

# Sensitivity Study of Stagger Angle on the Aerodynamic Performance of Transonic Axial Flow Compressors



Manjunath S. Dalbanjan and Niranjan Sarangi

**Abstract** Gas turbine engines are widely used in the aviation industries because of high power to weight ratio. The gas turbine engine consists of the three main components: compressor, combustion chamber, and turbine. The design of the axial flow compressor depends on various design parameters such as blade stagger angle, blade height, blade chord, rotor tip clearance, and stator tip clearance. Blade stagger angle is a critical parameter which plays vital role in performance of the gas turbine engines. In this study, a steady state analysis has been carried out to understand the effect of rotor and stator blade stagger angle on the aerodynamic performance of single stage axial flow transonic compressor through three dimensional viscous analysis using ANSYS CFX 19.2 software. The analysis was carried out for various stagger angle configuration varying from  $24^\circ$  to  $36^\circ$  for rotor and  $06^\circ$  to  $18^\circ$  for stator. The study concluded that as the rotor stagger angle is increased from  $24^\circ$  to  $36^\circ$ , the mass flow is reduced by 16.5%, peak pressure ratio by 2.8%, and increase peak efficiency by 6.8%. The effect of change of stator stagger angle from  $06^\circ$  to  $18^\circ$  is insignificant.

**Keywords** Axial flow compressor · Stagger angle · Tip clearance · Surge margin · Computational Fluid Dynamics (CFD) · Pressure ratio · Efficiency

## 1 Introduction

The majority of the gas turbine power plants and particularly in aircraft applications use axial flow compressor. It is a difficult task to achieve high efficiency and a wide operating envelop of compressor without compromising the performance of the compressor. The axial flow compressor performs efficiently and meets the design

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requirement with in the compressor operating envelope. Aerodynamic instability occurs due to adverse pressure gradient in the compressor which will significantly affect the performance of the compressor. Various studies are carried out to understand the effect of blade stagger angle. Schobeiri [1] showed improvement in gas turbine efficiency at off design condition by adjusting the compressor and turbine blade stagger angles. A Gaussian probabilistic density function simulation approach used by Zheng et al. [2] to study the impact of random stagger angle variability on the compressor performance showed that the blades with variation of the stagger angle cause more uneven pressure distribution across span of the blade. Chun et al. [3] considered a low hub to tip ratio transonic compressor to understand the effect of stagger angle on aerodynamic performance. Schnoes et al. [4] studied various stagger angle configuration to design a multi-stage axial flow compressor using optimization technique. The study concluded that the stagger angle effect on the efficiency of the compressor is dominant. Cao et al. [5] have carried out experimental studies on contra-rotating axial flow pump to understand the pressure distribution across span of the blade by adjusting the stagger angle. Myoren et al. [6] concluded that for an axial compressor rotor in transonic stage by variation of stagger angle, the stalling margin can be enhanced using the multi-objective optimization technique. Cho et al. [7] concluded by decreasing the stagger angle at the mean radius and increasing at hub and tip, resulting in improvement of efficiency and optimum pressure. Li et al. [8] describes that the decrease in the stagger angle will reduce the turbulence at suction side of the rotor blade. Oyama et al. [9] developed an optimization technic to obtain the desired stagger angle for the given design parameters to achieve the required aerodynamic performance. Yoon et al. [10] studied the influence of design parameters on tip vortex in a four stage axial flow compressor by considering stagger angle, tip clearance, and flow coefficient. Most of the studies are limited to stagger angle of rotor at design point. Available literature in public domain indicates that substantial work has not been done on the sensitivity of the stagger angle on rotor and stator and its combined effect on aerodynamic performance.

The emphasis of this study is to evaluate the sensitivity of the stagger angle on rotor and stator blades and to investigate its effect on the aerodynamic performance of a single stage axial flow transonic compressor are discussed in detail in this paper.

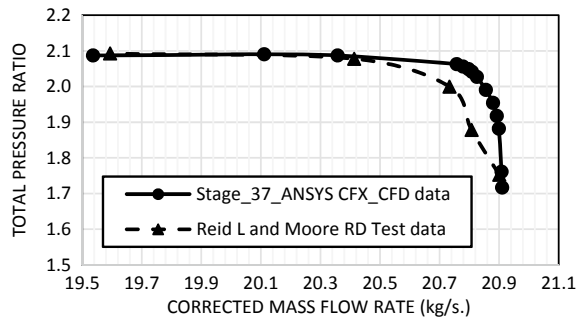
## 2 Methodology

The aerodynamic analysis is carried out to study the sensitivity of blade stagger angle on the aerodynamic performance parameters by using commercially available 3D Computational Fluid Dynamics (CFD) software package. The computational methodology followed for the analysis process is explained below.

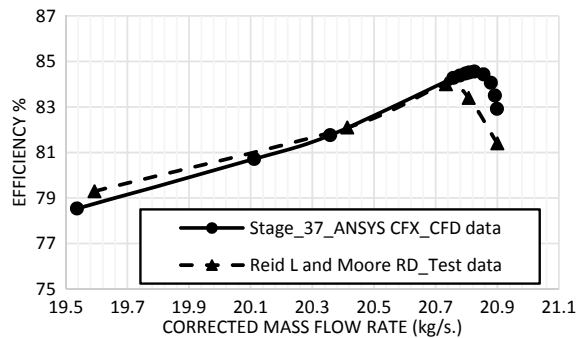
## 2.1 CFD Code Benchmarking Study

Computational fluid dynamics (CFD) has emerged as a most powerful tool for determining the flow physics inside the casing for turbomachinery applications. Before adopting any CFD software, a careful validation should be done by considering a standard test case. A standard transonic compressor test cases NASA stage 37 by Reid and Moore [11] is considered for the assessment of CFD tool. A detailed analysis is done and the predicted performance parameters are compared with the experimental test results of NASA stage 37. The detail CFD code benchmarking analysis is done in previous study by Dalbanjan and Sarangi [12]. The outcome of the validation study showed that the efficiency is slightly higher predicted by CFD tool compare to the published test data. Whereas pressure ratio and mass flow are showing good match within 1%. The validation of predicted and measured performance plots are shown in Figs. 1 and 2.

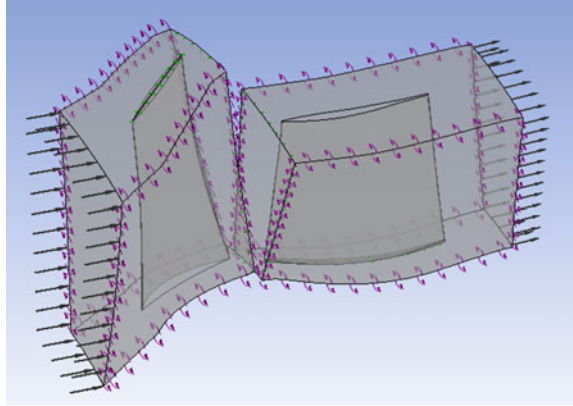
**Fig. 1** Mass flow rate versus pressure ratio for NASA stage 37



**Fig. 2** Mass flow rate versus efficiency for NASA stage 37



**Fig. 3** Geometric model of rotor and stator domain



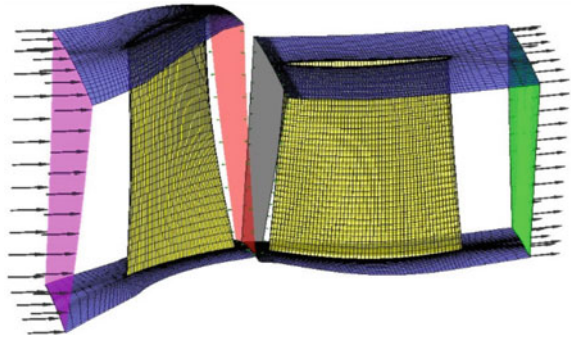
## 2.2 Solid Modeling of Blade Geometry in Flow Domain

The geometry is modeled using 3D modeling software. The geometric model consists of extended inlet domain, rotor blade, stator blade, and extended outlet domain so that the output flow parameters are measured after the flow settles down. Single passage with rotor and stator domain extended up to 1 chord of downstream blade was considered for analysis to reduce the computational time. Figure 3 shows the rotor and stator geometric model.

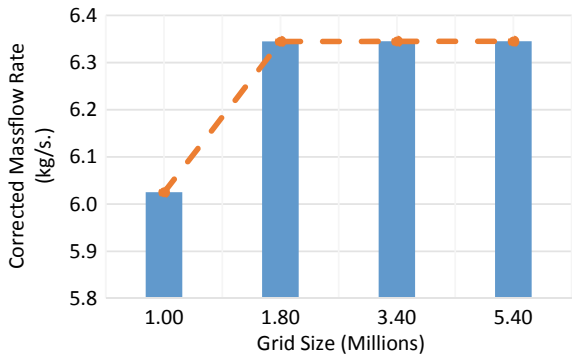
## 2.3 Grid Generation and Grid Convergence Study

A grid independence study was carried out to ensure that the results obtained are independent of the grid size or elements. The mesh topology used for flow domain is H-type and O-type near the rotor and stator blade surface to capture the boundary layer. In the tip region, 10 elements were created and nonmatching grid topology was used with appropriate alignment of interface. Mesh distribution was varied in spanwise, streamwise, and circumferential direction for rotor and stator domains, near the blade surface and casing wall more dense grid was selected to maintain the  $y^+$  value of less than 1. The grid quality requirement was met with minimum and maximum angle being greater than  $20^\circ$  and less than  $165^\circ$ , respectively. The advection scheme and high resolution turbulence numeric were selected along with the  $k-\omega$  shear stress transport (SST) turbulence viscosity model. The  $k-\omega$  SST model accounts for the transport of turbulent shear stress and gives highly accurate predictions of the onset and the amount of flow separation under adverse pressure gradients. Mesh for both rotor and stator stage domain were generated as shown in Fig. 4. Four different cases by varying number of grids in rotor and stator blades were considered. Figures 5 and 6 show the plots of performance parameters like corrected mass flow and pressure ratio with respect to grid size. It is observed from Figs. 5 and 6 that the grid-2 onwards

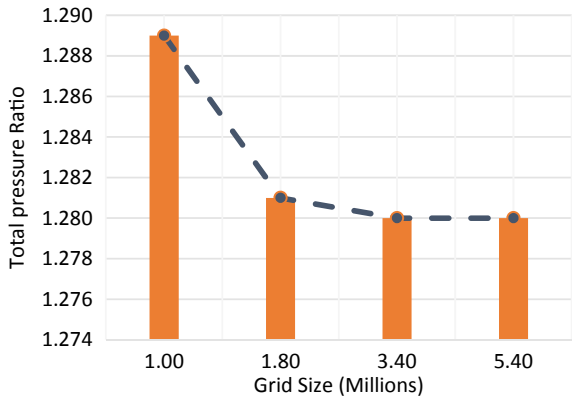
**Fig. 4** Mesh domain for rotor and stator



**Fig. 5** Comparison of corrected mass flow rate for various grid configuration



**Fig. 6** Comparison of total pressure ratio for various grid configuration



the variation in parameters are insignificant. Thus, for further analysis grid-3 was selected with grid size of 3.40 million elements. Table 1 shows the performance parameters for different grid sizes.

**Table 1** Comparison of all four grid cases at 100% speed

Cases	Grid Size (Millions)	Corrected mass flow (kg/s.)	Total pressure ratio
Grid-1	1.00	6.0251	1.289
Grid-2	1.80	6.3451	1.281
Grid-3	3.40	6.3453	1.280
Grid-4	5.40	6.3454	1.280

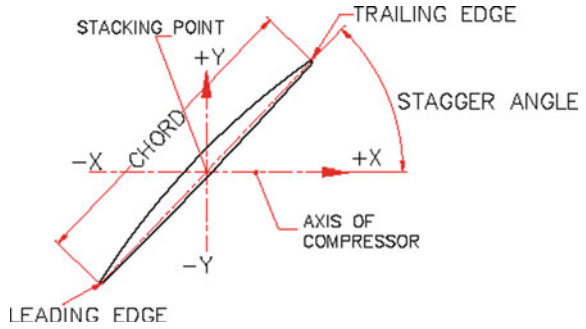
## 2.4 Boundary Condition

The appropriate boundary conditions were applied for the flow domain, at inlet of rotor stagnation pressure and stagnation temperature. Stage mixing model as an interface between rotor and stator was selected. Mixing plane calculates the circumferentially averaged flow parameters from the rotor and passes to the inlet of the stator. Static back pressure was applied at the exit of stator. For blade, hub and casing adiabatic and no-slip wall boundary condition was specified. The numerical simulation was performed using multi-block grid with pressure-based implicit solver ANSYS CFX 19.2 software. Constant speed characteristics at design speed were generated by gradually increasing the Static back pressure at the stator outlet. After reaching the peak efficiency point, the static back pressure was increased in the smaller steps of 1 kPa to capture the near stall point and to obtain a converged solution. The residual convergence level for all the parameters is specified as  $1 \times 10^{-6}$  to get more accurate solution.

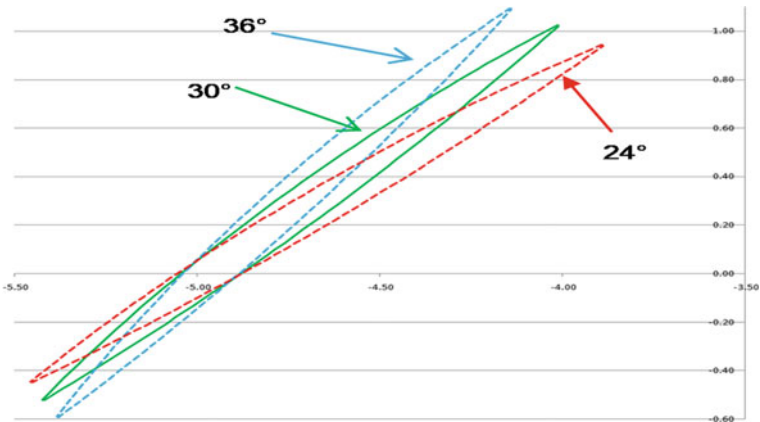
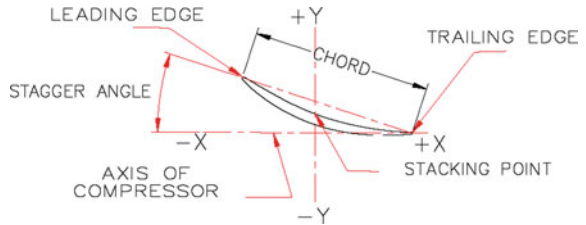
## 2.5 Solution Methodology

The study involves variation of stagger angle for rotor. In the first phase, the analysis is performed by keeping the stator stagger angle at  $12^\circ$  which is obtained from the design and varying the rotor stagger angle from  $24^\circ$  to  $36^\circ$  with the increment of  $1^\circ$  at the design speed. The optimum rotor stagger angle is selected as  $30^\circ$  obtained from the analysis which is meeting the design mass flow requirement. The second phase involves variation of stator stagger angle from  $06^\circ$  to  $18^\circ$  at the increment of  $1^\circ$  by keeping the rotor stagger angle constant. Figures 7 and 8 show the rotor and stator stagger angle with respect to axis of the compressor. Figure 9 shows the variation of rotor stagger angle at the mid span of the blade for  $24^\circ$ ,  $30^\circ$ , and  $36^\circ$ , and Fig. 10 shows the variation of stator stagger angle at the mid span of the blade for  $06^\circ$ ,  $12^\circ$ , and  $18^\circ$ .

**Fig. 7** Rotor stagger angle with respect to axis of the compressor



**Fig. 8** Stator stagger angle with respect to axis of the compressor



**Fig. 9** Variation of rotor stagger angle

### 3 Results and Discussion

The simulation was carried out at design speed, and the constant speed characteristics were generated for various stagger angle configuration for rotor and stator.

Figure 11 shows the corrected mass flow rate versus total pressure ratio for various combination of stagger angle configuration ranging from 24° to 36° for rotor blade. It can be observed that the increase in stagger angle from 24° to 36° the mass flow

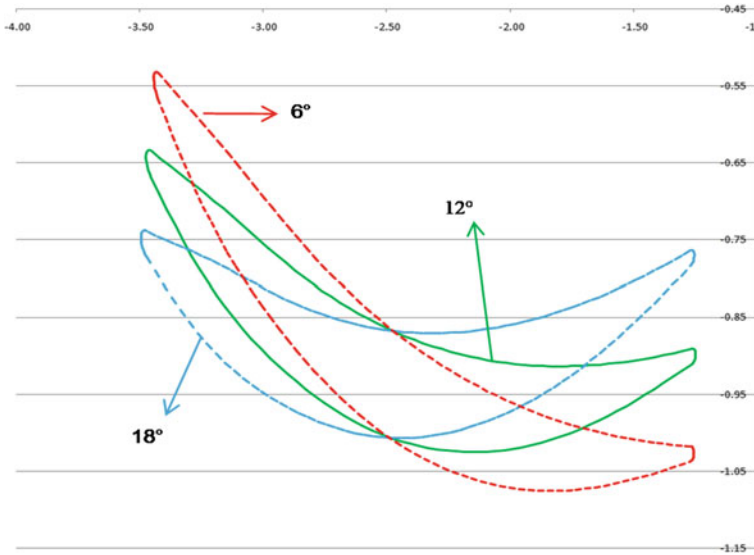


Fig. 10 Variation of stator stagger angle

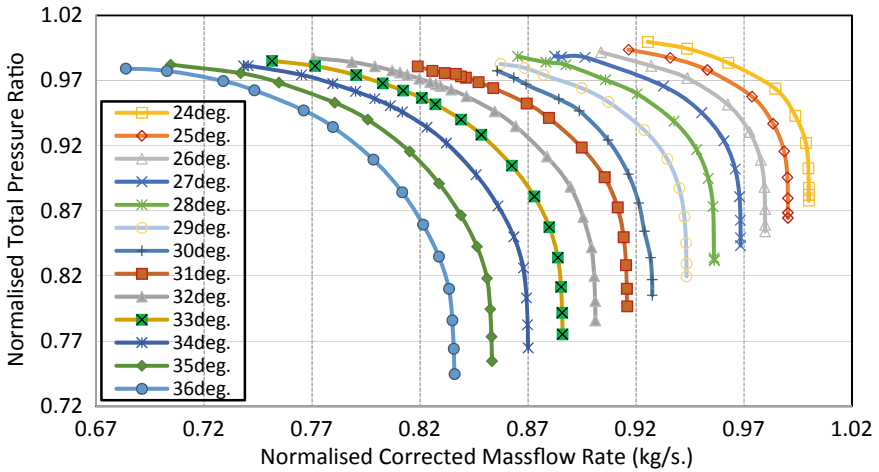
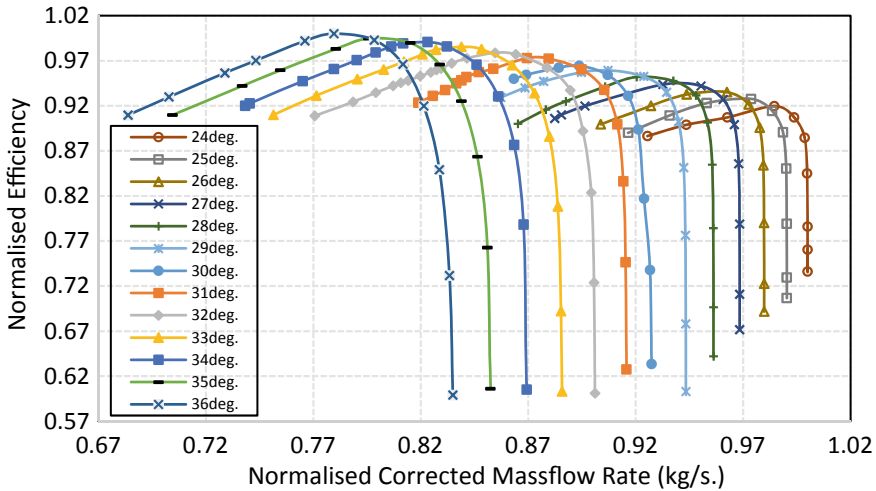


Fig. 11 Normalized corrected mass flow rate versus total pressure ratio for rotor

swallowing capacity is reduced with slight drop in pressure ratio. The mass flow is reduced by 16.5% and peak pressure ratio is reduced by 2.8% when stagger angle is increased from 24° to 36°. This shows that the mass flow is more sensitive to rotor stagger angle. The plot also shows the zone of maximum and minimum mass flow for the configuration under consideration. It provides the ready reference for selection of rotor stagger angle to achieve the design mass flow.





**Fig. 12** Normalized corrected mass flow rate versus efficiency for rotor

Figure 12 shows the corrected mass flow rate versus efficiency for various combination of stagger angle configuration ranging from  $24^\circ$  to  $36^\circ$  for rotor blade. It is observed that as the stagger angle is increased from  $24^\circ$  to  $36^\circ$  the efficiency is increased with drop in mass flow. The mass flow is reduced by 16.5% and peak efficiency is increased by 6.8 points when stagger angle is increased from 24 to  $36^\circ$ . The lower mass flow is due to the reduction in blade passage area as the stagger angle is increased and the increase in efficiency is due to the absence of separation zone at the rotor outlet. This shows that the rotor efficiency is sensitive to the stagger angle. The plot also shows the zone of maximum and minimum peak efficiency for the configuration under consideration. It provides the ready reference for selection of rotor stagger angle to achieve the design efficiency.

Figure 13 shows the corrected mass flow rate versus total pressure ratio for various combination of stagger angle configuration ranging from  $06^\circ$  to  $18^\circ$  for stator blade. It is observed that there is no significant variation of mass flow and pressure ratio due to variation in stagger angle of stator blades from  $06^\circ$  to  $18^\circ$  and the loss in the performance parameter is negligible.

Figure 14 shows the corrected mass flow rate versus efficiency for various combination of stagger angle configuration ranging from  $06^\circ$  to  $18^\circ$  for stator blade. As the stagger angle is increased from  $06^\circ$  to  $18^\circ$ , the efficiency is decreased with minimal variation in mass flow. The peak efficiency is decreased by 3.5 points when stagger angle is increased from  $06^\circ$  to  $18^\circ$ . As the stator stagger angle is increased, the losses are higher due to flow separation at the pressure side near the trailing edge of stator which results in lower efficiency.

Figure 15 shows the blade loading plot for rotor at mid span for various stagger angle configuration. It is observed that increasing stagger angle improves the static pressure at the expense of degradation in blade performance. Rotor is performing

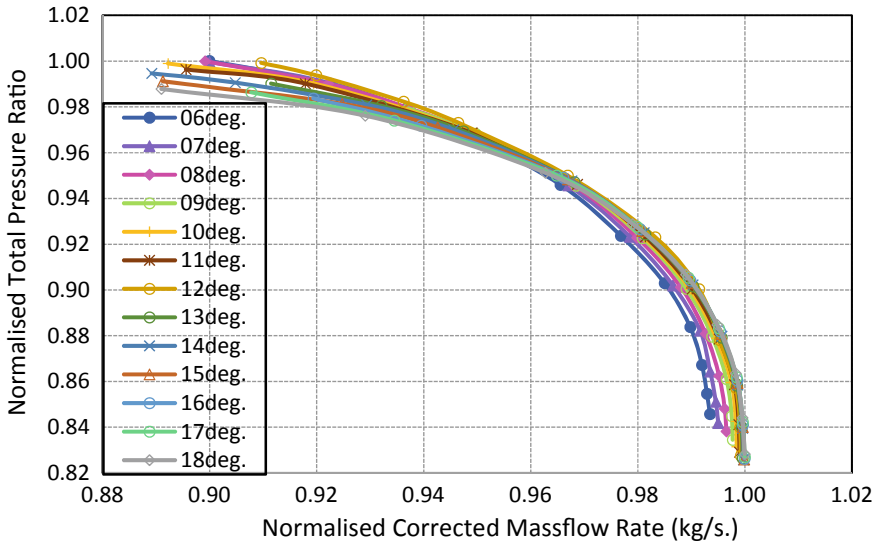


Fig. 13 Normalized corrected mass flow rate versus total pressure ratio for stator

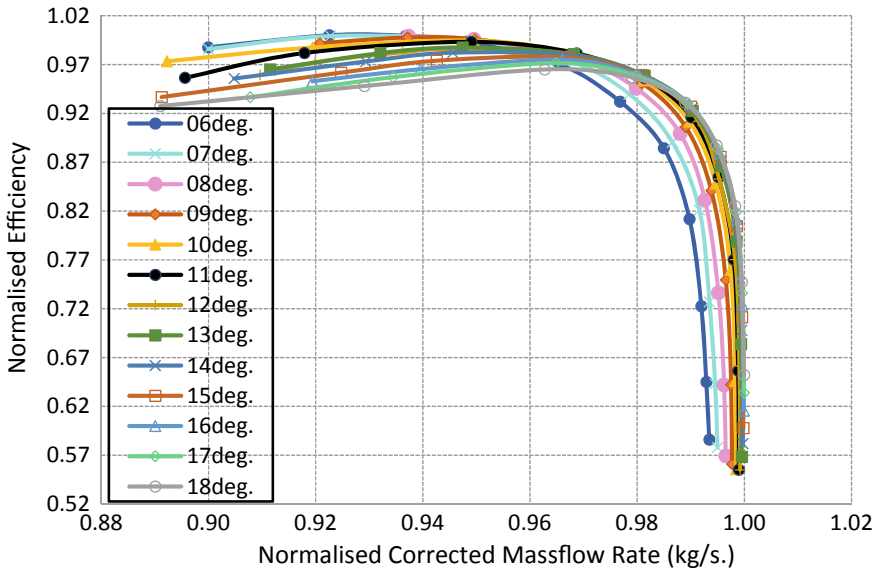
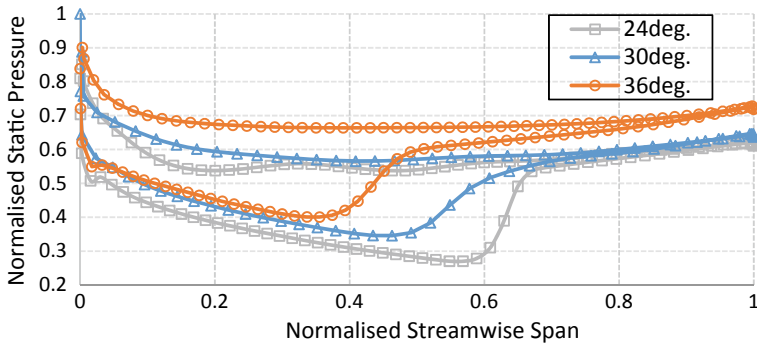
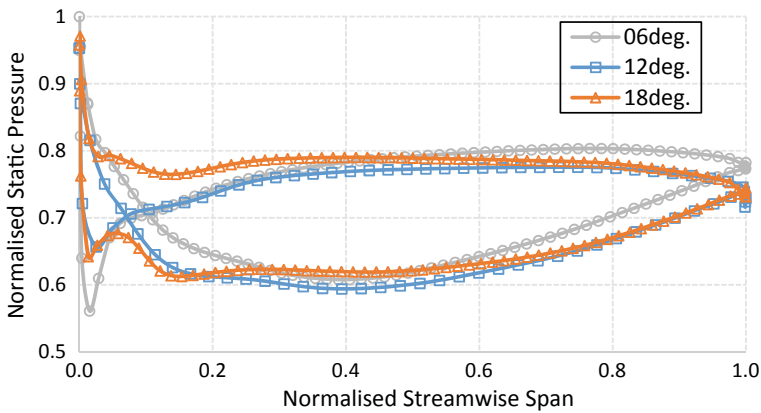


Fig. 14 Normalized corrected mass flow rate versus efficiency for stator



**Fig. 15** Blade loading for rotor



**Fig. 16** Blade loading for stator

better up to 60% of span when stagger angle is  $24^\circ$  but as stagger angle is increased to  $36^\circ$  there is a gain of 15% in static pressure at the expense of spanwise degradation in blade loading.

Figure 16 shows the blade loading plot for stator at mid span for various stagger angle configuration. It is observed that the effect of stagger angle is not sensitive on the blade loading parameters of stator. The stator is performing as per design requirement.

## 4 Conclusions

A 3D steady state viscous CFD analysis is carried out to investigate the sensitivity of stagger angle on the aerodynamic performance of a transonic single stage axial

flow compressor using commercial code ANSYS CFX 19.2 software. The analysis is carried out for various stagger angle configuration varying from  $24^\circ$  to  $36^\circ$  for rotor and  $06^\circ$  to  $18^\circ$  for stator. It is observed that as the rotor stagger angle is increased from  $24^\circ$  to  $36^\circ$ , the mass flow is reduced by 16.5%, peak pressure ratio is reduced by 2.8% and peak efficiency is increased by 6.8 points. It is interesting to see that the stator is not sensitive to the stagger angle and the effect is insignificant when the stator stagger angle is varied from  $06^\circ$  to  $18^\circ$ . The study concludes that the rotor stagger angle is highly sensitive for controlling the mass flow and efficiency compare to stator stagger angle. The data generated in this study provides the ready reference for selection of rotor and stator stagger angle for meeting the requirements of aerodynamic design parameters such as mass flow and efficiency.

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