Study of Tip Leakage Flows in Tandem Bladed Low Speed Axial Flow Compressor

by

Rohan Rajmohan Gundu

(18AE30015)

Under the guidance of

Dr. C.S. Mistry

DEPARTMENT OF AEROSPACE ENGINEERING INDIAN INSTITUTE OF TECHNOLOGY, KHARAGPUR

ABSTRACT

Tandem rotors provide a high pressure rise with a minimum number of stages compared to a conventional rotor[1]. The application of tandem blading is only justified by prescribing high aerodynamic loading on both blades which increases the concern for associated tip leakage and corner separation losses. The study reveals the interaction of tip leakage flow from the fore and aft blade with the end wall flow and its influence on the main stream flow. The presence of an aft blade behind the fore seeks additional caution for optimum management of tip leakage flow. However, including Tandem Rotors induce unwanted three dimensionality in the flow physics of the compressor. These emerge mainly at the boundaries of the rotor blades and the compressor casing. These three dimensionalities arise mainly due to the tip leakage flows, formation of tip vortices, endwall flows, momentum loss in the flow, etc. The fundamental discussion in this report is about the Tip Leakage Flows and the effects they have on the axial flow of the compressor. The leakage flows from the individual blades incorporate with the primary flow of the compressor as well as with the flows coming from the shroud boundary layers resulting in momentum deficit fluid at the exit of the rotor blade and the blade passage region. As far as the blade parameters are concerned, an axial overlap of 3%, a Flow coefficient of 0.9 and a Blade loading of 1.04 was kept constant throughout the study to analyze the flow and understand the flow physics. A detailed computational study was performed on the ANSYS 2019 R3 software to study the impact of tip leakage flows.

ACKNOWLEDGEMENTS

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CONTENTS

NOMENCLATURE

The following nomenclature has been used in the report for the representation of various physical properties as mentioned:

C	: Chord (unit: mm)	a	: Axial spacing (unit: mm)
s	: Pitch (unit: mm)	t	: Pitch wise spacing (unit: mm)
FB	: Fore Blade	AB	: Aft Blade
PP	: Percent pitch	AO	: Axial overlap
$\bar{\xi}$	: Abs vorticity vector	θ	: Camber (unit: degrees)
γ	: Stagger (unit: degrees)	β	: Flow angles (unit: degrees)
$\Delta\beta$	: Deflection (unit: degrees)	ρ	: Density (unit: kg/m^3)
P_{ref}	: Ref. static pressure (unit: Pa)	P_{local}	: Local static pressure (unit: Pa)
P_0	: Total Pressure (unit: Pa)	DF	: Diffusion Factor
	: Total pressure loss coefficient	C_p	: Coefficient of pressure
DOR	: Degree of reaction	LS	: Load split
C_a	: Axial Velocity (unit: m/s)	σ	: Solidity
ψ	: Stage Loading Coefficient	φ	: Flow Coefficient
PS	: Pressure Side of Blade Airfoil	SS	: Suction Side of Blade Airfoil
$\bar{\omega}$	: Velocity vector		

Suffix

1	: Fore blade	2	: Aft blade
11	: Fore blade LE	22	: Aft blade TE
T	: Tandem	m	: Mean span

INTRODUCTION

Modern day world has a constraint of fuel and the airline industry is moving towards enhanced fuel economy solutions. The fuel economy is directly related with the thrust produced and weight of the entire aircraft. The fuel efficiency will increase if (i) The thrust produced is increased for the same amount of engine weight or (ii) If the same amount of thrust is produced for reduced weight of the entire aircraft. For either of the cases, the net thrust produced per weight of the aircraft should increase. This can be achieved by reducing the weight of the aircraft by not compromising on the thrust produced. The weight of the aircraft includes a major portion of the engine of the aircraft. With hundreds of metallic components used for a single stage, most of the weight of the engine is occupied by the compressor. The compressors basically inject energy into the fluid by rotating at high speeds using airfoil blades. This is directly related to the thrust produced by the engine. Thus, to produce higher thrust at low weights and relatively compact size engines, a promising prospect is to reduce the overall engine size is to reduce the number of stages in axial compressors for a given pressure rise. Conventional axial flow compressors suffer from the risk of flow separation and stall upon higher aerodynamic loading which limits their overall size for a given overall pressure rise. It becomes obligatory to seek alternative design strategy for axial compressors which fulfills the requirement of high stage loading with least aerodynamic losses.

Tandem bladed rotors provide a commendable solution to the aforesaid requirement. A huge amount of work has been gone into developing the domain of Tandem Bladed Axial Compressors. A Tandem Bladed Rotor consists of two blades arranged one after the other on a single rotor working in unison. The elemental geometry for a Tandem Bladed Rotor is shown below in **figure 1** The aerodynamic advantage offered by the tandem configuration is based on two effects. The first being increase of circulation (blade loading) around the fore blade due to the downstream presence of the aft blade. The second effect being the re-energization of the suction surface boundary layer achieved by the nozzle flow emerging from the inter-blade gap [**Fig. 2**]. These phenomena together make the tandem bladed configuration more efficient enabling a higher

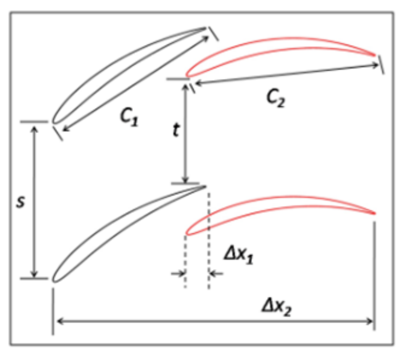


Figure 1: Tandem Bladed rotor geometry [2]

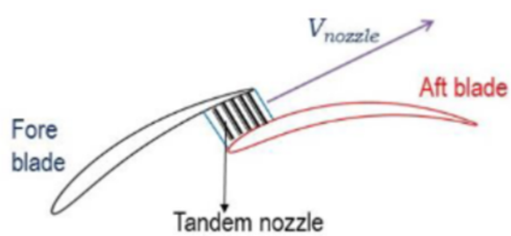


Figure 2 : Tandem Nozzle Flow

fluid deflection than the conventional rotor blade system. The application of tandem blading concept in compressor design was demonstrated in the report by Spraglin[3] in the early 50's. The earliest prototype developed with a tandem bladed rotor was designed and tested by Sheets[4]. The author reported a rotor efficiency of 94% with permitted losses which was considered remarkable during that time. Sangers[5] conducted an exhaustive study on various parameters of tandem configuration and claimed promising potential for future development of tandem cascades.

Tip leakage losses are flows flowing from the PS of the airfoil to the SS owing to a small pressure difference. The reason for this pressure difference is the shape of the airfoil. The flow escapes to the SS from a tiny but decisive gap existing between the blade tip and the shroud casing of the compressor called the Tip Gap / Tip Clearance. These flows mix with the primary flow to create vortices thus resulting in a low momentum region. Tip leakage losses and end wall flow losses contribute to a significant fraction of total aerodynamic losses associated with axial flow compressors. The efficiency of an axial flow compressor can be severely compromised due to interaction of unmanaged tip leakage flow with the main flow. The adverse effects of large tip clearance on the efficiency and stability of axial compressors has been investigated by Smith[6] . The passage flow blockage strongly depends on tip clearance flow due to increase in displacement thickness across the blade rows with high tip clearance. The mixed tip leakage flow and boundary layer flow results in a pocket of momentum deficit fluid between rotor and stator passage near the casing region as shown in **figure 3**. Very negligible loss of momentum in the gap region itself but mixing of the clearance flow with core flow is responsible for losses This report describes, studies and compares the effect of leakage losses into the primary flow for two cases of tip gaps (i) 1mm Tip Clearance and (ii) 2mm Tip Clearance.

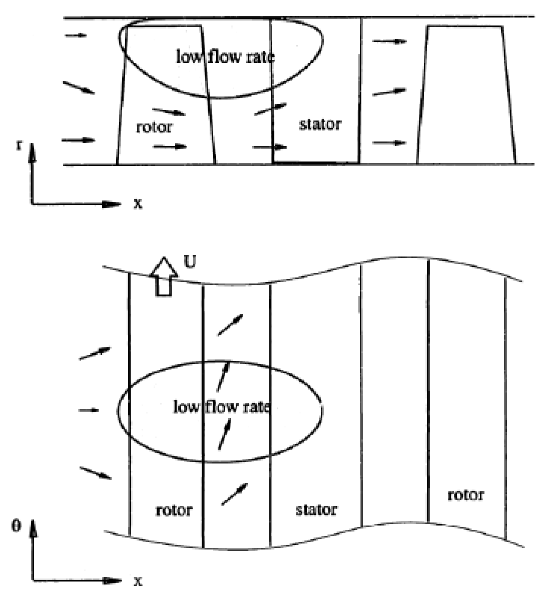


Figure 3: Effect of flow Blockage [7] [8]

With emphasis on realization of tandem bladed concept for commercial application we have put forward the purpose of studying the effects of tip leakage losses in tandem domain configuration. The design and implementation of tandem bladed concept requires detailed understanding of flow physics as it is more complex in presence of two rotating blades along with stator. The justification for increased structural complexity is liable to meet halfway with higher end wall losses. The flow behavior near the tip region plays an important role in terms of overall pressure rise. A comprehensive literature for tip leakage flow management exists for conventional axial flow compressors. A far reaching writing for tip leakage flow streams the executives exists for regular pivotal stream blowers. Not much accessible literature examines the significant perspectives about the choice of hub dividing between two blades undoubtedly. The absence of accessibility of point by point understanding roused us to complete orderly review to understand the flow physics of the tip leakage flows by changing the tip clearance between the blade tip and the shroud and by keeping the AO and PP to be constant at 3% and 82 % respectively. The 3% AO for the rotor blade is kept at the tip of the blade and it gradually decreases to 0% near the hub region.

DESIGN OF BASELINE TANDEM BLADED ROTOR

The following parameters are used for design and analysis of the base stage.

$$\varphi = \frac{C_a}{U} \qquad (1)$$

$$\psi = \frac{P_{0,22} - P_{0,11}}{\rho C_a^2} \qquad (2)$$

$$C_p = \frac{P_{local} - P_{ref}}{\frac{1}{2}\rho C_a^2} \qquad (3)$$

The tandem blade is designed for a pressure rise coefficient of 1.04 and a flow coefficient of 0.9 . Design parameters for the fore blade, aft blade and the effective tandem blade are summarized in Table 1, 2 and 3. Diffusion Factor for the individual blades is calculated by substituting the respective blade angles in the equation (4) whereas the formulae for Axial Overlap (AO), effective chord (C_{eff}), Load Split (LS) and Percent Pitch (PP) are represented by the equations (5) , (6) , (7) and (8). The aerodynamic design consists of a controlled spanwise loading distribution targeted to achieve the maximum possible pressure rise coefficient

$$DF = 1 - \frac{cos(\beta_{11})}{cos(\beta_{22})} + \frac{cos(\beta_{11}) * (tan(\beta_{11}) - tan(\beta_{22}))}{2\sigma_{eff}} \quad \text{.....(4)}$$

$$AO = \frac{\Delta x_1}{\Delta x_2} \quad \text{.....(5)}$$

$$C_{eff} = \frac{C_1 + C_2}{1 + AO} \quad \text{.....(6)}$$

$$LS = \frac{DF_1}{DF_1 + DF_2} \quad \text{.....(7)}$$

$$PP = \frac{t}{s} \quad \text{.....(8)}$$

Parameters	Hub	Mean	Tip
ψ_1	0.38	0.6	0.74
φ_1	1.35	0.9	0.72
σ_1	1.19	0.8	0.6
DF_1	0.39	0.53	0.57

Table 1 : Design parameters of fore blade for base case

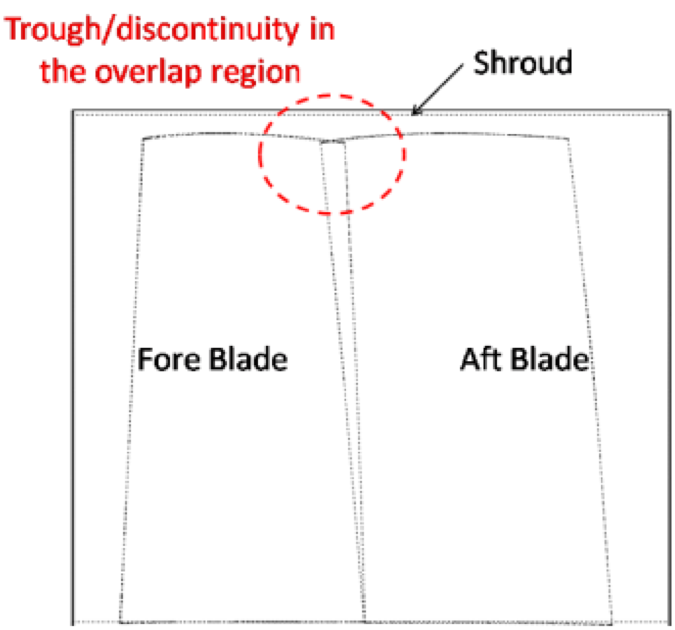
Parameters	Hub	Mean	Tip
ψ_2	0.17	0.45	0.72
φ_2	1.35	0.9	0.72
σ_2	1.19	0.8	0.6
DF_2	0.11	0.37	0.56

Table 2: Design parameters of aft blade for base case

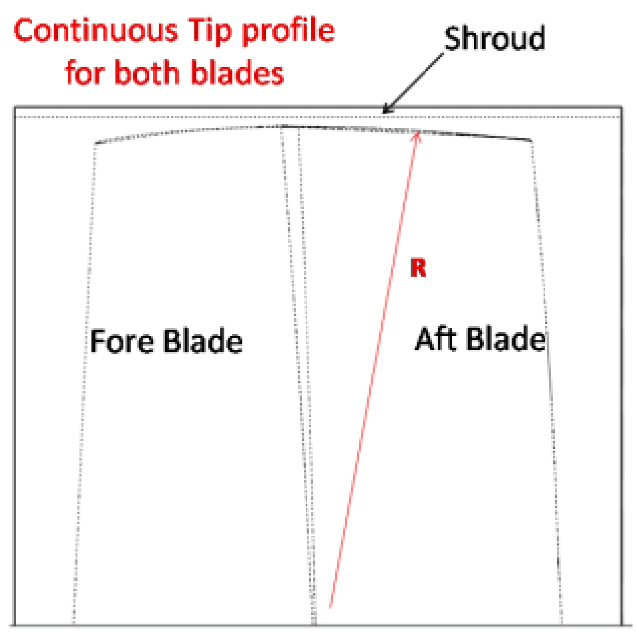
Parameters	Hub	Mean	Tip
σ_T	2.24	1.50	1.12
DF_T	0.34	0.58	0.73

Table 3: Design parameters of Tandem blade for base case

Conventional design methodology of projecting each blade on an independent cylindrical surface used for single rotor design cannot be used for designing the tandem rotor blade. The tip profile for the tandem blade configuration was designed by projecting 3-D airfoils for both the fore blade as well as the aft blade on a single cylindrical surface [**Fig 4b**]. The reason for doing such is to provide a constant tip gap between the rotor blade and the shroud casing. While using the two cylindrical surface approach, a trough / discontinuity (as highlighted in **figure 4a**) appears. Many unwanted phenomena such as forceful three dimensionality of the flow occur near the trough / discontinuity resulting in very complex flow field structure along with growth of the wall boundary layer which hampers the overall performance of the rotor as well as stage.



(a) Discontinuos tip profile



(b) Continuous tp profile

Figure 4 : Tip profiles for Tandem Blade Rotor [2]

COMPUTATIONAL DOMAIN

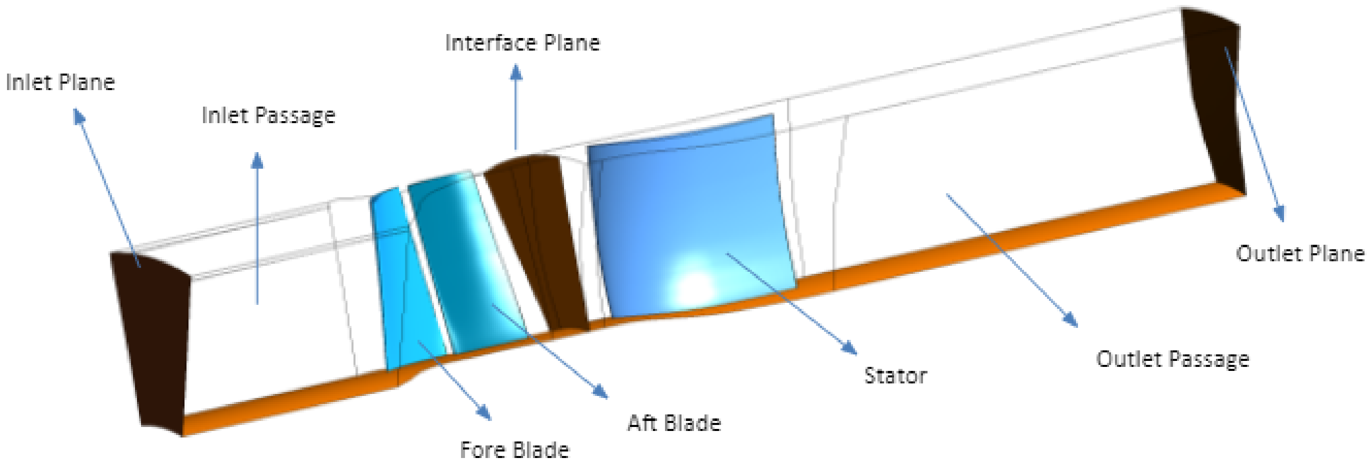


Figure 5: Computational Domain for the Simulations

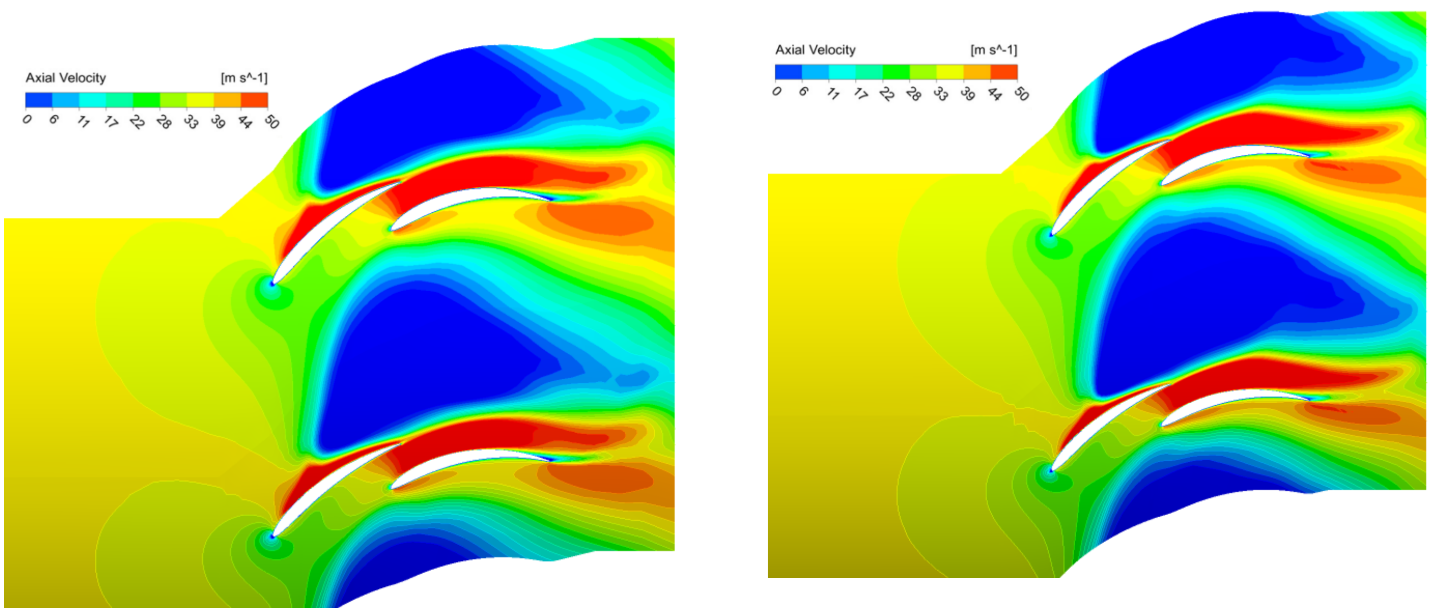
The flow domain has four distinct parts - a stationary inlet domain, a rotational tandem rotor domain (consisting of the fore blade and the aft blade), the stator and at the end, the stator exit domain [**Fig. 5**]. The inlet plane is at 1-rotor chord upstream of rotor and exit plane is 1.5-stator chord downstream of stator[2]. The stage is designed for a swirl free inlet so a uniform pressure profile is imposed on the inlet boundary. The rotating and stationary domains are connected by an interface plane. The present study is focused on the behavior of tip leakage flow with change in axial overlap. The interface plane should positively keep the relative positions of the flow domain fixed to capture the tip leakage flow with acceptable accuracy. The 'frozen rotor' interface is selected to connect the two domains as it captures the wake more efficiently. The commercially available solver ANSYS CFX® is used for solving the steady state Reynolds Averaged Navier Stokes equations (RANS). The suitability of CFX solver for studying the performance and flow physics of compressors has been demonstrated in the detailed study by Belamri et al.[7]. The two equation shear stress transport (SST) model is used for modeling turbulence to efficiently capture the vortices in near surface flows in adverse pressure gradient regime. The computational domain is discretized using ANSYS TurboGrid®. A multi-block structured grid with hexagonal elements has been implemented for computation. The objective of the study was to investigate the effect of tip leakage losses on the primary flow for changes in the Tip Clearance between the rotor tip and the shroud.

RESULTS and DISCUSSION

A detailed computational study to understand the effect of tip leakage flows on the primary flow of the low speed axial flow compressor was carried out for two sets of tip clearance values (1mm and 2mm). The tip leakage flows on interaction with the shroud boundary layer and the primary flow forms a low momentum bubble kind of region that progresses as it goes ahead through the axial direction. Unlike conventional rotors where this bubble keeps on growing in size, it mixes with the high velocity flow from the tandem nozzle region forming vortices and counter vortices until the flow reaches a stable state at the exit of the stator. To let this happen, the stator needs to be of a good diffusive nature.

Axial Velocity Contours: -

The axial velocity for an axial compressor is the one which enters the compressor along the axial (length of the compressor) and exits from the same direction. The axial velocity for all the three combinations decreases as the flow reaches the leading edge of the aft blade. As evident from **figures 6a** and **6b** , the small region between the leading edge of the at blade and the trailing edge of the fore blade acts like a nozzle as we see the flow accelerates through that region.



(a) 1mm tip clearance

(b) 2mm tip clearance

Figure 6 : Axial Velocity Profile for blades at 96% span location

The accelerated flow prevents flow separation at the *SS* of the at blade. If we focus on the area just above the *SS* of the fore blade, we see a relatively high velocity region for both the blades. However, the region seems to be more intense for the 1mm gap region. This is due to the effect of tip leakage flow. For low tip clearance blades, the tip leakage flow is less compared to the high tip gap blades. This is the reason why we see a relatively larger momentum deficient region for the 2mm tip clearance blade at 96% span location.

Blade Loading : -

One of the most important design parameters for the blades is the blade loading. It is the non-dimensional measure of the pressure at a certain location. Looking at the blade loading curves, one can easily identify the flow direction and can get a rough idea about the flow physics of the required region. Blade loading curve plots the c_p [Eqn (3)] distribution of the blade curves. This report analyzes and studies the blade loading distribution of the fore blade at 90% and 96% span locations of the blade. At 90% span location, the loading seems to be coming as standard for both the blades. However the difference is visible at 96% span location. We see a dip in the blade loading value at around 10% chord for the suction surface for both the blades. Ater detailed investigation a conclusion was reached wherein the reason for the dip in the curve was found out to be the formation of tip leakage vortices. These leakage vortices were generated by the interaction of Tip Leakage Flow, the Shroud Boundary Layer and the Primary flow of the compressor. The strength of these vortices are evident from the blade loading curves of both the blades. The extent of dip in the blade loading curve justifies the strength of the leakage vortices.

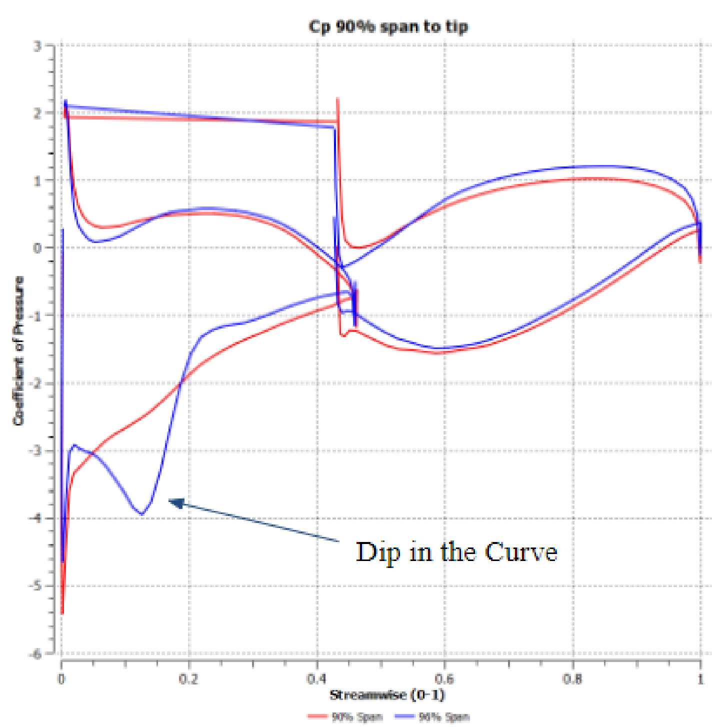


Figure 7: Blade Loading Curve for 1mm tip gap blade

On comparing the two blading curves, we can observe that the trough of the dip in the loading curve for the 1mm blade is at a higher location compared to the same for 2mm blade. This leads us to the conclusion that there is a higher pressure loss for the high tip clearance blade as compared to the low tip clearance blade. This in turn implies that the leakage vortices are stronger for the 2mm tip gap blades than they are for the 1mm tip gap blades. The leakage flows for the 2mm tip gap blades are higher in magnitude than those for 1mm tip gap blades. The location of the tip leakage vortices however is the same for both the cases as evident from the **figures 7** and **8**. The blade loading curve for the rest of the blades displays no such anomalous behaviour giving results as per expectation got from the literature review.

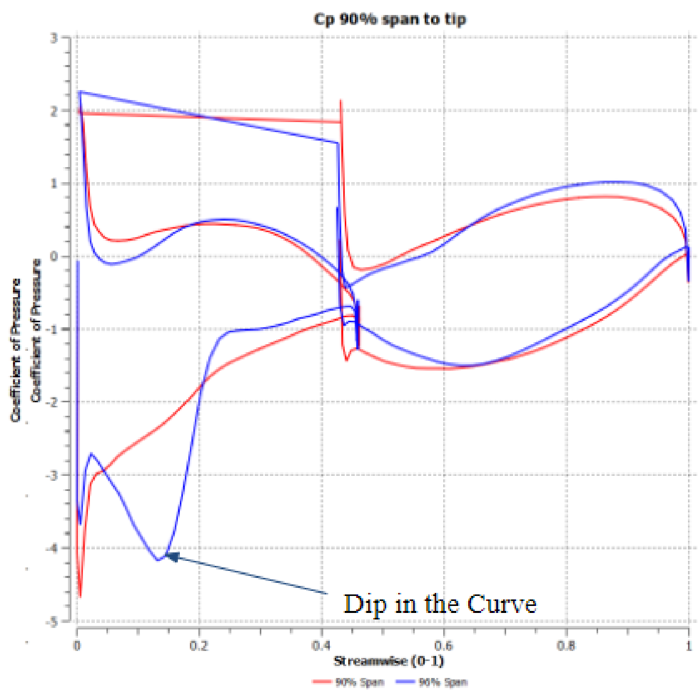
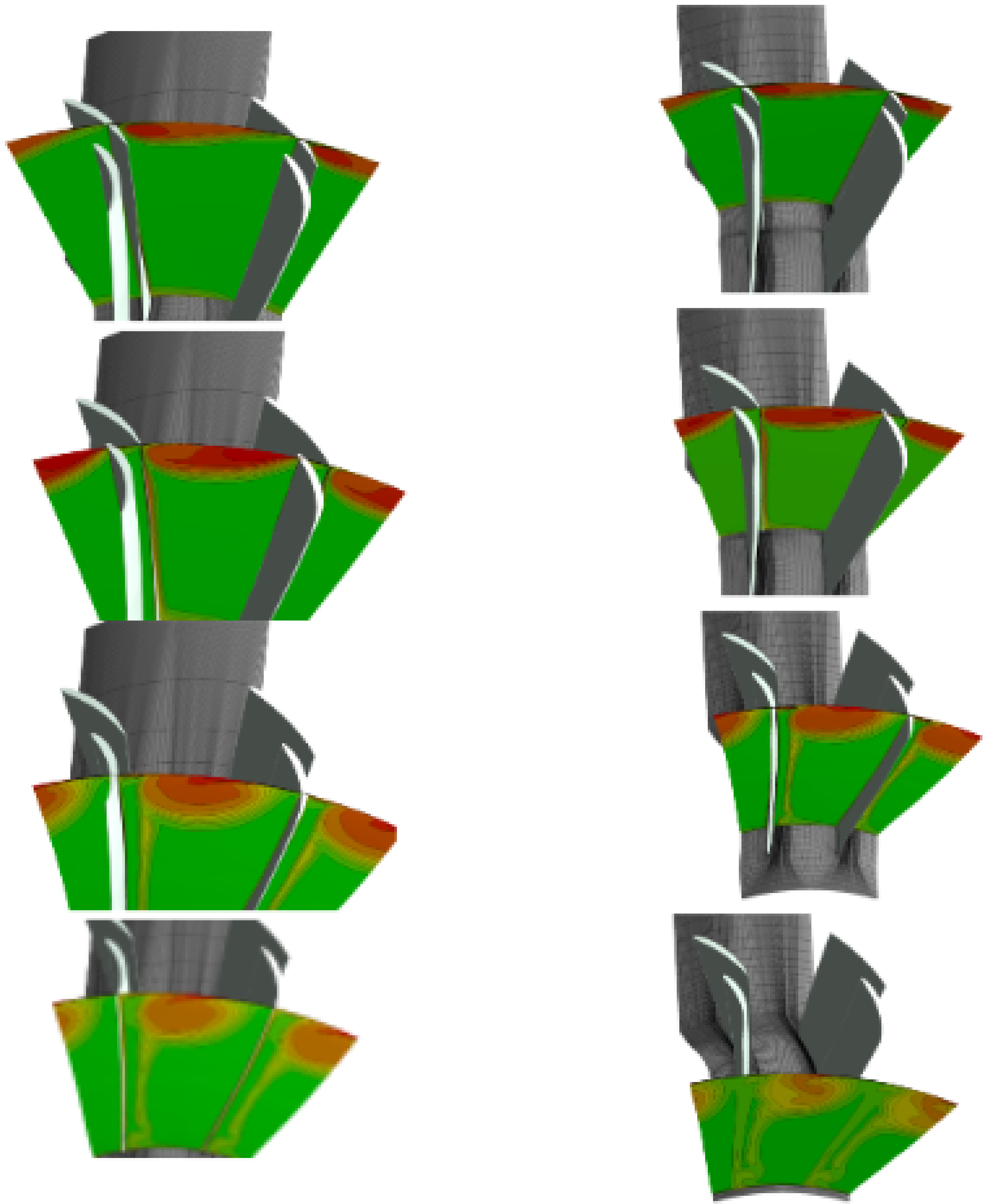


Figure 8: Blade Loading Curve for 2mm tip gap blade

Entropy and its effects at the Rotor Exit Flow Properties: -

The presence of two highly loaded blades results in dual wake structure formation at the rotor exit. The resulting wake structure at the rotor exit is a consequence of interaction of evolving wakes with the shroud boundary layer and tandem nozzle flow through the blade passage. The twin wake system from the rotor is a unique flow feature of tandem configuration and requires more attention during the design stage for the rotor as well as for the downstream stator. The above-mentioned flow variables can be best understood by following the entropy growth in the blade passage. **Figures 9a** and **9b** show the entropy progression from the leading edge of the fore blade to the trailing edge of the aft blade and finally at the exit of the rotor. The entropy at the exit of the rotor exhibits some interesting phenomena. As a result of rotor wakes and endwall irreversibility, we see a non-uniform contour for entropy too. The flow seems to have a higher irreversibility for the 1mm tip gap blades as compared to the 2mm tip gap blades. As a consequence of higher aerodynamic load split a well-defined sharp wake is seen to exist at the TE of the fore blade for all configurations. The result of interaction of tip-leakage flow from the fore blade and the end wall region is also observed as a region of higher entropy with increased gradient towards the end wall. It can be observed that entropy generation in the endwall region increases with positive overlap. This is attributed due to increase in fore blade loading and it is resulting in stronger tip leakage flow. This complex system

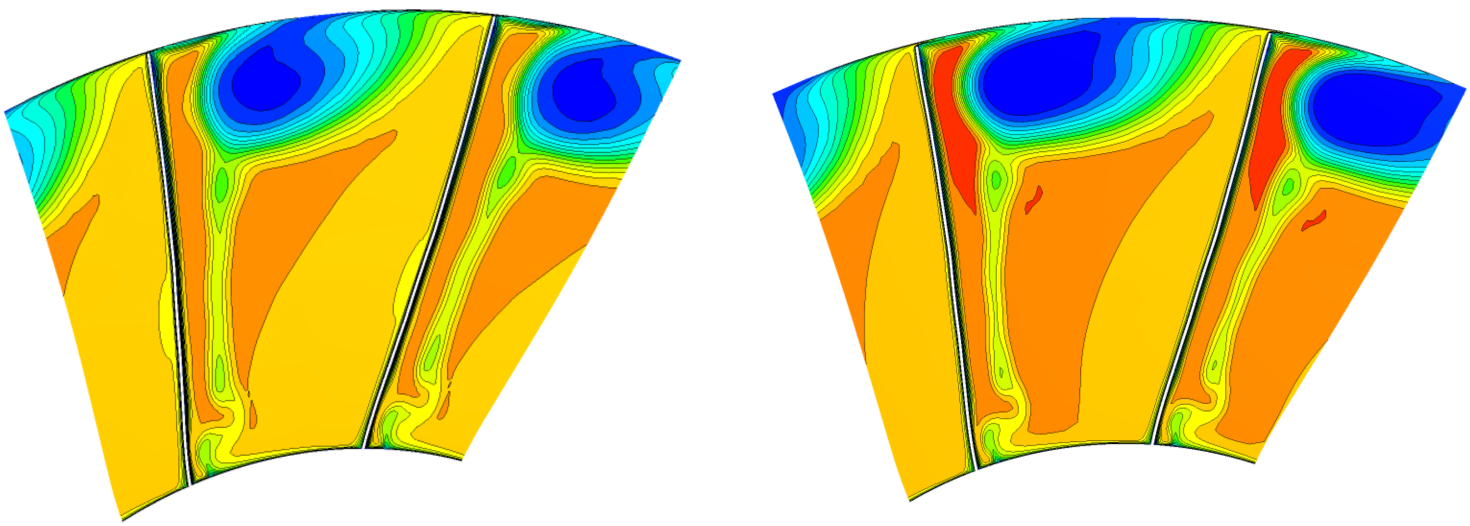


(a) 1mm tip gap blades

(b) 2mm tip gap blades

Figure 9: Entropy (Non-Dimensional) Contour varying from FB Leading edge to AB Trailing edge

of irreversibility moves downstream with increase in both spanwise and circumferential direction. At the TE of aft blade, the contribution of aft blade loading is clearly evident with formation of aft blade wake coming along with endwall loss. It can be observed that tip leakage flow structure from the aft blade merges well with the tandem nozzle flow and the identification of individual contribution of losses becomes most difficult at this stage. Interestingly at the rotor exit, both the wakes and endwall irreversibility haven't merged and can be distinguished. This results in a non-uniform velocity flow field at the rotor exit plane [**Fig 10**].



(a) 1mm tip gap blades

(b) 2mm tip gap blades

Figure 10: Rotor Exit Axial Velocity Contours

The low momentum bubble at the rotor exit changes its spanwise as well as circumferential extent with the change of tip clearance and consequently the change in tip leakage velocities. This is the same as the TE stall of the aft blade itself. The momentum bubble gets detached from the end wall as well from the aft blade wake. The axial velocity contour exhibits a certain low momentum region near the suction surface of the blade. This region was created due to the vortex formation occurring at the leading edge of the fore airfoil owing to the mixing of the tip leakage flow with the primary flow. The region by observation is less intense for low tip gap blades. However there is a considerable difference of magnitude beside the low momentum region near to the suction surface.

Blade Passage Vortices:-

Blade passage vortices are formed by the interaction of leakage flow and the endwall boundary layer. It is the strength and structure of the vortex at the rotor exit which determines the aerodynamic complexities of the downstream stator. The vortex structure in this report is discussed as 'normalized helicity' as proposed by Inoue et al.[9] in the detailed discussion of tip leakage vortex and breakdown. The normalized helicity is defined as :-

$$\xi_s = \frac{\bar{\xi} \cdot \bar{\omega}}{|\xi||\omega|} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(9)$$

Mathematically it represents the cosine of angle between the absolute vorticity and velocity vector. Thus, the magnitude of normalized helicity tends to unity in the vortex core region. The range of normalized helicity is ±1 and the sign indicates the direction of the swirl of the vortex relative to the streamwise velocity component. The vortex core passes through the maximum value of normalized helicity and streamline passing through the immediate vicinity of maxima are twisted. Three vortex structures are evident in all the cases - (a) Tip leakage vortex of fore blade, (b) Leakage vortex due to blade stall and (c) A counter vortex structure. The vortex due to tip leakage flow from the fore blade is clearly evident and identified as 'V_1'. The primary effect of tandem nozzle flow is to reduce the intensity of 'V_1' as seen from the helicity contour at the TE of the fore blade. As observed from the streamline distribution, the tip-leakage flow from the aft blade combines with nozzle flow. This complex interaction results in formation of a second vortex structure 'V_2'. The delayed formation of 'V_2' is attributed to the tip profile of both the blades. The tip profile of both the blades consists of projecting the tip coordinates on a single cylindrical surface rather than two individual blade tip profiles. This results in decreased tip clearance at the overlap region. The quick intensity gain of 'V_2' results in a counter-vortex structure designated as 'V_C'. The exit plane of the rotor contains two differentiable vortex structures due to the discussed complex interactions. The change of helicity sign indicates a possible breakdown of vortex structure (V_3) and a dedicated study is required in order to identify the exact reason for breakdown. The vortex structure 'V_2' weakens a bit as it proceeds towards the TE and finally exits the blade passage as 'V_3'. **Figures 11** and **12** indicates the blade passage vortex structures for tandem bladed rotors with tip clearance of 1 mm and 2mm

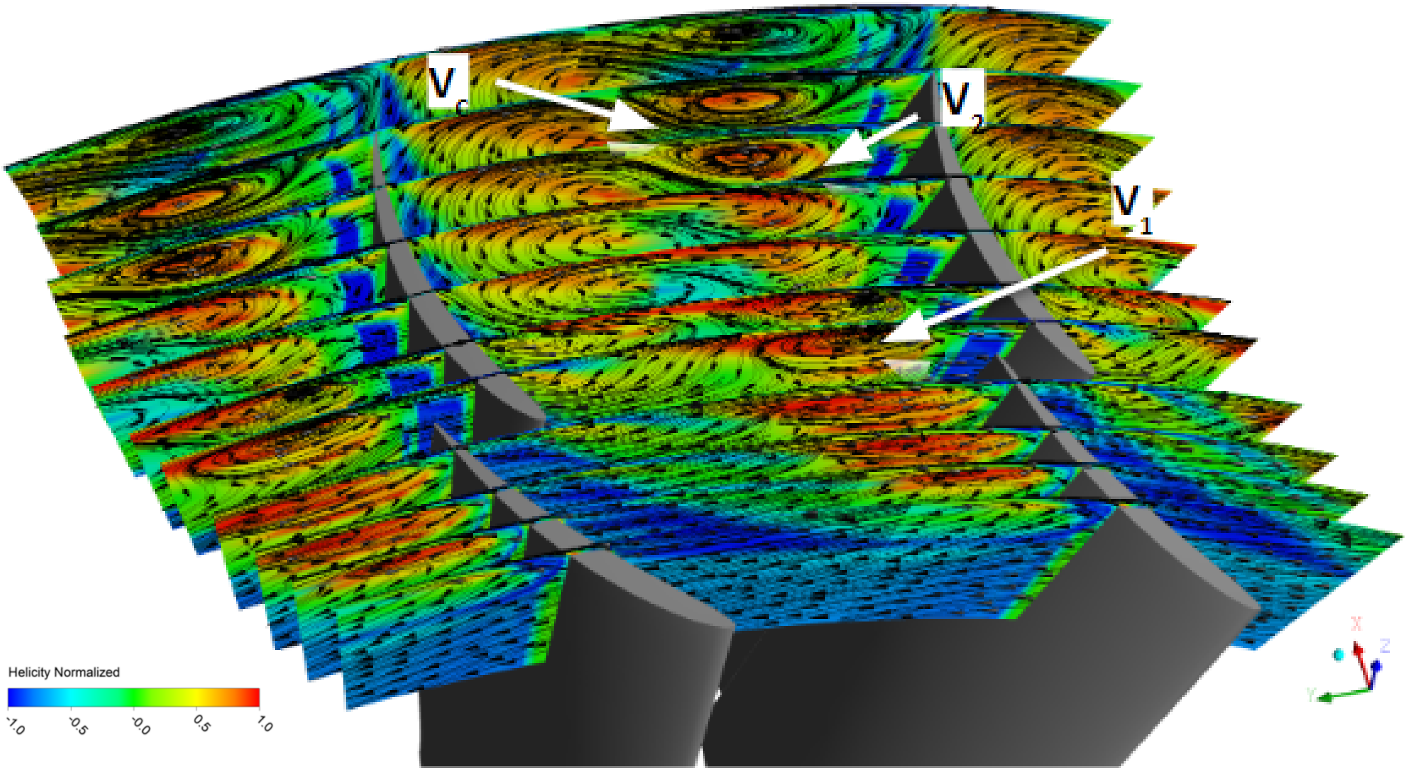


Figure 11: Blade passage Vortex for 1mm tip clearance blade

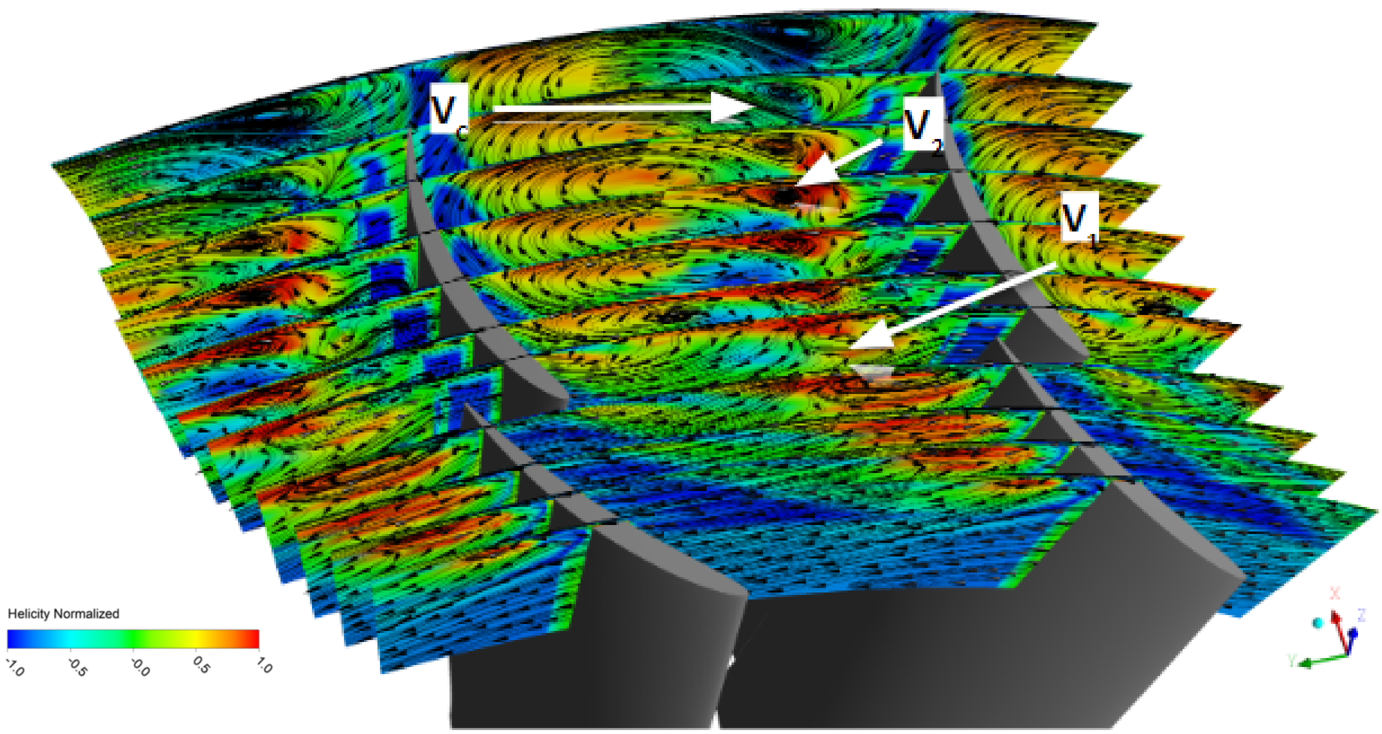


Figure 12: Blade passage Vortex for 2mm tip clearance blade

The vortex structure in **figure 11** of 1mm tip gap is very intense compared to **figure 12** of 2mm tip gap. The vortex structure V_2 having combined with the nozzle flow, the flow rapidly circulates into itself thus forming a helitically high vertex structure. These vortices have an adverse effect on the exit low properties such as axial velocity, entropy, and pressure of the flow. The vortex formation at the tip is one of the major reasons for loss of efficiency of a compressor. The flow at the exit of the rotor is quite complex and three dimensions causing irreversibility issues at the entry of the stator. As discussed earlier, we need a good diffusive stator to extract the work and increase the pressure of the moving / working fluid inside the axial flow compressor.

Stator Exit Velocity:-

This report has considered the application of a DCA Cambered Stator [**Fig.13**[7]] to counter the non-uniformities in the rotor exit flow field. The DCA camberline facilitates different LE and TE metal angles with controlled turning distribution along the chord. Basically, it consists of two circular arcs which can be blended at any desired chord location giving more flexibility to manage the flow. One of the strategies in approaching such a design is to limit the flow diffusion in the stator passage by implementing custom tailored airfoil shapes. The adoption of circular arc camberline for such higher turning may lead to mid-loaded flow configuration. After iterating through a number of other camber line shapes e.g., exponential, parabolic, polynomial etc. it was decided to implement the double arc (DCA) camberline owing to its flexibility of aerodynamic loading control. The stator design parameters considered for design are :-

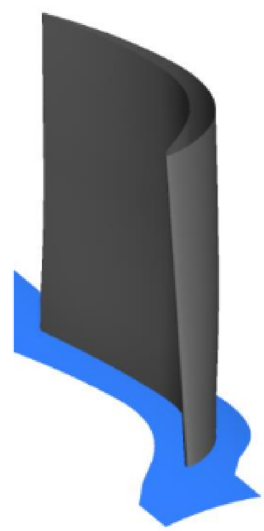
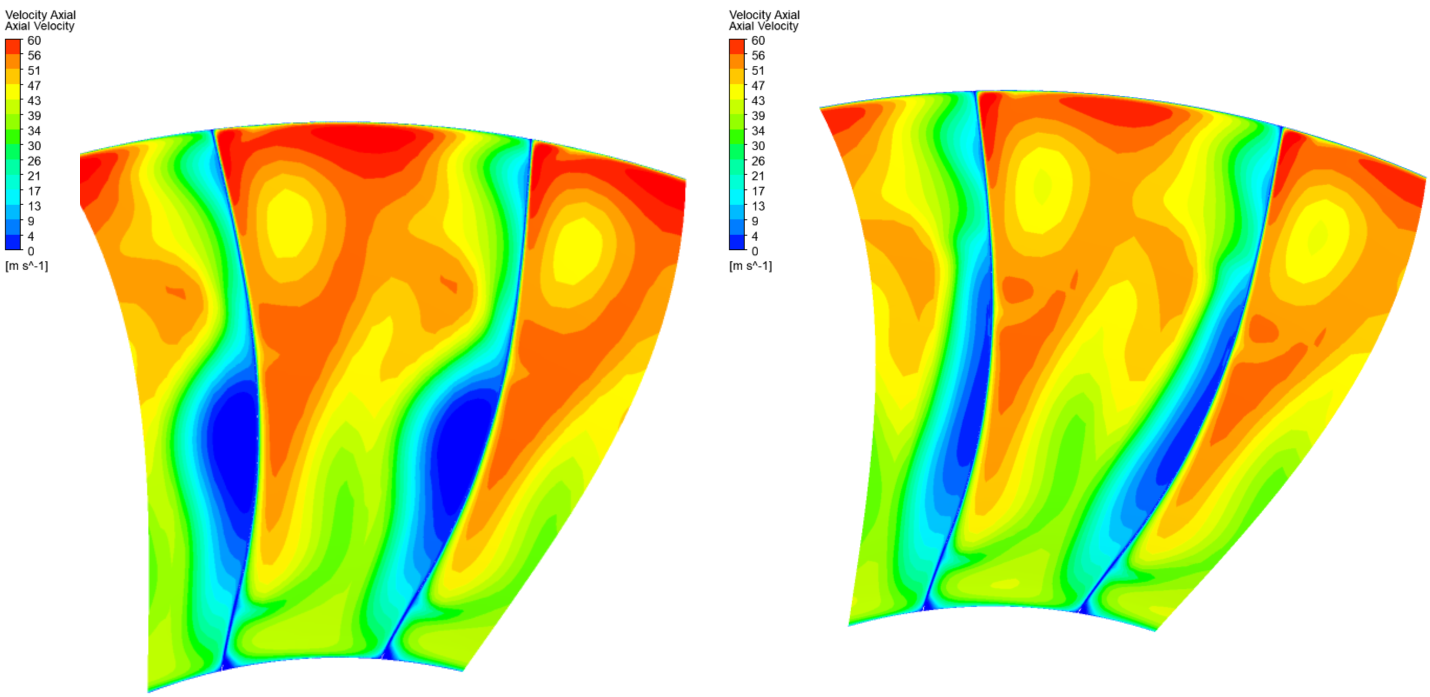


Figure 13 : DCA Stator Model

Parameters	Hub	Mean	Tip
$\Delta\sigma_S$	53.25	59.72	62.58
DF_S	0.53	0.70	0.81

Table 4: Stator Design Properties

The three dimensionality and non-uniformity of the flow field associated with tandem bladed rotors are readily carried on to the down placed stator. For the present stage design, the non-uniform velocity profile at the rotor exit possesses the most challenging inflow condition for the stator. As we look into the exit velocity profile [**Fig. 14(a)** and **Fig.14(b)**] at the stator, we see some interesting phenomena happening.



(a) 1mm tip gap blades

(b) 2mm tip gap blades

Figure 14: Stator Exit Axial Velocity Contour

The most keen observation is the flow separation occurring at the around 40% span location of the stator. The 1mm tip gap blade for stator has an increased flow separation compared to the 2mm tip gap blade for stator. This flow separation is essentially due to extreme forward loading which results in higher blade curvature near the LE. The spanwise migration of the boundary layer is more gradual in case of DCA cambered stator in comparison to conventional. stator. The radial pressure gradient has been controlled to a significant degree especially in the lower-midspan regions. The fore loading is primarily a consequence of changes in pressure distribution on pressure side of airfoil

CONCLUSIONS

The objective of the study was to study the effect of tip leakage flows for changes in the tip gaps / tip clearance of the blades. The flow structure in a fixed Axial Overlap and Percent Pitch condition was analyzed and studied thoroughly for two different tip clearances namely 1mm and 2mm. The conclusions of the study were as follows: -

1. For a change in the tip gap, the tip leakage velocity changes. Decrease in the tip gap causes an increase in performance of the blade thereby causing an increase in the blade loading (as evident in the blade loading chart).
2. The axial velocity at 96% span location has an increased volume of low momentum region for higher tip clearance blades.
3. Entropy gain for the flow is higher in case of the 1mm tip gap blade. Implying higher three dimensionality for the flow there in the rotor exit region.
4. The blade passage vortex structures were quite vigorous and intense for the 2mm tip gap blades owing to the reason that higher amounts of tip leakage losses occur at the fore blade as well as the aft blade compared to the 1mm tip gap blade.
5. The flow separation in the stator exit velocity field is higher in case of the 1mm tip gap flow due to extreme forward loading of the blades.

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