Numerical Studies on the Effect of Slotted Casing Treatment on the Performance of a Transonic Axial Flow Compressor

A. K. Shivayogi¹, Q. H. Nagpurwala² and M. D. Deshpande³

1 - M. Sc. [Engg.] Student, 2 - Professor, Centre for Rotating Machinery Design, 3 - Professor and Head, Research M.S. Ramaiah School of Advanced Studies, Bangalore 560 054

Abstract

Casing treatment, or incorporation of slotted casing around the rotor, is one of the few techniques to delay the inception of flow instability such as rotating stall, thus enhancing the operating range of an axial compressor. However, the improvement in stall margin through casing treatment is accompanied by a penalty of reduction in compressor efficiency. Several studies have been carried out to optimise the casing treatment configurations in terms of global compressor performance parameters, but there is no clear understanding of the flow physics, which brings about the changes in the compressor performance and stalling behaviour. Further, a given casing treatment configuration may not yield the same results for all compressor builds.

This paper deals with the numerical studies on the effect of slotted casing treatment, having axial and radial skews of 15° and 45° respectively, on the overall performance and stall margin of a transonic compressor rotor. The steady state CFD simulations were performed at 100% design speed, with two axial positions of the treatment slots covering 100% and 38% of the rotor tip axial chord from leading end.

The results are compared with those obtained on the base line compressor with solid casing. It is shown that the presence of casing treatment significantly alters the stall margin and rotor efficiency compared to the solid casing. Detailed study of the flow field near the rotor tip and within the slots has shown migration of low energy fluid at the rotor pressure surface into the slots and re-emergence into the main flow at the rotor suction surface. The interaction between the casing treatment slots and the blade passage flow is relatively stronger at the stall mass flow rate. A favourable restructuring and energising of the boundary layer flow near the casing in the presence of treatment slots seems to be responsible for stall margin improvement. However, the recirculation of the fluid produces large entropy gradients in the tip region across the rotor, causing a decrease in compressor efficiency.

Key Words: Transonic Axial Compressor, Casing Treatment, Skewed Slots, Stall Margin, Flow Instability

Nomenclature

C Chord, mm
k Turbulence kinetic energy, m²/s²
LE Leading edge
m Mass flow rate, kg/sec
N Rotational speed, rpm
PR Pressure ratio
PS Pressure side
SS Suction side
TE Trailing edge
tm Maximum thickness, mm
ε Dissipation rate, m²/s³

1. INTRODUCTION

Requirement of high specific thrust in modern aero engines has lead to reduced number of compressor stages with higher pressure ratios in single stages. The high aerodynamic stage loading has, in turn, lead to greater tendency of flow instabilities like rotating stall and surge in multistage compressors. Rotating stall is usually initiated at the rotor tip where the blade loading is high in terms of diffusion factor and the flow is dominated by tip leakage vortices, which make strong impact on the performance of the compressor. In order to improve the stall margin and efficiency of the axial flow compressors, many options have been tried by the researchers. Casing treatment (Fig.1) is regarded as one of the most widely researched options and has shown potential for improving the stable operating range of axial compressors. Although, many investigations have been carried out to understand the specific impact of casing treatment on the compressor flow field, the fluid dynamic reason behind the effectiveness of casing treatment in improving the stall margin accompanied by a reduction in compressor efficiency, still remains to be explored.

Fig. 1 Casing treatment model

A number of researchers have studied different casing treatment configurations to understand their effect on compressor stall margin and efficiency. Nezym [1] carried out experiments to investigate the effect of circumferential grooves on axial flow compressor performance, using both traditional and new designs. It was shown that the grooves with different depth distribution, located only at the entrance and exit of blade tip projection, increased the efficiency. Osborn and Moore [2], and Bailey [3] also carried out similar kind of studies, limited to experimental
evaluation of compressor overall performance. Wilke and Kau [4], and Hembera et al [5] conducted numerical studies on a transonic axial flow compressor with the casing having axial semi circular slots over the rotor tip chord. Two different axial positions of the slots were studied. It was concluded that instead of a massive destruction of the tip leakage vortex (TLV), it was more advantageous to manipulate the tip leakage flow in the upstream part of the blade chord, which had favourable effect on efficiency. Hong Yang [6] carried out numerical unsteady flow computations on a transonic axial flow compressor for end wall recirculation treatment. Nine recirculation segments, parallel to the engine axis were modelled per blade pitch. It was shown that the casing treatment effectively extended the stall margin by improving the flow conditions near the casing and by weakening the tip leakage vortex. Gourdain et al [7] performed unsteady numerical simulation of casing treatment in a subsonic axial compressor stage. Semicircular bend type splitled slots covered only 20% of the tip chord. It was found that the boundary layer separation on the last 50% of the rotor chord and at the trailing edge was due to the blockage flow and the stator potential effects. Due to this, the improvement in stability margin was observed to be less. Xingen et al [8] performed unsteady numerical simulations on a subsonic compressor with axially skewed slot casing treatment, covering only 60% of the tip chord. It was found that depending upon the position of the suction surface and the rotor blade tip with respect to the slot opening, the pressure driven flow within the casing treatment re-circulates. Due to this, energy is supplied to the high loss, high blockage clearance flow, which affects the leakage of the flow downstream. Influence of the incidence angle, tip clearance vortex and the blade suction side separation were investigated by Emmrich et al [9] on a subsonic axial compressor stage provided with skewed slot casing treatment opening in a plenum chamber. The plenum chamber was responsible for constant extraction of the fluid from the blade passage flow that resulted in reduction of tip clearance vortex. The fluid particles inside the treatment slots were believed to convert rotor torque constantly into higher temperature, which caused reduction in compressor efficiency. Steady state numerical simulations were carried out by Kau and Wilke [10] to investigate the influence of casing treatment on the tip leakage flow and its resulting vortex. The loss in total pressure inside the vortex core, depending upon the blade loading, was the main reason for the vortex breakdown. Casing treatment enhanced the stall margin by delaying the onset of vortex breakdown, through weakening or even destroying the characteristic tip leakage vortex. It is realised that the work reported till now shows a broad mix of experimental and numerical studies. Several types of casing treatments have been considered. Despite the success of these demonstrations, challenges still exist for implementation in full-scale multistage compressors in a more economical way. The studies were mostly limited to the effect of casing treatment on the global performance of compressor stages. The fluid dynamic processes in the casing treatment slots are still not fully understood. In the present study, the flow through a typical transonic compressor rotor with skewed slot casing treatment has been numerically simulated to understand the fluid dynamic mechanism around the rotor tip region and within the treatment slots. The flow field is investigated with casing treatment slots positioned to cover 100% and 38% of the rotor tip chord. The simulation results are presented in terms of global compressor performance and detailed flow behaviour in the tip clearance and slot regions.

2. GEOMETRIC MODEL

A transonic axial flow compressor rotor of NACA design [11] was selected for the present study. The compressor stage, designed for a pressure ratio of 1.47 and an adiabatic efficiency of 89%, comprised 21 rotor blades and 18 stator blades. The rotor inlet tip relative Mach number was 1.13, and the rotor tip chord and inlet hub-tip radius ratio were 0.0825m and 0.516 respectively. More details on the compressor design can be found in ref. [11]. Since the mesh density and computational time required would be large if the whole stage were considered, it was decided to carry out the analysis for rotor domain alone. It was also verified through the literature that most of the numerical studies involving casing treatment were carried out with rotor alone. Geometric modelling of the rotor was carried out with the help of CATIA software. A rotor tip clearance of 0.583mm was included in the model. The flow domain was extracted by selecting a single blade passage and the same was extended about one blade chord upstream and one blade chord downstream for flow stabilisation. Figure 2 shows the rotor computational domain selected for study.

![Fig. 2 Flow domain for numerical study](image)

3. BOUNDARY CONDITIONS AND SOLVER SETTINGS

The flow domain was discretised using ICEM-CFD software with due regard to acceptable skewing angles of the volumes. It was ensured that the determinants were greater than 0.28 and the angles were greater than 18°. The grid density was refined near the hub, casing wall, rotor LE and TE, and in the tip clearance region. The mesh density was fine enough to keep Y+ values less than 50 at all locations so that the flow physics at the walls could be captured. Figure 3 shows the meshed model of the rotor domain. The simulations were performed using a commercially available solver, FLUENT. The k-ε model, with standard wall functions, was chosen for turbulence modelling. Due to convergence problem in
k-ε model near tip clearance, the analysis was carried out with the k-ω SST turbulence model for all the simulations with and without casing treatment as recommended by Chen et al [12] and Yamada et al [13]. Steady state computations were performed by applying a circumferentially averaging interface between the rotor and the casing treatment domains. Relative reference frame method was used. The imbalance of mass, momentum and energy between the inlet and the outlet was closely monitored and the convergence criterion of 1x10⁻⁶ was specified.

Before starting the analysis, relevant boundary conditions were specified in FLUENT, as shown in Fig.4. No-slip and adiabatic conditions were imposed on the solid walls. The casing wall was non-rotating (stationary wall). The hub and blade walls were modeled with rotating wall conditions. Periodic boundary conditions were specified at the outer boundaries that ensured that all the variables had the same value on the cyclic faces. The inlet flow was assumed to be axial. The treated casing slot domain was also kept stationary and the flow parameter communication between the treated casing and the rotor domain was through an interface.

![Fig. 3 Computational grid of the transonic axial flow compressor rotor](image)

![Fig. 4 Rotor flow domain with boundary conditions](image)

3. GRID INDEPENDENCE STUDY

A grid independence study was carried out with different grid sizes with 0.52 million, 0.72 million and 0.94 million cells for the baseline solid casing model. In order to capture the stall point accurately, the back pressure (static) was reduced to the lowest stable value. The near stall point was judged to be the last stable point prior to incurring a numerical stall [4]. The same criterion was applied for both solid casing and treated casing in establishing the numerical stall point. For all cases the stall point was established within 100Pa in terms of rotor back pressure. Published experimental data [11] was also used to compare the predicted results obtained from different grid sizes. Figure 5 shows the overall performance of the rotor with solid casing for three grid sizes. It is found that there is not much variation in the results obtained with 0.72 million and 0.94 million grid cells. Hence, a computational grid with 0.72 million cells was used for further study. As can be seen from Fig. 5, the peak experimental pressure ratio of 1.5 is higher than the computationally obtained peak pressure ratio of 1.48. The experimental mass flow rate at the choke condition (22.58 kg/sec) is higher than the predicted choke mass flow rate (21.82 kg/sec). At design point, the mass flow rate given by the computations is 22.36 kg/sec, which is slightly higher than the experimental value of 22 kg/sec.

![Fig. 5 Comparison of computational performance of rotor with experimental results and grid independence study](image)

4. GEOMETRY OF CASING TREATMENT SLOTS

The treated casing configuration was selected based on the data published by Emmerich et al [9]. A schematic of the treatment slot configuration is shown in Fig. 6 and the geometric data are presented in Table 1. A total of 168 slots were considered for the present simulations. Thus, each rotor passage covered a total number of 8 full slots. An open area of 67% was kept, while the slot width was 2.2 mm and the open space between the slots was 4 mm. The slots had axial and radial skew angles of 15º and 45º respectively, the later in the direction of rotor rotation. The radial height of the slots and the plenum height were 11 mm each. The treated slots were arranged to cover the rotor tip axial chord by 100% (CT-100) and 38% (CT-38).

In all simulations with and without casing treatment, the rotor tip radius and the tip clearance were kept constant, so that the changes in the flow structure could only be associated with casing treatment. The number of grid cells in the tip clearance was refined to 8 points in order to gain a better resolution around the interface. Grid embedded in the blade tip clearance consisted of 60 × 8 × 68 cells in the chordwise, spanwise and pitchwise directions, respectively. The whole grid system had 6,98,580 cells in the rotor domain and 1,74,628 cells in the casing slots.
4. OVERALL ROTOR PERFORMANCE

Fig. 7a and 7b show the overall rotor performance in terms of total pressure ratio and efficiency for the solid casing and the two cases of treated casing, CT-100 and CT-38. Casing treatment slots appear to induce an increase in stall margin with a drop in compressor efficiency. It is observed that the CT-38 configuration yields better performance compared to the CT-100.

Comparison with the results of the solid casing show a deviation in pressure ratio of about ~5% in case of CT-100 and 0.9% in case of CT-38 at a mass flow rate near maximum efficiency point. The drop in efficiency is about 2.5% for CT-38 and 15% for CT-100. The simulations could be extended beyond the stability limit of solid casing to lower mass flow rates of 13 kg/sec and 16.6 kg/sec for CT-38 and CT-100. Consequently, the stall margin improvements with CT-38 and CT-100 configurations are 28% and 15% respectively. The extension of the stable operating range due to casing treatment is quite substantial and is in line with the results reported in the literature.

5. DETAILED FLOW ANALYSIS

5.1 Definition of Flow Analysis Planes

Relative Mach number contours are plotted at different planes for baseline and treated casing rotors and the flow structure is studied at mass flow rates corresponding to the maximum efficiency point and stall point of the solid casing. The flow contours are plotted in spanwise planes along the chord as well as in blade-to-blade planes near the tip of the rotor blade and in the casing treatment slot region. These reference planes are shown in Fig. 8 to 12.
Fig. 10 Transverse reference planes (CT-38)

Fig. 11 Meridional reference planes in pitchwise direction with respect to the treatment slots

Fig. 12 Spanwise planes (blade to blade)

The meridional planes (Fig. 11) are chosen at three pitchwise locations at the rotor tip. The first part of each plane (upper part) passes through the casing treatment slots and extends into the plenum at 45° to the radial direction; and the second part extends radially into the rotor blade passage. One such plane, passing through the midchord, is shown in Fig. 11. However, for CT-38, these planes coincide with the leading edge, middle and trailing edge of the slots.

5.2 Rotor Flow Field at Solid Casing Stall Point

The distribution of relative Mach number in the blade to blade plane in the tip region for the operating point corresponding to stall mass flow rate of solid casing is shown in Fig. 13. Owing to the increase in positive incidence at stall point, the location of peak suction surface Mach number is shifted towards the leading edge (Fig. 13 (a) and Fig. 13 (d)). Peak suction surface Mach number for solid casing is about 1.5.

For CT-100 geometry, a series of high Mach number patches (Fig. 13(b) and Fig. 13(e)) near the suction side indicate the tip flow entering the casing treatment slots. Only at the trailing edge, there is a low velocity region showing some flow blockage. In the rear portion of the blade on the suction surface, there is an indication of flow separation and there is low energy fluid zone attached to the pressure surface (Fig. 13 (b)). Between the suction surface separation region and the low energy fluid zone, there is a region of high Mach number of the order of 0.8. Further downstream, the flow decelerates to lower Mach numbers.

For CT-38 geometry, the acceleration near the leading edge on the suction surface is not predominant. The tip clearance vortex develops at about 50% of the tip chord and grows downstream, covering almost 90% of blade pitch in the tip region. The initiation of tip leakage vortex is shifted downstream compared to the smooth casing. It is noted that for CT-38 configuration,
the separated flow region is more or less similar to the solid casing and the flow field in the casing treatment of CT-100 configuration seems to be better compared to the other two cases. Inspite of this, the stall margin improvement in case of CT-38 is higher than for CT-100.

The radial distribution of relative Mach number in the streamwise planes for the three test cases at the operating point corresponding to the stall mass flow rate of solid casing is shown in Fig.14. It can be observed that the regions of high suction surface Mach number near the leading edge are slightly below the blade tip. In case of casing treatment CT-38, the high suction surface Mach number zone is almost 10% of the blade span below the blade tip. The radial extent of the tip clearance vortex at midchord and at the trailing edge is clearly observed.

![Fig. 14 Radial distribution of relative Mach number in streamwise planes for solid casing stall flow rate](image)

**Fig. 14** Radial distribution of relative Mach number in streamwise planes for solid casing stall flow rate

A decrease in relative total pressure from leading edge to mid chord but the reduction is less than that for CT-100 but more than the solid casing. Beyond midchord, the relative total pressure tends to increase and comes to the level of the solid casing. The relative total pressure loss across the rotor in the tip region is lowest for CT-38 configuration.

![Fig. 15 Relative total pressure distribution on streamwise Planes 1, 2 and 3 at solid casing stall flow rate](image)

**Fig. 15** Relative total pressure distribution on streamwise Planes 1, 2 and 3 at solid casing stall flow rate

The distribution of radial velocity in the streamwise transverse planes corresponding to the leading edge, middle and the trailing edge of the casing treatment slots is shown in Fig. 16. Also, the radial velocity distribution in the pitchwise meridional planes is shown in Fig.17. It has been observed that the flow physics in the casing treatment slots at stall flow rate of solid casing is more or less same as that at the peak efficiency operating points. However, owing to higher compressor loading at stall point, the magnitude of radial velocities of the flow into and out of the slots is higher than the radial velocities at peak efficiency operating point.

![Fig. 16 Distribution of radial velocity in three streamwise planes with respect to the treated slots at solid casing stall flow rate](image)

**Fig. 16** Distribution of radial velocity in three streamwise planes with respect to the treated slots at solid casing stall flow rate

![Fig. 17 Distribution of radial velocity in three pitchwise planes with respect to the treated slots at solid casing stall flow rate](image)

**Fig. 17** Distribution of radial velocity in three pitchwise planes with respect to the treated slots at solid casing stall flow rate

The radial distribution of relative Mach number in the spanwise reference Plane-4 and -5 at respective stall points of CT-100 and CT-38 is shown in Fig.18. It can be observed that the regions of high suction surface Mach number near the leading edge are slightly below the blade tip. In case of casing treatment CT-38, the high suction surface Mach number zone is almost 10% of the blade span below the blade tip. The radial extent of the tip clearance vortex at midchord and at the trailing edge is clearly observed.

![Fig. 18 Distribution of relative Mach number on spanwise reference Plane-4 and -5 at respective stall points of CT-100 and CT-38](image)

**Fig. 18** Distribution of relative Mach number on spanwise reference Plane-4 and -5 at respective stall points of CT-100 and CT-38
5.3 Rotor Flow Field at CT-100 and CT-38 Stall Point

It is worthwhile comparing the flow field at the respective rotor stall points of CT-100 and CT-38 treatment configurations. The pitchwise relative Mach number distribution for the three cases is shown Fig.18. It is observed that at high positive incidence angles (low mass flow rates), the flow quality is much worse than for the operating point corresponding to the solid casing stall flow rate. Almost the entire blade pitch is covered with low energy fluid and its radial extent is also increased to 15% blade span from the tip. Overall, the flow conditions at the respective stall points of rotor with CT-38 and CT-100 casing treatment configuration are worse than that at the stall point of solid casing, owing to much larger positive incidences at low flow rates.

Referring to Fig. 7, it is evident that the stall margin for casing treatment with 38% axial chord coverage is substantially higher than the smooth casing and casing treatment of 100% axial chord coverage. The flow field within the rotor passage, as discussed above, shows that the blockage caused by the tip leakage flow is comparable for the two cases of solid casing and CT-38 casing treatment. Inspite of this the CT-38 configuration shows an improvement in stall margin. It is widely reported that the compressor stall is initiated by the separation of the suction surface boundary layer near the leading edge under the influence of large positive incidence angles. In the present case, although the casing treatment with 38% axial chord coverage shows considerable blockage at the rotor at the trailing edge but still the leading edge flow separation is suppressed. And this may be due to the favourable interaction of the flow within and outside the casing treatment slots. It is likely that the ejection of the fluid from casing treatment slots near the suction surface causes energisation of the boundary layer and delays its separation. This may be one of the reasons for improvement in stall margin.

5.4 Stall Margin Improvement Mechanism

Mechanism of flow entering the slot near the suction surface and ejecting into the main flow near the pressure surface can be clearly seen in Fig.19. The pitchwise section is taken at 1 mm inside the slots from the casing inner radius corresponding to a radius of 222 mm. The distribution of radial velocity is shown for the two configurations, CT-100 and CT-38. It is observed that in case of CT-100, the flow enters the slots at locations close to the blade pressure surface. However, the ejection of the fluid is initially closer to the suction surface up to about 30% chord from the leading edge (Fig. 19 (a)). Thereafter, the ejection location moves farther away from the suction surface. Hence, the effect of fluid ejection will be limited only to the forward portion of the blade suction surface.

In case of CT-38 (Fig. 19 (b)), the fluid entry and ejection is similar to the CT-100 but the ejection velocities are lower than CT-100. Upstream of the blade leading edge, the radial velocities are almost zero, indicating weak fluid interaction between treatment slots and the blade passage flow.

![Fig. 20 Radial distribution of total temperature near rotor TE at solid casing stall flow rate](image1)

![Fig. 21 Radial distribution of entropy near rotor TE at solid casing stall flow rate](image2)

Figure 20 shows the radial distribution of total temperature near rotor trailing edge at mass flow rates corresponding to the solid casing stall point. It is evident from Fig. 20 that the temperature rise across the rotor blade in the tip region is more in case of CT-100 compared to solid casing and CT-38. This temperature rise is due to recirculation of the flow within the casing treatment and the rotor tip region, as shown in Fig. 17. In case of CT-38, only a part of the blade is covered under the treated casing and hence the extent of recirculation is less compared to the CT-100, which leads to lesser temperature rise.
Fig. 22 Comparison of radial isentropic efficiency for rotor with and without casing treatment at solid casing stall flow rate

Figure 21 shows the radial distribution of entropy at mass flow rate corresponding to stall point of the solid casing. Since the temperature rise is more in case of CT-100, the entropy rise is also high compared to the CT-38 and solid casing. This could be one of the reasons for decrease in rotor efficiency (Fig. 22) with casing treatment compared to the solid casing. Also, the casing treatment slots slightly reduce the pressure difference between the blade pressure and suction side near tip region, which is the main driving force for the tip clearance vortex. Hence the tip clearance vortex strength is reduced. Furthermore, the removal of fluid by the casing treatment near rotor LE leads to a reduction of the blockage and improves the flow structure (Fig. 16).

6. CONCLUSION

- The predicted overall performance of the compressor with solid casing (baseline) agreed well with the available experimental data.
- Two different axial coverage of radially and axially skewed casing treatment slots over the blade tip chord showed improvements in compressor stall margin compared with the solid casing. A drop in isentropic efficiency was noticed in both the cases.
- Flow recirculation within the treatment slots and the rotor tip region causes large increase in temperature across the rotor (tip region) resulting in large change in entropy. This, in turn, is responsible for drop in compressor efficiency with casing treatment.
- Casing treatment with 38% axial coverage over rotor tip chord showed highest improvement of ~28% with decrease of 0.4% in compressor efficiency.
- Casing treatment with 100% coverage over rotor blade tip chord showed a relatively lower improvement in stall margin of ~12% with a higher associated drop in compressor efficiency of 12%.
- The flow behaviour at the rotor tip region was analysed for understanding the flow physics. It is observed that the displacement of low momentum fluid on the blade suction surface is responsible for stall margin improvement.
- There is a strong interaction of the rotor flow and the casing treatment resulting in fluid entry into the slots near pressure surface and ejection into the main flow near the suction surface. The fluid motion within the slots and in the plenum is skewed radially and axially.

REFERENCES