Published by :
http://www.ijert.org

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 6 Issue 10, October - 2017

Effect of Rotor Tip Casing Treatment on the Performance and Stall Margin of a Transonic Axial Compressor Stage

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***Abstract* - This paper deals with the numerical investigations on the effect of mean camber line casing treatment slot on the performance of a NACA transonic axial compressor stage. From the literature review on it is revealed that in transonic compressor tip leakage flow interacts with end wall boundary layer, core flow and shock waves. To improve the flow in tip region researchers started using casing treatments. Casing treatments consists of slots, grooves or cavities on rotor casing, and recirculation type. From experimental and numerical studies it is found that casing treatment improves stall margin. Effect of casing treatment on axial compressor rotor tip region has shown some positive results in improving the stall margin and hence increasing engine operating range. Different type of casing treatments are employed on different axial compressor rotor/stage geometries. Investigation of axial compressor stall control by casing treatment can be done with less difficulty either by experiment or numerical computation. Because of easy design and some extent of assured positive results, casing treatment has attracted axial compressor researchers to pursue continuous investigative studies till date.**

. The mean camber line casing treatment has shown positive results in improving stall margin to still lower mass flow rate. Total pressure ratio with casing treatment was less than base line. Parametric studies were carried out varying slot height from 11mm and 4mm for constant number of 181 slots.

Key Words* - *Axial Transonic Compressor, Trailing Edge Crenulation, Stall Margin, Cfd, Gas Turbine.

INTRODUCTION

In the area of turbomachinery, either compressor or turbine flow in rotor tip clearance region is very complex. Boundary layers develop on casing surface and tip leakage flow interacts with endwall boundary layer resulting in total pressure loss. Different geometry modifications are done to in improve flow quality and reduce flow losses. Performance evaluation for these modified axial compressor geometries is done through experimental studies and numerical investigation. Numerical investigation is done using CFD codes. The goal of geometry modification is to study its effect on the above mentioned flow physics i.e., flow development, stall and surge, tip leakage flows, secondary flows end wall blockage etc.

Axial compressor stall control is studied with different and verity of casing treatment geometries. In the literature review different type of casing treatment discussed briefly with results and conclusion from the past investigations and research work. However, there was not much clarity on the optimal geometry of the casing treatment slot. In the current project work also a new type of casing treatment will be investigated to improve axial compressor stall performance.

Emmrich et al. [1] have investigated a highly subsonic axial compressor stage with casing treatment both experimentally and numerically for understanding the flow physics. Effect of casing treatment with axial and radial skewed slots for a high subsonic axial compressor stage, improved stall margin but with slight loss in efficiency. The stabilizing effect is attributed to the extraction of low momentum fluid from the tip clearance vortex at the blade pressure side. Wilke and Kau [2] has investigated the effect of casing treatment on vortex breakdown near stall point. From analysis results it is found that the rolling-up mechanism of the vortex got weakened or even blocked by the modeled grooves and slots. The losses inside the remaining vortex core were not as intensive as in a fully developed tip leakage vortex. This helped in understanding that the vortex breakdown is responsible for stall in the untreated case. This revealed that in casing treatment delays vortex break down and helps in moving surge line to lower mass flows rates. Wilke and Kau [3] have shown that cavities or slots can successfully reduce the detrimental three- dimensional flow phenomena

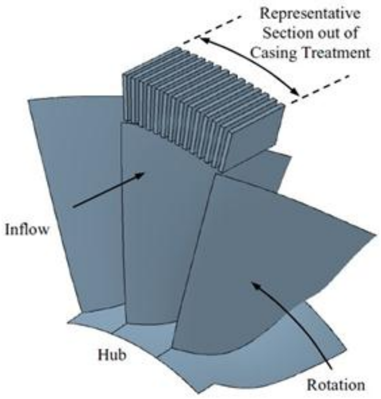

Fig.1 casing treatment slot on rotor tip region of a compressor blade [1]

originating at the rotor tip gap. But the authors did not observe same effect at different design speeds with the tip in nearly subsonic conditions. The losses introduced with the CT by the recirculation of the fluid were larger than the effects of the removal of low-momentum fluid. Hence same CT configuration may not give same advantages over performance of same compressor.

Xingen Lu et al [4] has carried steady state CFD simulation on series of discrete square-edged axial skewed slots located over the rotor tip provide the end-wall treatment. It is understood that in untreated casing wall the tip clearance vortex trajectory was found to move upstream as the mass flow rate decreased. Unsteady simulations are more important to understand the complete flow mechanism for rotor with casing treatment.

Chen Haixin et al. [5] simulated the flow in a transonic axial compressor NASA Rotor 37 with the help of in-house developed code. The Circumferential Grooves Casing Treatment (CGCT), used to find out improvement of the stall margin for axial flow compressor and the influence of the height of the tip clearance, was studied for three cases. When the tip gap was small, stall was found to start from the burst up of trailing edge separation or trailing edge vortex, which was enhanced by the blade tip leakage flow. For the small tip gap configuration, the grooves in the mid-chord were more effective, while for the large tip gap configuration, the leading edge grooves worked well.

Voges et al. [6] have worked on a single-stage transonic axial compressor with casing treatment (CT). Their configuration consisted of 3.5 axial slots per rotor pitch in order to investigate the predicted extension of the stall margin characteristics, both numerically and experimentally. The experimental results showed a rather moderate effect in terms of stall margin extension of about 10% at design speed and 15% at 65% of the nominal speed. Legras et al. [7] indicates that circumferential grooves limit the expansion of the tip leakage vortex perpendicular to blade chord. But it generates lot of several secondary tip vortices due interaction with leakage mass flow. It delay onset of rotating stall by decreasing differential pressure across suction and pressure surface in rotor region. This reduced pressure difference in turn reduces the leakage mass flow involved in tip leakage vortex formation. Vishwas Iyengar and Laxmi N. Sankar [8] has worked on passive technique of self recirculating casing treatment on NASA Rotor 67. It was shown that control concept can potentially alleviate the instabilities at lower mass flow rates and hence increase the stable operating range of the compressor. This was traced to the removal of low momentum fluid downstream of the blade trailing edge, and the energizing effects of the flow injection on the leading edge vortex flow field in the tip region. Junqiang Zhu and Wuli Chu [9] has worked new type of bent skewed groove casing treatment shown in and investigated the influence of this type of casing treatment configuration on the isolated rotor performance, The simple axial skewed groove casing treatment were also investigated experimentally with the help of an axial flow compressor rotor test rig and the performance was compared with that of the new design.

From the foregoing discussion it is evident that the introduction of casing treatment slot has potential benefits in terms of increased stall margin and hence operating range at particular rotational speed. Hence, the objective of the present research was to investigate the effect of rotor mean camber line casing treatment in a transonic axial compressor rotor and stage performance through CFD simulations in order to understand the flow behavior of the compressor blades under design and off-design operating conditions. The simulation results are presented in the form of overall compressor performance and flow development in and around the rotor tip with casing treatment region.

NOMENCLATURE

G	Grid
k	Turbulence kinetic energy, m^2/s^2
P	Pressure, Pa
ε	Dissipation rate, m^2/s^3
η	Efficiency

Abbreviations

CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
GT	Gas Turbine
CT	Casing Treatment
LE	Leading Edge
NACA	National Advisory Committee of Aeronautics
PR	Pressure Ratio
RANS	Reynolds-Averaged Navier-Stokes
rpm	Revolutions per Minute
TE	Trailing Edge

TRANSONIC COMPRESSOR

A NACA transonic axial flow compressor rotor was selected for the present investigation because the geometric data and experimental performance of this compressor were readily available in published literature [10,11,12]. The selected axial compressor comprised 21 rotor blades and 18 stator blades. The stage was designed for a total pressure ratio of 1.35 and a rotor tip Mach number of 0.9. The 3D CAD model of the compressor stage is shown in Fig.2 and the design parameters are given in Table 1. More details on this compressor can be found in Ref. [10,11]. Only rotor was considered in the present investigations.

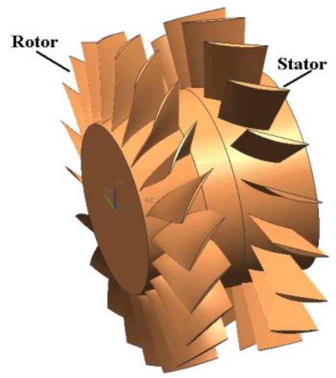

Fig.2 3-d cad model of NACA transonic compressor stage

TABLE 1 DESIGN SPECIFICATIONS OF NACA TRANSONIC COMPRESSOR ROTOR [10, 11]

Sl.No.	Parameter	Value
1	No. of Rotor Blades	21
2	No. of Stator Blades	18
3	Pressure Ratio	1.35
4	Corrected Mass Flow Rate	22.05 kg/s
5	Rotor Speed	13205 rpm
6	Isentropic Efficiency	89%
7	Relative Mach No.	0.75 (hub)/1.13 (tip)
8	Tip Radius	219.89 mm
9	Hub Radius	115.44 mm
10	Tip Clearance	0.62 mm

GEOMETRIC MODELING AND COMPUTATIONAL FLOW DOMAIN

Owing to rotational symmetry, the CAD model for only one rotor blade passage was generated in CATIA V5 R19 software, as shown in Fig.3. The computational domain was extended upstream and downstream of the rotor for better flow stabilisation. Based on our experience an upstream domain extension of twice the blade chord and a downstream domain extension of three times the blade chord are quite reasonable. Figure 3 also shows the boundary conditions applied to the walls and to the planes of symmetry. Discretization of the flow domain with hexahedral cells was carried out using ANSYS ICEMCFD software. The nodes of this grid are placed on the families of the lines so that the mesh gets aligned with the flow, and therefore, this provides a better mesh for streamline bodies. Further refinement of the grid elements was possible by just increasing the number of nodes, making use of the available options. The meshed model of the computational fluid domain is shown in Fig. 4.

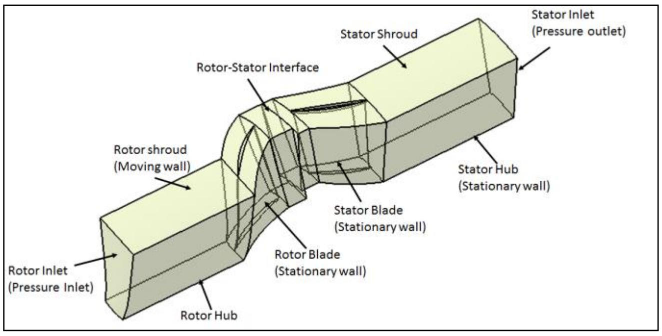

Fig.3 computational domain comprising one rotor and stator blade passage

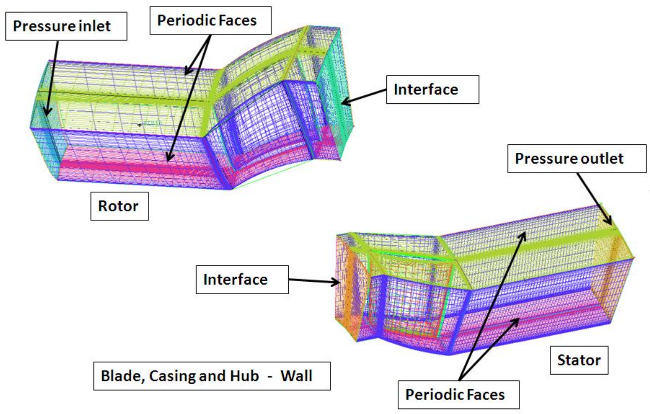

Fig.4 discretized computational domain for one rotor blade passage

SOLVER SETTINGS

The software used in the present simulations was ANSYS CFX Ver 12.0. Three dimensional, viscous, steady state pressure based implicit segregated second order upwind scheme was used to solve the Reynolds-Averaged Navier-Stokes (RANS) equations. Standard *k-ω* turbulence model with standard wall function was used to solve the turbulent flow t model is quite widely used in turbomachinery flow simulations. The rotor domain along with the hub wall rotated around the axis of rotation. The casing wall was given rotation in a direction opposite to the rotor. No-slip flow condition was imposed on all the solid walls. The inlet flow conditions comprised total pressure, total temperature (both at seal level) and flow angle, whereas static pressure was specified at the exit. All simulations were converged to a residual value of 10^{-5}.

GRID INDEPENDENCE STUDY AND VALIDATION

Grid independence study was carried out in order to check the effect of grid size on predicted results. Numerical simulations on the compressor stage were performed with different grid densities. Initially, a coarse grid (G1) was created with approximately 0.5 million elements. Then, to check the grid independence, two more grids, G2 and G3, were generated with approximately 1.1 million and 2.2 million cells respectively. Grid is created using ANSYS ICEMCFD software. Compressor performance was obtained at 100% design speed for different mass flow rates from choke to stall point by varying the exit static pressure. The stall point was resolved within a back pressure differential of 50 Pascal.

The stage performance characteristics in terms of the variation of total pressure ratio and isentropic efficiency with corrected mass flow rate are plotted in Fig. 5 and 6 respectively for all the three grids. The grids with 1.1 million and 2.2 million elements do not show any significant difference in the results. The values of maximum pressure ratio and maximum isentropic efficiency are nearly 1.43 and 90% respectively and are approximately same for both the finer meshes, G2 and G3. The experimental performance data of the compressor rotor from Ref. [10] are also plotted in Fig. 5 and 6 for comparison. Both total pressure ratio and efficiency are over-predicted in the choke region and under-predicted towards stall point. But, the mass flow rate is under-predicted in the choke region. The experimental peak pressure ratio of 1.47 is higher than the computationally obtained peak pressure ratio of 1.43. The experimental mass flow rate at the choke condition is 21.82 kg/s, and this is higher than the predicted choke mass flow rate of 22.3 kg/s. Overall the predicted pressure ratio and efficiency show reasonable agreement with the experimental data.

Based on the grid independence study, the grid G2 with approximately 1.1 million hexahedral elements, was selected as the final mesh for further computations.

The experimental test facility described in ref. [10] and computational set up used in the present investigations suggest the following possible reasons for deviation between results of CFD simulations and experimental data:

- The real flow field is highly unsteady, but the numerical simulations were carried out assuming steady state conditions.
- The inlet and exit domains of the experimental test facility were much longer compared to the computational domains used in the numerical simulation.
- The inlet conditions or parameters in the experiments [10] were not uniform across the radius owing to the presence of the hub nose cone and substantial thickness of inlet hub and casing boundary layers. The inlet axial velocity in the experiments varied from hub to tip owing to initial flow stagnation on the nose cone, but the computations were carried out with the assumption of uniform inlet axial velocity.

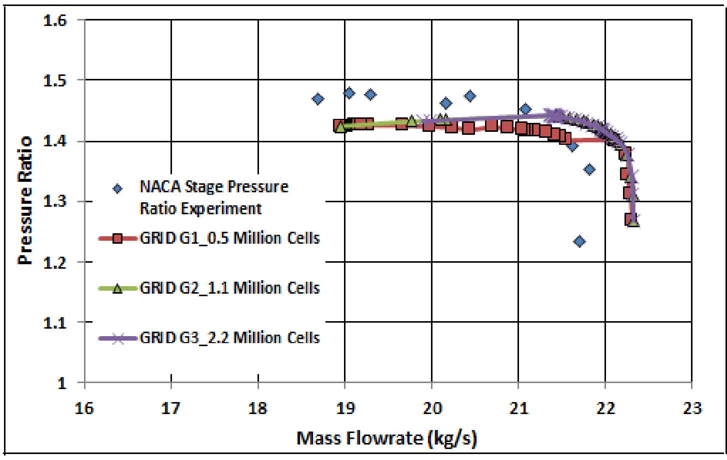

Fig.5 grid independence study – variation of total pressure ratio with mass flow rate

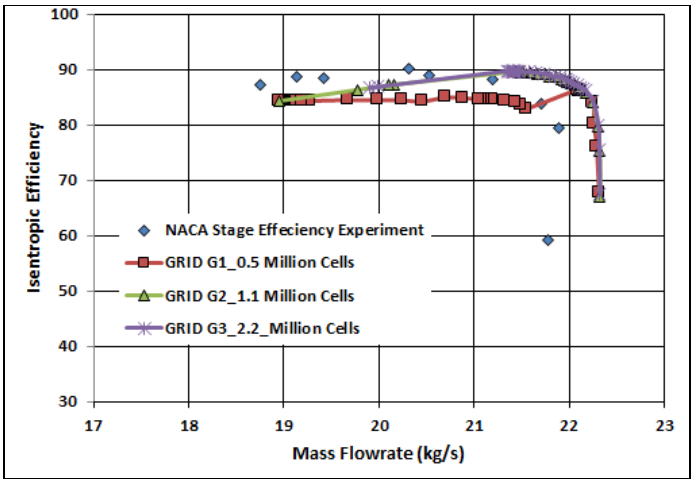

Fig.6 grid independence study – variation of rotor efficiency with mass flow rate

MODELING OF COMPRESSOR STAGE WITH CASING TREATMENT SLOTS

There are many geometric parameters, number of slots and pitch of slots, aspect ratio, etc. As a first attempt to investigate the effect of mean camber line casing treatment on the performance of an axial compressor rotor, attention was paid only to vary the number and aspect ratio of the slots. The CAD models of the compressor stage with two different casing treatment configurations (CT-1 and CT-2) were generated using CATIA V5 R19 software. CAD model of compressor stage with casing treatment configuration (CT-1) is shown in the Fig.7. The design specifications of casing treatment slots for two configurations are given in Table 2.

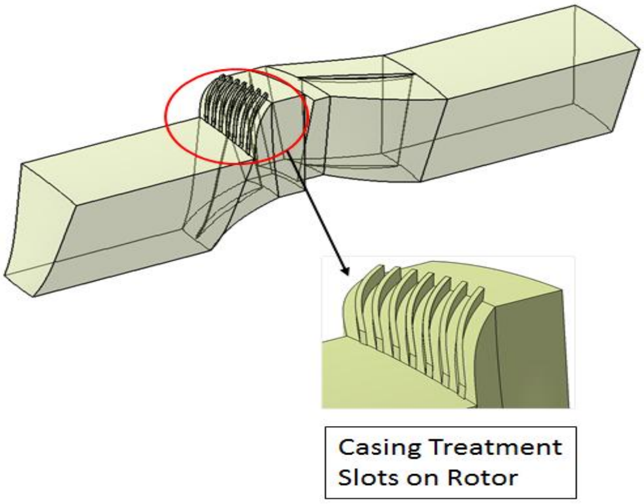

Fig. 7 compressor stage with casing treatment slots

Table 2 design specifications of trailing edge crenulation slots

Sl. No.	Nomenclature	Slot size, mm		Aspect ratio	Number of slots per periodic sector
		Height	Width		
1	NACA Stage with CT-1	11	2	5.5	7
2	NACA Stage with CT-2	4	2	2	7

The computational model, flow domain and boundary conditions for casing treated stage model were similar to those used for baseline stage model with smooth shroud wall. A grid size of >0.5 million cells was used in rotor zone for both the design cases of CT-1 and CT-2. Computational domain of compressor stage with rotor tip mean camber casing treatment slots (CT-1) is shown in the Fig.8. The actual computational grid sizes for compressor stage with casing treatment are given in Table 3.

Table 3 computational grid size for stage with casing treatment slots

S. No.	Computational Grid	No. of Hexahedral Cells
1	Baseline Compressor Stage	0.95 million
2	Compressor Stage with Casing Treatment slots	1.23 million

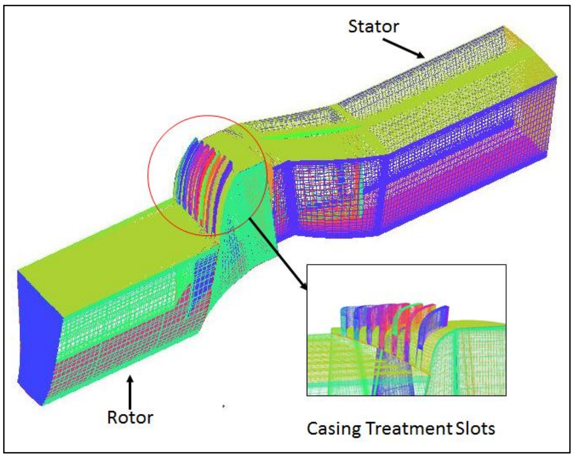

Fig. 8 computational grid for compressor stage with casing treatment slots

EFFECT OF MEAN CAMBER LINE CASING TREATMENT SLOTS

CFD simulation of the NACA transonic compressor stage with two different casing treatment geometries was carried out using ANSYS CFX software. The performance of compressor stage with casing treatment was obtained in terms of variation of total pressure ratio and isentropic efficiency with mass flow rate, and the results were compared with those of baseline rotor without casing treatment.

The performance variation in terms of total pressure ratio and isentropic efficiency with respect to mass flow rate for NACA compressor stage is shown in Fig.9 and Fig.10. In the choke region, the mass flow rates for casing treated compressor stage are slightly higher than the base line stage. From the performance plot is observed that for mass flow rates between choke and stall point, total pressure ratios with casing treated compressor stage are lower than the baseline stage. Total pressure ratio of compressor stage with casing treatment (CT-1) is very less than the baseline stage. In case of compressor stage with casing treatment (CT-2) shows lowest pressure ratio than baseline stage but higher the stage with casing treatment (CT-1). From this observation it concluded that height of casing treatment slots has significant impact on the compressor stage pressure ratio. Towards stall mass flow rate, the pressure ratios for casing treated stages are under-predicted than the baseline stage. Stall point mass flow rate (17.27 kg/s) for stage with casing treatment (CT-1) is lower than the baseline stage stall mass flow rate (18.9 kg/s), which results in increased stall margin of 8.6%. Lower stall mass flow rate also leads to increased engine operating range. Overall, the stage with casing treatment (CT-1) produces lower stall mass flow rate compared to baseline and stage with casing treatment (CT-2) in the entire mass flow rate range.

Isentropic efficiency of baseline stage and casing treated stage (CT-2) predicted from CFD simulations are higher in choke region and lower towards to stall region compared to experiment values and .Efficiency of casing treated stage (CT-1) is under-predicted both in choke and stall region.

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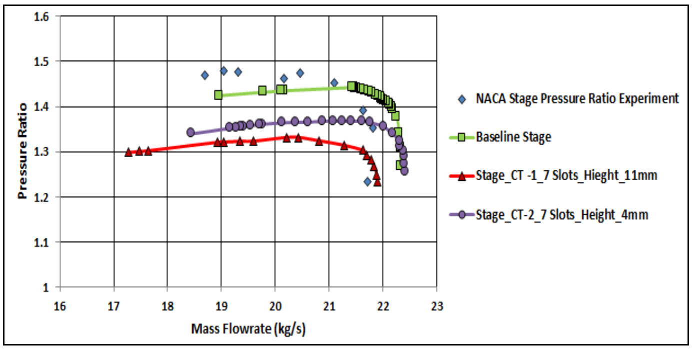

Fig.9 variation of total pressure ratio with mass flow rate at design rotor speed

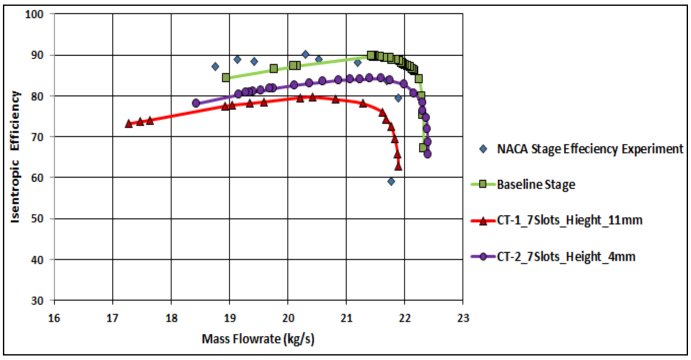

Fig.10 variation of isentropic effeciency with mass flow rate at design rotor speed

SPAN WISE FLOW DEVELOPMENT AT ROTOR EXIT

The flow through the computational domain is analyzed in the NACA compressor stage without and with mean camber casing treatment. Two flow parameters, *viz*. exit relative Mach number and exit entropy at the same operating condition (in terms of mass flow rate) for all the three cases are examined. Two streamwise planes are considered of which the streamwise Plane-1 is created almost at the rotor blade trailing edge, whereas the streamwise Plane-2 is created ~8 mm behind the rotor blade trailing edge. On these two streamwise planes, the contours of relative Mach number and entropy for the baseline rotor as well as for the rotors with casing treatment case CT-1 and CT-2 are plotted. Figure 11. shows the variation of exit relative Mach number on streamwise Plane-1. The presence casing treatment slots has retarded the momentum of fluid at the tip region as shown in Fig. 11(b) and 11(c). Another noticeable feature is the low velocity patch at the casing endwall of casing treated rotor is more than the baseline rotor. It is evident that casing treatment slots is able to bring about a re-distribution of the flow field in the compressor blade passage and more so at the blade trailing edge.

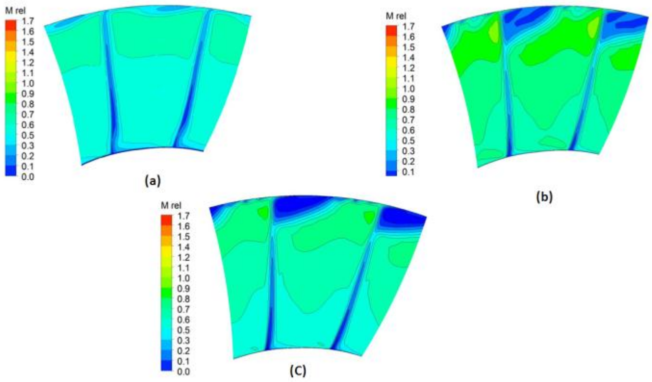

Fig.11 variation of relative Mach number on streamwise plane-1 at near-design mass flow rate of 21.6 kg/s: (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-1

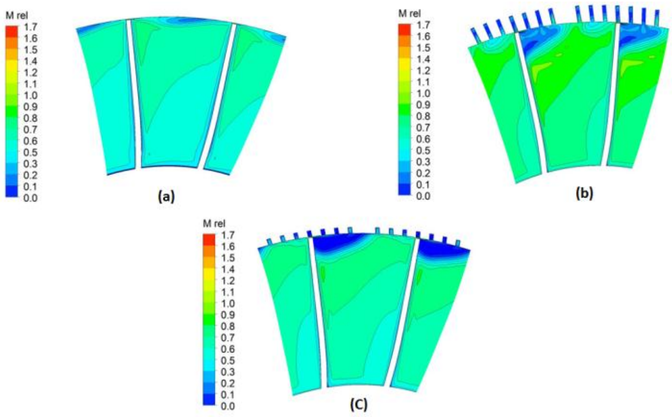

Fig.12 variation of relative mach number on streamwise plane-2 at near design mass flow rate of 21.6 kg/s: : (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

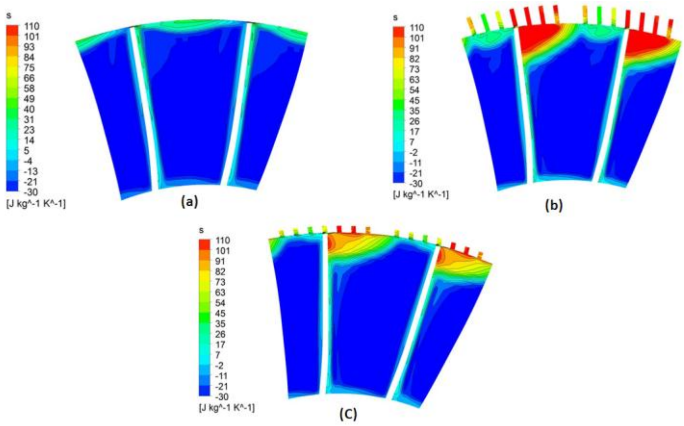

Fig. 13 variation of entropy at streamwise plane-2 at near design mass flow rate of 21.6 kg/s: (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

The variation of relative Mach number on streamwise Plane-2 and at the operating point is shown in Fig.12. The low momentum fluid zone in the mid-passage at the casing

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ISSN: 2278-0181
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endwall is more in the presence of casing treatment slots. This is behavior can be accounted for increase blockage for the compressor core flow which leads increase in entropy and total pressure loss. Figure 13 shows the entropy distribution on Plane-2 at near design operating point (21.6 kg/s mass flow rate). The entropy at the endwall is significantly increased when the blades are incorporated with casing treatment slots. Rotor with CT-1 configuration shows more entropy at tip endwall and hence it has very low pressure ratio compared baseline rotor and rotor with CT-2 configuration.

Similarly Fig. 14 and 15 show the variation of relative Mach number and entropy respectively on the streamwise Plane-2 at a mass flow rate corresponding to the stall mass flow rate of baseline stage, and compressor stages with CT-1 and CT-2 configurations. Referring to Fig. 14(b) and 14(c), the separated flow region near rotor blade suction surface at the casing endwall is more than in case of baseline rotor Fig.14(a). This low momentum fluid zone is more in rotor with CT-1 configuration. Rotor with CT-1 configuration has larger slot depth (11 mm) and appears to be more effective in reducing energy of the fluid at tip region. In case of rotor with CT-2 configuration, slot depth is small (4 mm) and has less contribution in reducing energy of fluid at tip region compared rotor with CT-1. This degradation in the quality of flow in rotor tip region is responsible for the enhancement of total pressure loss for the casing treated rotor.

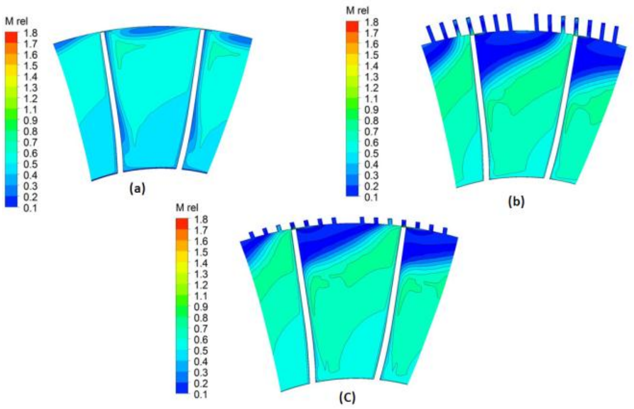

Fig.14 variation of relative mach number at stream wise plane-2 at mass flow rate equal to stall mass flow rate: (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

Figure.15(b) and 15(c) show an overall redistribution of entropy in the streamwise Plane-2 for the casing treated rotors compared to the baseline rotor (Fig.16a). Increase in entropy at rotor tip region is observed in both cases of casing treated stages. Effect of this increased entropy has affected stage performance by reducing total pressure and efficiency in stall region.

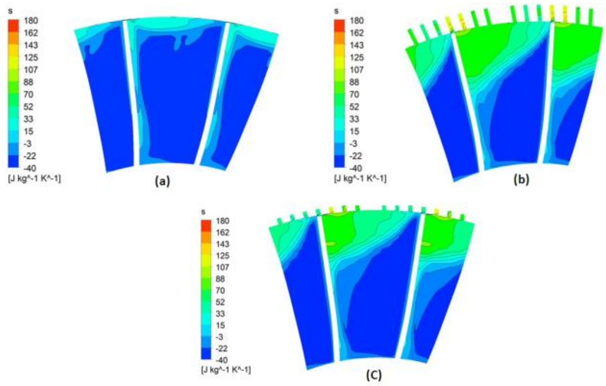

Fig.15 variation of entropy at stream wise plane-2 at mass flow rate equal to stall mass flow rate: (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

FLOW DEVELOPMENT IN BLADE TO BLADE PASSAGE

The effect of casing treatment slots can further be explained by examining the flow quality through the blade passage. The variations of relative Mach number through the blade passage at mid span and near tip section of the rotor are shown in Fig.16, 17, 18 and Fig.19 for mass flow rates corresponding to near design point and stall point of baseline rotor respectively. At near design point (Fig. 16) and mid span location, there change in the flow structure at the trailing end of rotor and stator due to the presence of casing treatment slots is quite evident. Figure.16 (a) and (c) clearly shows presence of low momentum fluid in trailing wake zone of both rotor and stator. In Fig 16 (b) with casing treatment configuration CT-2, low momentum fluid is energized compared to other two cases. The trailing wake width in casing treated rotor (CT-2) and compressing stator is different from baseline stage rotor and stator. Figure 18. shows the mid span relative Mach number variation at stall mass flow rate of baseline stage, stage with CT-1 and CT-2 configuration. Here also it is clearly visible that casing treatment slots has significant effect on trailing wake zone in both rotor and stator blade passage. In case casing of casing treated compressor stages trailing edge wake width of rotor and stator is less than in the baseline stage.

The effect of casing treatment is more pronounced in the tip region. Even at near design mass flow rate Fig.17, there is low energy fluid near the tip of the baseline rotor after mid-chord due to casing and blade boundary layer build up and tip leakage vortex. This retarded flow region is more in case of casing treated compressor stages, stage with CT-2 configuration has more separation towards rotor trailing edge.

At flow rate corresponding to stall mass flow rate of baseline stage, stage with CT-1 case and stage with CT-2 case (Fig.19), the low energy fluid zone in the tip region of the baseline rotor spreads both in tangential and axial direction (upstream), and in blade span direction also. This low energy separated flow is responsible for onset of stall. However, for the stages with casing treatment slots, allowing the mass flow rate to be further reduced with consequent increase in stall

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margin. Even though with more low momentum fluid on, compressor stage with stage with CT-1 configuration increment in stall margin by 8.6%, compared to baseline stage. This is observation is in contradiction to logical assumption that reducing momentum of the fluid leads to flow separation and compressor rotor stall initiates. Above contradicting observation is accounted due to tip leakage vortex formation and interaction with casing and blade surface boundary layer and also secondary flows. Differential fluid pressure across pressure and suction surface of rotor blade causes the tip leakage vortex. Prominent tip leakage vortex formation is observed in case of base line compressor stage without casing treatment, where incase of stage with casing treatment slots core tip leakage vortex is not observed.

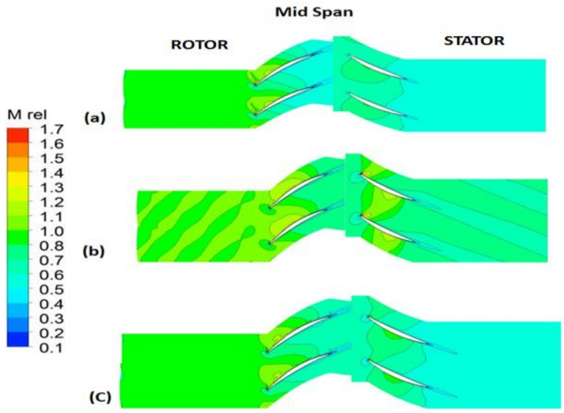

Fig.16 variation of relative mach number on blade-blade planes at midspan at near-design mass flow rate of 21.6 kg/s : (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

Hence presence of casing treatment slots avoids formation of strong tip leakage vortex flow and increases the stall margin compared to baseline sage. Intensity of low momentum fluid in rotor tip region is less in case of stage with CT-2 configuration and therefore stall margin improvement very is small (2.6%) compared to baseline stage.

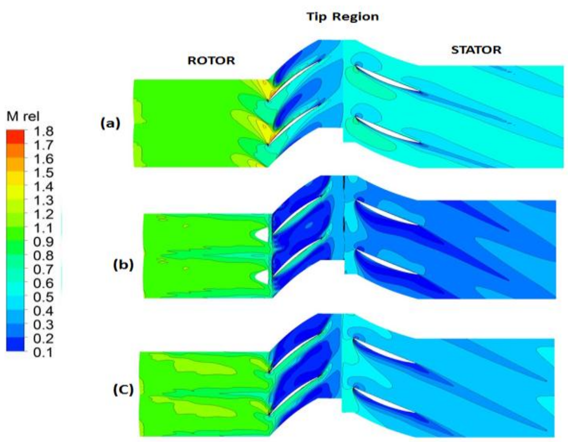

Fig.17 variation of relative Mach number on blade-blade planes near blade tip at near-design mass flow rate of 21.6 kg/s : (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

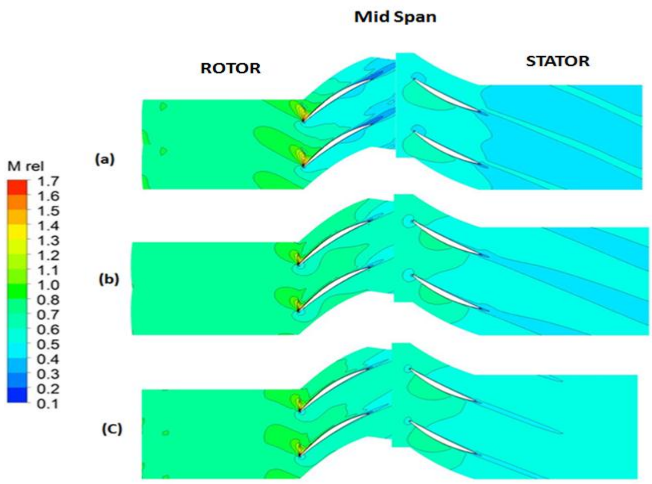

Fig.18 variation of relative mach number on blade-blade planes at midspan at stall mass flow rate: (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

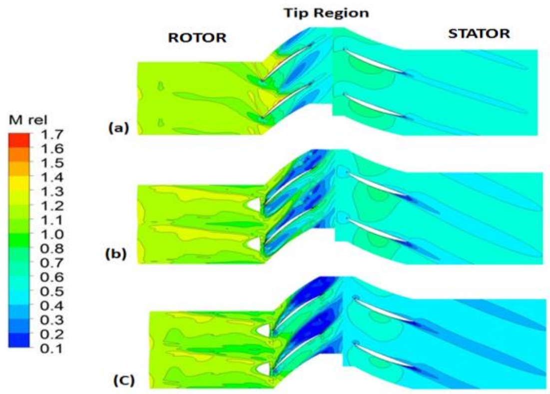

Fig.19 variation of relative mach number on blade-blade planes near blade tip at stall mass flow rate: (a) baseline stage, (b) casing treated stage case ct-1, and (c) casing treated stage case ct-2

Figure. 20 show the tip leakage vortex captured by absolute velocity flow stream lines. In fig.20 (a) tip leakage vortex shape is visible after 15 % of axial of chard length from leading edge of the rotor blade. Flow stream lines in tip region of casing treated rotors Fig. 20(b) and 20(c) has no significant tip leakage vortex formation. Actual tip leakage flow vortex in baseline rotor tip region is captured by plotting swirl velocity component flow stream lines. In Fig.21 (a) strong tip leakage flow formation and vortex growth towards blade passage is clearly visible. In case of stage with CT-1 configuration as shown in Fig.21 (b) it is visible that there is no significant tip leakage flow vortex formation and hence it is attributed to increase in stall margin.

The above discussion on investigation results explains possible reasons for extension of stable operating range of the casing treated compressor stages. As reveled from available literature on transonic axial compressor stage, casing treatment increases stall margin with a penalty of total pressure loss and low isentropic efficiency. It is to be noted that casing treatment slots breakdown strong tip leakage flow vortex which is responsible for onset of stall in transonic compressor rotor paassage. One reason for drop in pressure ratio and efficiency could be that in case of the compressor blades with casing treatment, entropy increases due to recirculation of flow in the narrow slot area from trailing edge to leading edge. The intermixing of the slot passage flow with core flow causes higher pressure losses, thus reflecting in lowering of the total pressure ratio. At the same time, the frictional heat generation due casing treatment slots and consequent temperature rise, decreases isentropic efficiency. It is required to confirm this through further investigations on high pressure ratio compressors with casing treatment.

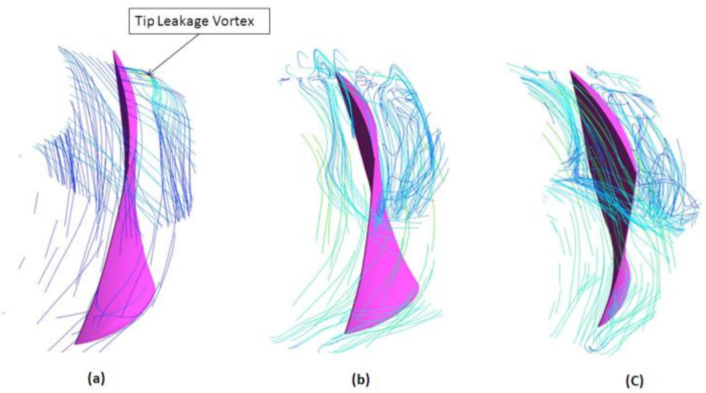

Fig.20 absolute velocity flow stream lines on blade-blade planes near tip region at stall mass flow rate: (a) baseline rotot, (b) casing treated rotor case ct-1, and (c) casing treated rotor case ct-2

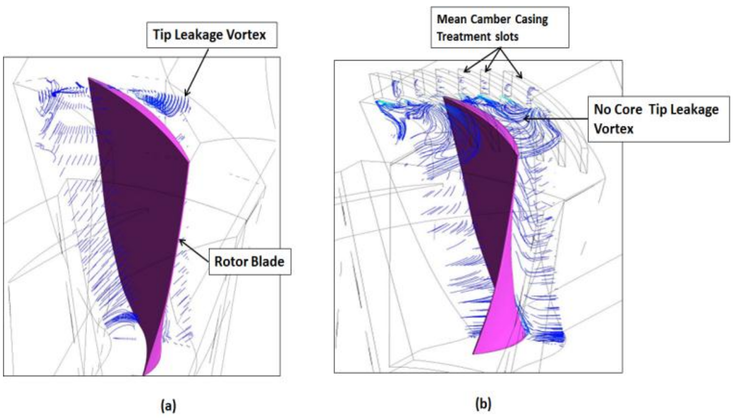

Fig.21 swirl velocity component flow stream lines on blade-blade planes near tip region at stall mass flow rate: (a) baseline rotor, (b) casing treated rotor case ct-1

CONCLUSIONS

Effect of rotor tip mean camber casing treatment slot on performance of transonic axial compressor stage is understood. Numerical investigations have been carried out on a transonic compressor stage without and with two configurations of casing treatment. The performances of the three compressor stages are examined in terms of overall performance and flow development through the blade passages. The important conclusions are:

- In general, the predicted performance of baseline rotor with without casing treatment agrees well with the published experimental data, with some under-prediction of total pressure ratio and isentropic efficiency.
- Casing treatment in transonic axial compressor stage shows significant flow redistribution in rotor passage, in all the three directions, axial, tangential and radial.
- The compressor stage with CT-1 configuration with slot aspect ratio of 5.5 shows lowest pressure ratio and efficiency in the mass flow rate range between design point and stall point. The compressor stall margin is improved by 8.6%.
- The compressor stage with CT-2 configuration with slot aspect ratio of 2 shows better performance in terms pressure ratio and efficiency in the mass flow rate range between design point and stall point. The compressor stall margin is increment is very less 2.6%.
- Casing treatment slots are able to influence the flow in the casing endwall region by suppressing tip leakage flow formation, which improves the stall margin.
- Intermixing of flow from slots with compressor core flow, seems to have increased the pressure loss with a consequent reduction in total pressure ratio. However, a possible temperature rise due to frictional heating reduces the compressor efficiency.
- Casing treatment plays vital role in increasing axial compressor stall margin and contributes to gas turbine engine operating range.

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