Numerical Study on the Effects of Dihedral Stators in a Transonic Axial Compressor

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This paper presents the effect of dihedral stators on flow behavior in a transonic axial compressor. A commercial flow solver was used to calculate the performance and flow characteristics of a transonic axial compressor with different shapes of stators modified by changing the shape of the stacking line. In a stator with a straight stacking line, large corner separation occurred between the suction surface and the shroud endwall and caused a large total pressure loss under high loading. In this study, the dihedral stator induced a significant pressure loss on the blade at the peak efficiency point owing to the increased corner separation area. The shroud dihedral generated a radial pressure gradient and caused a low-momentum flow to migrate from the shroud endwall toward the midspan, consequently decreasing the diffusion factor near the shroud endwall. Although the hub dihedral generates unexpected hub-corner separation causing a large total pressure loss over the entire operating range, the loss region makes the high-momentum flow near the hub divert toward the upper part of the passage. Therefore, the amplitude of the low-frequency term according to the shroud-corner separation also decreased, and the stall limit of the compressor was improved with the hub dihedral stator.

Key Words: Transonic Axial Compressor, Corner Separation, Dihedral Stator, Diffusion Factor, Fast Fourier Transform

Nomenclature

- m: corrected mass flow rate
- P: pressure
- U: rotating velocity
- W: relative velocity
- ω: yaw angle
- γ: pitch angle
- θ: circumferential component
- ρ: density
- σ: solidity
- ωo: total pressure loss coefficient

Subscripts
- s: static
- t: total
- LE: leading-edge
- TE: trailing-edge
- 1: rotor inlet
- 2: rotor outlet
- 3: stator outlet

Superscripts
- : pitchwise mass average
- =: overall mass average
- a: area average
- m: mass average

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1. Introduction

A transonic axial compressor with a high pressure ratio, high efficiency and sufficient stall margin is commonly desirable for gas turbines in aircraft and industrial applications. To improve the performance of state-of-the-art compressors, internal flow phenomena such as wake, separation, and tip leakage flow must be understood. In the stator of a transonic axial compressor, the hub-corner separation and tip leakage flow of the upstream rotor during low-mass flow result in non-uniform distribution of spanwise loading. In particular, corner separation, which occurs frequently between the endwall and the blade suction surface, dominates the secondary flow in compressor passages. Corner separation is a three-dimensional phenomenon that causes substantial pressure loss, creates a blockage in the flow, and limits the operating range of the compressor. Therefore, suppressing or delaying corner separation can improve the performance of axial compressors.

Three-dimensional blading techniques, such as the sweep and dihedral methods, are effective for reducing corner separation and other detrimental flow phenomena that degrade compressor performance, so many researchers have attempted to find an optimal blade shape. Use of dihedral blades in a compressor or turbine can improve the stability of an aircraft. A dihedral blade in a compressor controls the radial flow distribution inside the passage. In a linear compressor cascade, a positive dihedral blade reduces endwall loading but increases total pressure loss at midspan, and the increased pressure gradient in the spanwise direction induced by the dihedral blade effectively suppresses corner separation. In highly loaded transonic axial compressors,
the dihedral blade can control the three-dimensional flow: the dihedral causes unloading near the endwall and flow redistribution in the spanwise direction; this off-loading effect of the dihedral blade is more powerful in blade rows rather than classical secondary flow. A bowed blade with a double-sided dihedral at both hub and shroud endwalls maximizes the dihedral effect and can reduce endwall loss and flow blockage at a low-mass flow rate. The overall static pressure ratio of a multistage compressor with bowed stators increases due to the reduction in corner separation when the compressor is highly loaded.

In the present study, the stator geometry was modified by changing the stacking line to determine the effect of the dihedral star in one-stage transonic axial compressor performance. Flow quantities such as diffusion factor and stall margin were evaluated, and the transient characteristics of corner separation were analyzed using the Fast Fourier Transform (FFT) through steady and unsteady flow solvers, respectively.

2. Test Configuration

2.1. Geometric specifications

A numerical study on the effects of dihedral stators in the performance of an axial compressor was conducted using a NASA Stage 37 transonic axial compressor. This single-stage compressor was designed as part of a program to design the inlet stage core compressor for an aircraft engine. The geometrical configuration and measuring planes where the flow quantities were obtained are shown in Fig. 1. Detailed specifications of the compressor model are given in Table 1.

![Flow path and measurement positions of the Stage 37 compressor.](image)

**Table 1.** Design specifications of NASA Stage 37 compressor.

| Parameters               | Rotor                      | Stator                      |
|--------------------------|----------------------------|-----------------------------|
| Design mass flow rates   | 20.188 (kg/s)              | 20.188 (kg/s)               |
| Rotational speed, rpm    | 17185.7                    | 17185.7                     |
| Tip speed, m/s           | 454.136                    | 454.136                     |
| Tip clearance, cm        | 0.0356                     | 0.0356                      |
| Aspect ratio             | 1.19                       | 1.26                        |
| Solidity                 | 1.75(hub)/1.3(tip)         | 1.47(hub)/1.3(tip)          |
| Number of blades         | 36                         | 46                          |

Fig. 1. Flow path and measurement positions of the Stage 37 compressor.

2.2. Dihedral stators

The blade stacking line passes through the mass center of the two-dimensional blade section at each span, and it is usually straight. Gümmerr and Wenger presented that pure sweep is obtained by leaning the stacking axis only in the direction of the chord, and pure dihedral is obtained by moving the endwall sections perpendicular to the chord direction in the direction of lift. In the present study, the dihedral was imposed by bending or tilting the stacking line of the blade only in the circumferential direction because the axial gap of Stage 37 between rotor trailing-edge and stator leading-edge is too small.

The objective of this study was to evaluate the effects of a dihedral stator in a transonic axial compressor so that only the stacking line changed while the cross-section at each span remained unchanged. It is generally known that a positive dihedral with an obtuse angle between the suction surface and the endwall is more advantageous than a negative dihedral with an acute angle. In the present study, the dihedral stators were characterized by a dihedral angle of 30 and dihedral depth of 20% for span, as shown in Fig. 2. Four test cases with different stators were tested while leaving the other part of the stacking line in the core flow region as a radial straight line. The stators have a shape among the reference, hub dihedral (hub endwall only), shroud dihedral (shroud endwall only) and bowed blade. The shape of the bowed blade is defined by applying a double-sided dihedral to the stacking line at both hub and shroud endwalls.

3. Numerical Procedure and Computational Grid

A commercially available solver, ANSYS CFX 13.0, was employed to calculate the performance and analyze the internal flow-field of the compressor stage. In this code, three-dimensional compressible Reynolds-averaged Navier-Stokes equations were solved using a finite volume method for unstructured grids. The implicit time-stepping scheme was used for the steady and unsteady simulations. The working fluid was considered to be an ideal gas, and the flow inside the compressor was assumed to be fully turbulent. The k-ω SST turbulence model was used to obtain the turbulent viscosity and precisely predict the separation induced by the adverse pressure gradient. The converged simulation data...
was obtained when the residual was smaller than $10^{-6}$ and constant for a sufficient amount of time. The target compressor consisted of a rotor with 36 blades and a stator with 46 blades. In the present study, the number of stator vanes was increased to 48 while keeping the stator solidity constant. Therefore, the ratio of rotors to stators was consequently set to 3:4 with a 30° pitch angle for each computational domain.

Figure 3 shows the computational grid generated using ANSYS TurboGrid. The computational domain was composed of hexahedral unstructured meshes with the growth rate in a wall-normal direction of 1.6; an O-type grid was used near the blade surface, and H/J-type grids were used elsewhere. As shown in Fig. 4, the adiabatic efficiency and total pressure ratio at the peak efficiency point vary only slightly with the variation in grid size being approximately 1- to 2-million. Thus, a total number of about 1,500,000 grids was determined to define the three-rotor/four-stator flow passages, where the single rotor and single stator have 254,589 and 171,595 nodes, respectively. To calculate the tip clearance flow, 12 nodes in the spanwise direction were inserted into the tip clearance. Consequently, the non-dimensional wall distance ($y^+$) was checked below 5 in a simulation result.

As boundary conditions, the total pressure and total temperature were specified at the inlet of the rotor, and the circumferentially-averaged static pressure at midspan was applied at the outlet of the stator. To get results over the whole operating range from choke to stall, both steady and unsteady simulations were conducted by increasing the outlet static pressure step-by-step. The operating range was calculated until a converged solution could no longer be obtained. The adiabatic no-slip condition was applied to the solid wall boundary, and the periodic condition was used in the circumferential direction of the domain. At the interface between the rotor and stator passages, the mixing plane was used for steady simulations with all three-rotor/four-stator passages, but the sliding plane for unsteady simulations.

4. Results
4.1. Code validation
To validate the calculation results, measurements were compared with numerical results for the reference compressor geometry with a straight stator at 70, 90, and 100% design speeds (Fig. 5). The X-axis shows the normalized mass flow rate by the choking mass flow rate at 100% design speed. The flow quantities used in calculating the total pressure ratio and adiabatic efficiency were measured at the rotor inlet (Station 1), rotor outlet (Station 2), and stator outlet (Station 3). Although the total pressure ratio at 100% design speed is
slightly underestimated, the numerical results agree well with the measurements regarding the overall performance tendency.

To determine the flow characteristics of the reference compressor, the total pressure losses of the rotor and stator outlets (Stations 2 and 3) were obtained under different operating conditions with steady calculations, as shown in Fig. 6 and Fig. 7. The rotary total pressure loss coefficient (Eqs. (1) and (2)) was used downstream of the rotor to investigate the flow behavior without any rotational effect, and the total pressure loss coefficient (Eq. (3)) was calculated downstream of the stator.

\[
P_{t,rot} = P_t + \frac{1}{2} \rho (W^2 - U^2) \quad (1)
\]

\[
\omega_{rot} = \frac{\bar{P}_{t,1} - P_{t,rot}}{0.5 \rho U_p^2} \quad (2)
\]

\[
\omega = \frac{\bar{P}_{t,2} - P_{t,3}}{\bar{P}_{t,2} - \bar{P}_{t,3}} \quad (3)
\]

At the peak efficiency point (normalized mass flow rate = 0.99), the flow in the rotor lost considerable energy due to the interaction between the shock and the tip leakage vortex; the flow in the stator lost most of its energy in the upper part of the passage because downstream of the rotor directly affected the flow characteristics inside the stator passage. However, as the mass flow rate decreased (near stall, normalized mass flow rate = 0.93), the loss in the rotor increased in the circumferential and spanwise directions covering about 10% of the span from the casing endwall (Fig. 6(b)). The corresponding blockage increased the inlet flow angle at the leading-edge of the stator. Consequently, tremendous loss occurred in the upper part of the suction surface of the stator due to shroud-corner separation (Fig. 7(b)).

The hub-corner separation causes total pressure loss and decreases stability in the compressor passage. However, the hub-corner separation in a Stage 37 transonic axial compressor is so weak that it is mixed with the main flow even near stall in the reference case.

4.2. Effects of dihedral stators on performance

To investigate the effect of a dihedral stator on compressor performance, a loss analysis was conducted at each rotor and stator passage. Figure 8 illustrates the mass-averaged rotary total pressure loss coefficient of the rotor for four different dihedral stators at the peak efficiency point and near stall. The analysis predicts that rotary total pressure loss increased from the peak efficiency point to near stall as shown in Figs. 6 and 7. The dihedral stators, however, have the same level of rotary pressure loss in spite of different stator geometries. Furthermore, the tip leakage mass flow rate, which is not presented here, also shows the same trend. So it can be concluded that the dihedral stators have no effect on either flow characteristics or performance in the upstream rotor.

Figure 9 shows the effect of a dihedral stator on stator performance. Like the performance curves, the mass-averaged total pressure loss coefficient of the stator is plotted with respect to the mass flow rate for four different dihedral stator cases. As shown in the figure, the dihedral causes an increase in the total pressure loss of the stator. The hub dihedral generates larger total pressure loss than the reference case over the whole operating range, but it has a wider operating range to a lower mass flow rate than the reference case. The details of the operating range and loss will be discussed in the next chapter. When the shroud dihedral was used, the minimum total pressure loss increased, but at the low-mass flow rate, the total pressure loss became the same as the reference case. These results agree with those of Sasaki and Breugelmans, who showed that the dihedral reducing the load on the endwall increases the load at the midspan and therefore increases airfoil loss. At a low-mass flow rate, the pressure gradient in the spanwise direction increased, and the loss degradation maintained a balance between the gain at the endwall and the loss at the midspan. Therefore, the loss trend was similar to the reference case. When the bowed stator was used, the
Total pressure loss increased as in the hub dihedral: even though the bowed stator was influenced by both hub and shroud dihedrals, the hub dihedral has a more dominant effect on compressor performance than the shroud dihedral.

4.3. Steady flow analysis

According to the performance results, the effects of the dihedral on internal flow and loss generation depend on the dihedral position. To investigate a loss development and effect of flow distribution inside the stator passage precisely, the axial evolution of pitchwise-averaged loss was calculated.

As shown in Fig. 10, for the loss development in the reference case, hub dihedral and shroud dihedral were compared at particular span heights, 5%, 50% and 95%, near stall. The amount of loss was obtained from the loss difference between the inlet and each axial position (Eqs. (4) and (5)).

\[
\bar{\omega}_{\text{axial}} = \frac{\bar{\omega}_{\text{each axial plane}} - \bar{\omega}_{\text{inlet}}}{\bar{\omega}} \text{ each span height} \quad (4)
\]

\[
\bar{\omega}_{\text{overall}} = \frac{\bar{\omega}_{\text{each axial plane}} - \bar{\omega}_{\text{inlet}}}{\bar{\omega}} \quad (5)
\]

In the reference case, the loss at 50% and 95% span was larger than the overall loss and this implies that corner separation (Fig. 7(b)) is a major source of reduction in performance. The loss at 95% span rapidly increased and reached its maximum value at 20% of the axial chord, where corner separation was initiated. Thereafter, the loss decreased slowly because corner separation caused spanwise flow transfer from the shroud endwall toward the midspan. Therefore, the loss at 50% span increased until the trailing-edge, and the difference between 50% span and the overall loss decreased downstream of the trailing-edge due to the spanwise mixing. The loss at 5% span indicates the presence of high-energy fluid flowing in the lower part of the stator passage.

The two dihedral stators have different effects on pressure loss. The hub dihedral stator caused a large loss at 5% span, as indicated by the triangular symbols on the dashed line. The shroud dihedral stator increased the loss at 95% span, as indicated by the gradient symbols on the dotted line, and caused the onset point of corner separation to move upstream. Therefore, it is concluded that the dihedral stators locally generate additional loss at the location in which the dihedral is applied, and thereby decreases performance compared to the reference case. The entropy distribution downstream of the stator near stall (Fig. 11) supports this argument. Additional loss due to the hub-corner separation, which was absent in the reference case, occurred near the hub in the hub dihedral case. The size of the high entropy region, which is the result of shroud-corner separation, was larger than the reference case in the shroud dihedral case.

Even with different dihedral configurations, the relative overall loss distributions indicated by the black diamond symbols were indistinguishable from one another in Fig. 10. This observation suggests that a dihedral affects spanwise flow redistribution by reducing the load near the endwall rather than by reducing the axial loss itself. To quantify the effect of the dihedral on flow distribution, the Lieblein diffusion factor (DF) was calculated using the flow properties near stall (Fig. 12).

Generally, DF = 0.6 is a critical value, which is thought to
The corresponding shroud loss was subsequently suppressed at the 20% chord plane where corner separation initiates. As shown in Fig. 14, the signal of the reference case is composed of approximately two specific frequencies. To figure out the oscillating characteristic, each of the signal peaks was numbered. First, numbers 1 and 3 show the minor frequency, which has the smallest amplitude on the signal. It was recorded as 10,313 Hz, and it represents the rotor blade passing frequency (BPF). Second, numbers 1 and 2 show the major frequency, which has the biggest amplitude in the signal. It was recorded as 1,473 Hz, which has a lower frequency than BPF. At the neighbor passage, the dashed line shows a similar pattern to the solid one, so numbers 1 and 2 show the minor frequency, which has the smallest amplitude on the signal. It was recorded as 10,313 Hz, and it represents the rotor blade passing frequency (BPF). Second, numbers 1 and 2 show the major frequency, which has the biggest amplitude in the signal. It was recorded as 1,473 Hz, which has a lower frequency than BPF. At the neighbor passage, the dashed line shows a similar pattern to the solid one, so numbers 1 and
4 mean the speed of the low-frequency term. Consequently, it can be assumed that the speed of the low-frequency term depends on the number of the stator passage because it has four peaks at one cycle.

The major oscillating term has a lower frequency and higher amplitude than BPF in respect to the blockage. To figure out the low-frequency term precisely, the transient flow characteristics for four different dihedral stators were analyzed by FFT based on unsteady axial velocity at Station 3. The axial velocity signal was measured at the 10% and 90% span positions, respectively.

As shown in Fig. 15, the reference case has two peaks at 1,473 Hz and 10,313 Hz regardless of the span position, and the harmonic frequencies were also observed over the frequency ranges. The amplitude of the low-frequency term in the reference case is similar to BPF at 90% span, but it is smaller than BPF at 10% span. In other words, the low-frequency term, which dominates in the upper part of the passage, represents the unsteadiness of the shroud-corner separation. The transient characteristics of the corner separation are also detected at the hub and shroud dihedral stators. The low frequencies for the hub dihedral (dashed) at 10% span and the shroud dihedral (dotted) at 90% span are found to be shifted to a higher frequency compared to those for the reference case. The amplitude of the low frequency significantly increases in the shroud dihedral and bowed blade (dash dot) at 90% span. Consequently, the dihedral stimulates corner separation and additional loss at the location where the dihedral is mounted, so the corresponding unsteadiness increases at higher frequencies and larger amplitudes. The low frequency, however, obviously decreases at 90% span in the hub dihedral stator, and it even has a smaller amplitude in the bowed blade than the shroud dihedral. This means that the unsteadiness of shroud-corner separation is alleviated by applying the hub dihedral, as described in the steady calculation.

To get precise transient characteristics inside the stator passage, the numerical probes for stagnation pressure are along the blade chord away from the blade suction surface with about 10% pitch at the 90% spanwise position. As shown in Fig. 16, the spectra of all points are drawn in one contour plot with respect to their streamwise position. The abscissa, ordinate, and color represent the frequency, streamwise position, and logarithm of amplitude of the total pressure. A low-frequency term that is closely related to the shroud-corner separation is clearly visible, as well as the blade passing frequency near the shroud region. The low-frequency term dominates the whole stator passage regardless of the streamwise position. Although the hub dihedral has as high amplitude in the low-frequency term near the hub region, like the reference case, it consequently suppresses the shroud-corner separation, so the corresponding transient characteristics of the hub dihedral were subsequently alleviated and the flow redistribution was satisfied. These results illustrate the enhancement and extension of the operating range when using a hub dihedral stator.

5. Conclusions

Numerical simulations were conducted to study the effects of dihedral stators on the flow characteristics and performance in a NASA Stage 37 transonic axial compressor. Three-dimensional steady and unsteady computations were performed for four different stator geometries by changing the stacking line. The following results were obtained:

1. The steady calculation shows that the dihedral stators caused an additional loss inside the stator passage, but the dihedral stators had no effect on rotor performance. The total loss of the shroud dihedral slightly increased at the peak efficiency point. Although the extent of the low-loss region increased and the onset point of the shroud-corner separation moved upstream near stall, the total loss was almost the same as in the reference case because a spanwise pressure gradient forced a low-energy flow from the shroud endwall toward the midspan region.

2. The hub dihedral induced unexpected hub-corner separation inside the stator passage, the performance of the compressor dropped significantly over the whole operating range. However, the hub dihedral diverted the high momentum flow
from the hub region to the shroud endwall and subsequently suppressed the shroud-corner separation. This spanwise redistribution consequently reduced the diffusion factor above 30% span and expanded the operating range to a lower mass flow rate. The bowed blade also made the flow redistribute in the spanwise direction.

3) The unsteady calculation shows that the transient characteristics of the compressor flow are composed of two definite frequencies, blade passing frequency and low-frequency term near stall. The low-frequency term is directly related to shroud-corner separation, and the hub dihedral surely alleviated the unsteadiness of shroud-corner separation.

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