CFD ANALYSIS TO INVESTIGATE THE EFFECT OF LEANED ROTOR ON THE PERFORMANCE OF TRANSONIC AXIAL FLOW COMPRESSOR STAGE

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ABSTRACT
This paper describes the numerical study carried out to understand the impact of combination of forward and backward leaned rotor on the overall performance of a transonic axial flow research compressor of CSIR National Aerospace Laboratories, Bengaluru. The analyses were carried out using commercial Computational Fluid Dynamics (CFD) solver Ansys-CFX. Initially CFD analysis was carried out for the baseline rotor configuration and validated the results with the experimental data. Then, three new configurations of combined leaned rotors were modeled from the radially stacked baseline rotor by changing the circumferential curvature of original stacking line using three control points located at 0%, 50% and 100% of the blade span. Analyses were carried out for all the three configurations of modified geometries and the results were compared with the baseline configuration. The results revealed many interesting aspects which will be very useful for better understanding of the blade curvature effects on the shock and secondary losses within the transonic rotor. The discussions on the effect of combination of forward and backward leaned rotor and the consequent developments on the overall performance of the compressor are presented in detail in this paper.

Keywords: CFD, Lean, Transonic compressor, Shock, Tip Leakage Flow.

NOMENCLATURE
BL Backward Lean
FL Forward Lean
PS Pressure Surface
SS Suction Surface

1. INTRODUCTION
To attain higher pressure ratios, the compressors of modern aero-engine have to operate at higher tip speeds and hence, the rotor is affected by a transonic flow field that is subsonic at the hub and supersonic at the tip. Due to supersonic flows at the tip, there are intense shock waves generated near the blade tip. Thus the flow field that develops inside a transonic compressor rotor is extremely complex involving flow features such as shock waves, intense secondary flows and shock to boundary layer interaction which deteriorate the performance of the compressor [1]. In order to improve the aero-engine compressor performance and to extend its stable operating range, certain advanced design strategies have been employed. One of the methods is to impart 3-Dimensional blading concept such as sweep and lean to the compressor blades. Different definitions of sweep and lean are used in the literature. Sweep and lean are related to the stacking of the blade sections. The aerofoil stacking may be defined based on any two sets of perpendicular axes. A common choice in compressor community is to define sweep; movement of blade sections along the local chord direction and lean is the change in perpendicular to chord direction. In the present work, the lean is defined relative to the tangential directions. Literature cites many works on the effect of 'sweep' but comparatively only few works have been carried out on the effect of blade 'lean'. Most of studies reported on lean [2] involves by defining lean as movement of blade sections perpendicular to the local chord direction. Further research is required to be carried out to better understand the leaned blades.

Denton and Xu et al. [2] performed a CFD analysis and demonstrated that effect of chord wise sweep and lean on transonic fan is remarkably small with respect to efficiency and pressure ratio. Bergner et al. [3] performed numerical analysis and observed that the chord wise lean can give rise to significant changes in shock pattern. Benini et al. [4] performed a multi-objective design optimization on NASA rotor 37 and demonstrated that overall efficiency is improved by giving the blade a proper lean towards direction of rotation. Jang and Yi et al. [5] demonstrated a positive impact on the rotor performance by providing leaned blades. Benini and Biollo et al. [6] performed a CFD analysis and demonstrated that forward leaned rotor showed improvement in efficiency and aft leaned rotors exhibited wider operating range. In the
In the present study, the forward and backward lean is achieved by linearly translating the aerofoil blade sections towards and opposite to the direction of rotation respectively. The lean is achieved by linearly translating the aerofoil blade sections from hub to tip in such a way that the amount of movement is zero at the hub and for a given lean angle at mid and tip sections. The forward and backward leaned angles were selected based on the previous study carried out for 5°, 10° and 15°[13]. In that study, it was found that 15°FL has 6.06% improvement in operating range when compared to baseline at design speed and 5°BL has 1.03% improvement in total pressure ratio without loss in operating range. Based on these results, a new scheme of combined effect of forward and backward leaned rotors were selected such as 5°BL(at mid section)+15°FL(at tip section), 15°FL(at mid section)+5°BL(at tip section) and 5°BL(at mid section)+5°FL(at tip section) were selected and . Fig. 1 shows the schematic of a combined leaned rotor blade.

![Figure 1. Schematic of combined Lean.](image1)

The present case study focuses on the CFD analysis to understand the effect of combined leaned rotor on the overall performance of 1.35 pressure ratio single stage transonic axial flow compressor.

2. CFD ANALYSIS OF BASELINE COMPRESSORS

The compressor considered here is a single stage transonic axial flow research compressor having 21 rotor blades and 18 stator vanes installed at CSIR-National Aerospace Laboratories, Bangalore. The compressor has a design speed of 12930 RPM with a pressure ratio of 1.35 at a corrected mass flow rate of 22kg/s. Commercial CAD tool Solid Works was used for the modelling of the baseline compressor stage and to generate the flow domain. One rotor blade and one stator vane with an inlet and outlet was considered for the flow domain. The flow domain which was extracted from the entire stage is as shown in Fig.2.

![Figure 2. Flow domain](image2)

The entire flow domain was split into 6 sub-divisions for the convenience of grid generation. The sub-domains are named as Volume 1, Volume 2, Volume 3, Volume 4, Rotor volume and Stator volume and are shown in Fig. 3. As the rotor and stator volumes are high curvature in nature, they are meshed with unstructured grid and the other volumes have simple geometry, meshed with structured grid as shown in Fig.4.

![Figure 3. Volumes of fluid domain](image3)

![Figure 4. Meshing of Domain](image4)

3. GRID INDEPENDENCY STUDY

A grid independency study is necessary to ensure that the solution is not affected by number of grid elements. The grid independency study for the baseline configuration of the referred test compressor was performed earlier [7] for four different grid configurations.
i.e. 876584, 1036975, 1256773 and 1450000 mesh elements. Fig.5. shows the efficiency plot for all four cases compared with the experimental data. The results proved that beyond 1036975 mesh elements, the mesh becomes grid independent and the results closely match with the experimental ones. But for the better skewness, grid size of 1450000 was chosen for the further analysis.

![Adiabatic Efficiency (%) vs Corrected Mass Flow Rate (kg/s)](chart1.png)

**Figure5.** Comparative performance map of the baseline configuration at 90% design speed for various number of elements.

### 4. BOUNDARY CONDITIONS

The boundary conditions used for the analysis were the total pressure at entry and the static pressure at exit. The total pressure at entry was fixed as 92kPa, which is the local ambient pressure and the exit static pressure was varied. Initially, the exit static pressure was set below ambient condition and then, it was increased till the near stall condition to obtain the performance characteristics of the compressor. The domain interface type used was fluid-to-fluid and the mixing plane between the rotor and the stator interface was selected as ‘frozen rotor’. Since, the compressor consists of 21 rotor blades and 18 stator vanes, the pitch angle between them was specified as 17.14285 (360/21) degrees and 20(360/180) degrees respectively. The advection scheme selected here was the upwind first order scheme and the residual target was input as $10^{-7}$. The convergence of the numerical solution cannot be merely judged by monitoring the residual graphs and hence relevant quantities such as weighted average of mass flow rate was monitored for the solution convergence.

### 5. VALIDATION

The validation of the CFD results was done by comparing the performance characteristics of the compressor with experimental data at 80% and 90% of design speed. Fig.6. shows the adiabatic efficiency and total pressure ratio plotted against the corrected mass flow rate. From this graph, it can be seen that the CFD results of baseline configuration having good agreement with the experimental data which formed the basis for further analyses.

![Total Pressure Ratio vs Corrected Mass Flow Rate (kg/s)](chart2.png)

**Figure6.** Comparative performance map of the baseline configuration

### 6. NATURE OF FLOW

The most detrimental region in a transonic compressor is the tip and end-wall, where the tip gap tends to develop intense secondary flows. The pressure difference between the suction side and the pressure side drives the fluid through the blade tip gap, inducing a fluid jet which propagates into the mainstream flow. The interaction between the jet and the mainstream flow gives rise to a vortex ('tip clearance vortex') [11]. Interacting with the shock near the tip of the rotor blade, the tip clearance vortex is subjected to a sudden deceleration and this interaction has a key role on rotor stability. Depending on the intensity of the interaction, this phenomenon can induce compressor stall [12]. Aerodynamic blockage refers to reduction in flow area, which affects the work output and the mass flow capacity in an axial-flow
compressor. Sources of blockage in a transonic axial compressor includes the blockage generated by tip clearance flows, blade boundary layers, end-wall boundary layers and the shock formation in tip region of the blade [1].

The importance of blockage in compressor performance prediction is cited by Cumpsty[1] as follows: "Small changes in flow have a very large effect on the stage performance as it affects both the mass flow at choke and the amount of work done by the rotor". The blockage is perhaps the most important phenomenon in an axial compressor which alters the flow behavior in a rotor blade and eventually affects the performance of the compressor. Such a blockage was seen in the results of baseline compressor generated with the help of velocity streamlines and Mach number contours captured in the post processing part of the analysis. Figure 7. shows the 3-D streamlines taken at three different blade spans for baseline configuration at 100% design speed at choking conditions.

Fig.7(a) shows the velocity streamlines taken at 99.5% blade span, which depicts the tip leakage flow from the pressure side to the suction side of the blade. The tip leakage flow and the mainstream flow are at different angles of incidence and when they interact, they tend to form a vortex near the trailing edge of the blade. Fig.7(b) shows velocity streamlines taken at 95% of blade span which shows the vortex formed due to interaction of tip leakage flow with mainstream flow. This vortex spans down up to 90% of blade span from the tip beyond which the flow is streamlined. Fig.7(c) shows the velocity streamlines taken at 85% of the blade span, which indicates a smooth flow taking place over the blade surface.

![Velocity Streamlines for Baseline Compressor at Different Span](image)

<table>
<thead>
<tr>
<th>Shock near trailing edge</th>
<th>Shock at Mid chord</th>
<th>Shock near leading edge</th>
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<td>SS</td>
<td>PS</td>
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(a) Tip leakage flow at 99.5% span
(b) Interaction of Tip leakage flow with mainstream flow at 95% span.
(c) Smooth streamline flow at 85% span

Figure 7. Velocity streamline for baseline compressor at different span

Figure 8. Mach contours of baseline configuration at 95% span

Mach contours for the baseline compressor at 100% design speed are shown in Fig.8. These contours are taken at choking condition, maximum efficiency condition and near stall condition for blade span of 95%. From Fig.8(a), it could be clearly observed that at choking condition there is a strong shock formation normal to the blade suction surface (SS) near the trailing edge and it is also seen at mid-chord distance on the pressure surface (PS). As the loading of the blades are increased, i.e., as the back pressure is increased up to maximum efficiency
condition, the shock moves towards the mid chord of the blades on the SS and near leading edge on the PS. At near stall condition, the shock further shifts its position towards leading edge on SS and it is totally detached from PS. These Mach contours are presented here to depict the classic nature of a transonic compressor as indicated by John D Denton et al[2].

7. CFD ANALYSIS ON LEANED ROTORS
The baseline compressor rotor blade is radially stacked one and it was geometrically modified for the current study by incepting 5°BL(at mid section)+15°FL(at tip section), 15°FL(at mid section)+5°BL(at tip section) and 5°BL(at mid section)+5°FL(at tip section). CFD analyses of the new leaned configurations were carried out in similar manner as that of the baseline compressors. For all the new configurations the hub section was fixed and only the tip and mid-section of the blade was moved. Fig.10 shows the 3-D model of leaned blade configurations.

8. RESULTS AND DISCUSSION
The numerical simulation was carried out at 80%, 90% and 100% of the design speed for all the leaned blade configurations and the performance results are presented as follows.

1. The overall performance characteristics of the compressor stage.

The overall performance characteristics of the compressor stage at three different leaned configurations such as 5°BL+15°FL, 15°FL+5°BL and 5°BL+5°FL were obtained at 80%, 90% and 100% design speed of the rotor and compared with the baseline configuration. Fig.10 shows the comparative compressor performance map at three different operating speeds of the rotor. The total pressure ratio and percentage efficiency are plotted against the corrected mass flow rate. In Fig. 10(a), it is seen that at design and off-design speed the operating range is improved for 5°BL+15°FL case as compared to the baseline configuration and the improvement of 3.15%, 6.46% and 7.37% respectively in the case of 100%, 90% and 80% design speed condition with negligible change in efficiency as shown in Fig.10(b).

![Figure 9. 3-D Model of rotor configurations](image1.png)

![Figure 10. Comparative performance map of the baseline configuration.](image2.png)
1.04% and 1.51% and operating range of 2.3%, 3.1% and 6.8% respectively in the case of 100%, 90% and 80% design speed condition.

2. Radial distribution of total pressure ratio and relative Mach number across the rotor.

The radial distribution of total pressure ratio at rotor inlet and exit are plotted against percentage of annulus height of the blade at design speed for maximum efficiency condition to study the flow behavior of the compressor at different configurations. The inlet stream wise plane is arbitrarily chosen at upstream of leading edge of the rotor and exit plane is downstream of trailing edge of the rotor.

The radial distribution of total pressure ratio at rotor inlet and exit plane is downstream of trailing edge of the rotor and upstream of leading edge of the rotor respectively. The numerical results show that the total pressure ratio is equal to 1.3 at the hub region for all the configuration including the baseline case and it is increasing towards the blade tip. It is also observed that in the case of 5°BL+15°FL, after 50% of the blade span the total pressure ratio is decreasing as compared to baseline configuration. In case of 15°FL+5°BL configuration, after 60% of the blade span the total pressure ratio is increasing as compared to baseline configuration. In the case of 5°BL+5°FL configuration, the total pressure ratio is almost same as baseline case.

Fig.12. shows the radial distribution of the relative Mach number at upstream and downstream of the rotor for different configurations of the rotors. It is observed that, the relative Mach number is equal to 0.55 at hub and 1.11 at the tip for all the blade configurations including the baseline at upstream of the rotor. This figure indicates that the combined leaning of rotor has a minor effect on the upstream velocity distribution. At downstream of the rotor the relative Mach number is 0.45 at the hub and 0.55 at the tip for baseline configuration. In the case of 5°BL+15°FL, the relative Mach number is marginally increased throughout the blade span when compared to the baseline configuration. In the case of 15°FL+5°BL, after 60% of the blade span the relative Mach number is decreasing when compared to the baseline case and for the case of 5°BL+5°FL relative Mach number is almost same as compared to the baseline case but near to the tip section it is increasing compared to the baseline case.

The reason for variation in total pressure ratio and relative Mach number is supported by Mach number contours and entropy contours. Figures 14(a) to 14(d) show the Mach number contour on the suction surface of the rotor blade for all the lean configurations. The Mach number contours are taken at the maximum efficiency condition at the design speed. In these figures, it is seen that the shock is present at the mid chord of the rotor in case of baseline rotor and it is radially projected towards hub. It is observed that, for 5°BL+15°FL the shock strength increases thereby reducing the efficiency of the compressor, for 15°FL+5°BL it is observed the shock strength reduces by changing from normal shock into oblique shock and also for 5°BL+5°FL shock strength reduces by changing of normal to oblique shock. This shows the reason for variation in efficiency for the combined leaned blades when compared to baseline case. The reason for increase in efficiency is also supported by the entropy contours. Fig.15(a) to 15(d) show the comparative static entropy contours at the suction surface of the rotor blade at the maximum
efficiency condition for the design speed. In these figures, it is clear for 15°FL+5°BL entropy increases, for 15°FL+5°BL entropy decreases and also for 5°BL+5°FL entropy decreases, decrease in entropy generation shows increase the isentropic efficiency and vice versa.

The variation in operating range of the compressor by incorporating combined lean to the rotor can be explained by using Mach number contours and velocity streamlines. Figures.16(a) to 16(d) shows the comparative Mach number contours at the design speed taken at 95% of the blade span at the maximum efficiency condition. From Fig.16(a), it could be seen that, the normal shock is present at the mid chord on the suction surface of the rotor in the case of baseline case. Whereas in case of 5°BL+15°FL rotor as shown in Fig.16(b), the shock moved downstream this indicates that, 5°BL+15°FL allow the compressor to operate at lower mass flow condition i.e. increase in operating range. As shown in Fig.5.23(c), for 15°FL+5°BL the shock moved upstream this indicates that, 15°FL+5°BL does not allow the compressor to operate at lower mass flow condition i.e. decrease in operating range and as shown in Fig.5.23(d), for 5°BL+5°FL the shock moved downstream this indicates that, 5°BL+15°FL allow the compressor to operate at lower mass flow condition i.e. increase in operating range.

Figures.17(a) to 17(d) shows the comparative velocity streamlines at choking condition for the design speed of the compressor, which are taken at 95% span. Figure.17(a) shows the accumulation of low momentum fluid at the trailing edge of the baseline rotor. This is due to the strong interaction of tip leakage flow and the mainstream flow, which act as an obstruction to the flow which leads the compressor to the aerodynamic instability called ‘stall’. Figure.17(b) shows the velocity streamlines taken for 5°BL+15°FL case. This figure shows less accumulation of low momentum fluid, for 15°FL+5°BL it is observed that as shown in Fig.17(c), accumulation of low momentum fluid increased and as shown in Fig.17(d), for 5°BL+5°FL accumulation of low momentum fluid decreased when compared to the baseline compressor stage. Therefore, it could be inferred that due to increase in low momentum fluid (blockage) there is an decrease in the operating range and vice versa.
9. CONCLUSIONS

Based on the steady state numerical analysis investigation performed on an axial flow transonic compressor using ANSYS-CFX, the following conclusion are drawn.

- Providing small amount of backward lean at mid and more amount of forward lean at tip the operating range increases with negligible change in efficiency.
- Providing more amount forward lean at mid and small amount of backward lean at the tip the pressure ratio and efficiency increases with a drop in operating range.
- Providing small amount of backward lean at the mid and the same small amount of forward lean at the tip the efficiency, pressure ratio and operating range increased.
- To highlight, the blade combined lean has got a notable effect on the shock formation as well as on the accumulation of low momentum fluid at the trailing edge of the rotor.

REFERENCES