropy field in diffuser at 0.35 kg/s

4.4 Relative Total Pressure at Exit

Normalized relative pressure distribution at impeller exit (radius ratio 1.05) is shown in figure 11. The figure verifies the formation of jet-wake structures. Blade wakes downstream the full and splitter blades are clearly observed in all clearances. The contour shows asymmetric behavior of the flow for the two different channels between full and splitter blades. Within the left channel, the passage vortex between the two blades is found to occupy the central part near shroud. The passage vortex within the right channel is located near the full blade suction side shroud which is found to diffuse down towards hub with the increase in tip clearance. The distribution is found to be analogous to the experimental study of Hong et al. [6].

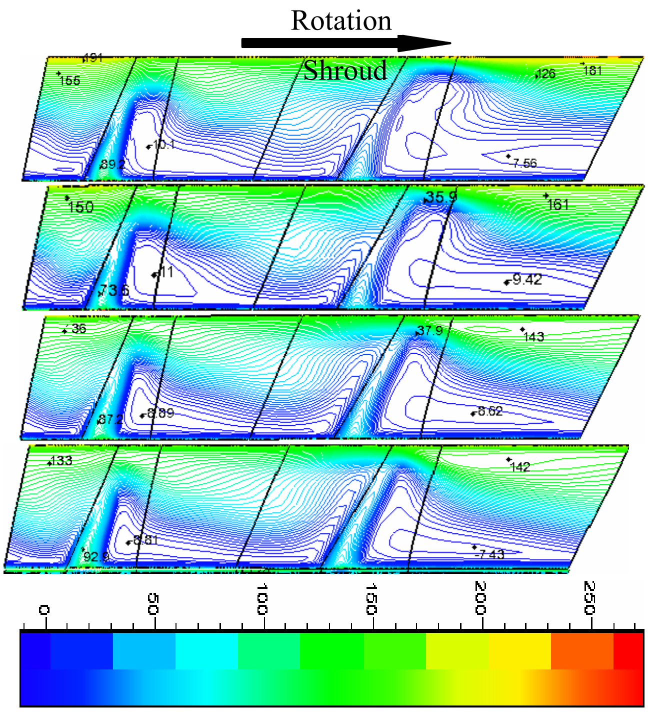

Fig. 10 Entropy field at impeller exit for 0, 0.1, 0.4, 0.6 mm clearance levels respectively at 0.35 kg/s

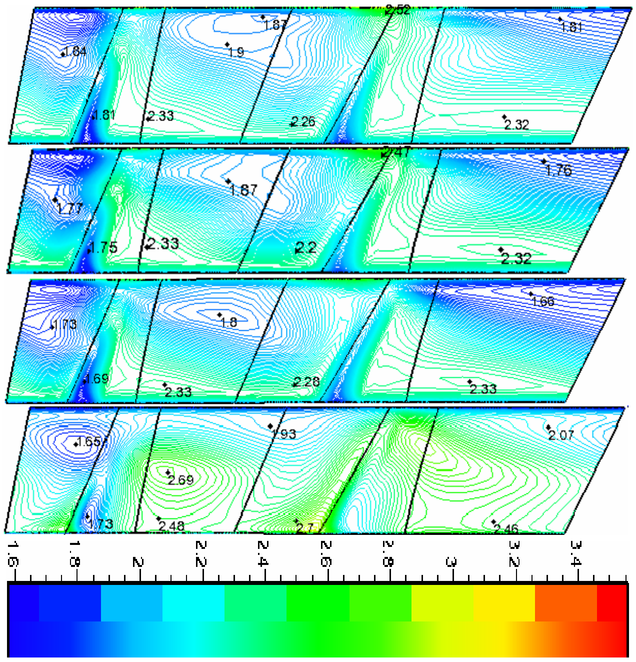

Fig. 11 Normalized relative total pressure field at impeller exit for 0, 0.1, 0.4, 0.6 mm clearance levels respectively at 0.35 kg/s

4.5 Blade Loading

Blade pressure loading at the tip of full blade for 0, 0.1, 0.4, 0.6 mm clearance levels respectively at 0.35 kg/s is shown in figure 12. The data is normalized from 1 to 0 arc length for the pressure side and 0 to 1 fraction of arc length for suction side. For zero gap, almost uniform loading is observed. A wide difference of pressure is found between zero gap and 0.1 mm clearance. Deterioration of tip static pressure with the increase in tip clearance is clearly evident from the figure.

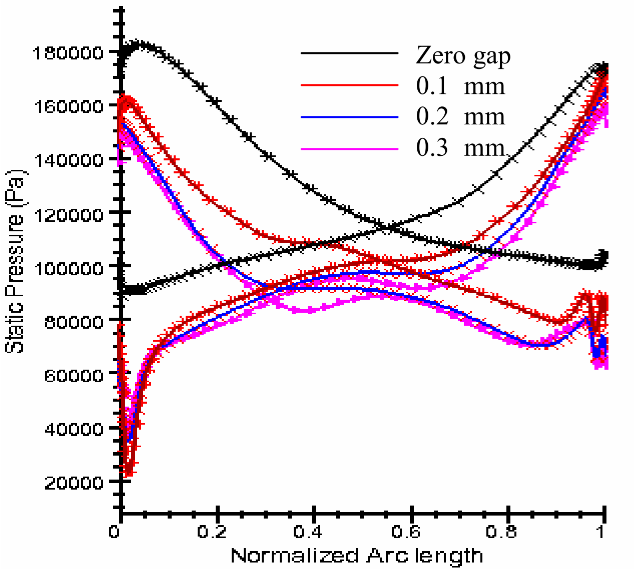

Fig. 12 Blade pressure at tip of full blade for 0, 0.1, 0.4, 0.6 mm clearances respectively at 0.35 kg/s

5 Conclusions

This paper describes a numerical study to investigate the effect of tip clearance on the performance and flow characteristics of a centrifugal impeller for turbocharger application. The stage was split into two sections i.e. rotating impeller and stationary diffuser. Overall performance maps of the stage along with separate impeller and diffuser characteristic curves are presented. Nine tip clearance levels, namely zero, 0.1mm, 0.2mm, 0.3mm, 0.4mm, 0.5mm, 0.6mm, 0.8mm and 1mm, have been investigated and the following results are obtained:

1. At mass flow rate of 0.35 kg/s, a loss of about 15 percent in peak pressure rise of the stage is observed as the tip clearance is increased from zero to 1mm. i.e. a linear assumption specifies a drop of peak pressure ratio by 0.036 for every 0.1mm increase in tip clearance. The peak efficiency has been reduced by 10 percent for the same case
2. From the stage perspective, no optimum clearance, other than zero tip clearance, has been found for the analyzed compressor.
3. The impeller efficiency is found to be maximum at tip clearance between 0.1 to 0.2 mm however minimum diffuser effectives is found at same clearance level.
4. The optimum thickness is 0.5 mm onwards in terms of diffuser effectiveness
5. A model $\frac{dP}{d\lambda} \cong -0.295$ found to fit the stage pressure ratio for all mass flow rates between 0.1 to 1 mm clearances.

Mass averaged flow parameters are presented showing the flow phenomena within the impeller and shroud. Loss characteristics are further investigated with the various plots of entropy. Relative total pressure is presented at the exit of impeller which is verifying the classical jet wake theory.

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