Role of Hub-Corner-Separation on Rotating Stall in an Axial Compressor

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A 3D computation was conducted to investigate the role of hub-corner-separation on the rotating stall in a low-speed axial compressor. It is generally known that tip leakage flow plays an important role in stall inception. However, not much attention has been paid to the role of hub-corner-separation on the rotating stall although it is a common flow feature in an axial compressor operating near the stall point. During our time-accurate unsteady simulation, we suspected that hub-corner-separation might be a trigger for the rotating stall. After an asymmetric disturbance is initiated at hub-corner-separation, this disturbance is transferred to the tip leakage flows and grows to become an attached stall cell, which adheres to the blade passage and rotates at the same speed as the rotor. When the attached stall cell reaches a critical size, it moves along the blade row and becomes the rotating stall. The rotating speed of the stall cell decreases to 79% of the rotor so the stall cell rotates in the opposite direction to the rotor in the rotating frame.

Key Words: Hub-Corner-Separation, Rotating Stall, Stall Inception

Nomenclature

\( c \): chord length
\( c_p \): pressure coefficient
\( c_{PT,\text{Rot}} \): rotary total pressure coefficient
\( D \): diffusion factor
\( E, F, G \): inviscid flux vector
\( E_v, F_v, G_v \): viscous flux vector
\( M_\alpha \): axial Mach number
\( p \): static pressure
\( p_T \): total pressure
\( p_{T,\text{Rot}} \): rotary total pressure
\( Q \): conservative variable vector
\( s \): spacing
\( U \): rotational speed of rotor
\( U_m \): rotational speed of rotor at mid-span
\( V \): absolute velocity on inertial coordinate
\( V_a \): area-averaged absolute axial velocity
\( W \): relative velocity
\( I, 2, 3 \): measurement positions
\( \beta \): flow angle
\( \phi \): flow coefficient
\( \rho \): density
\( \sigma_{M_\alpha} \): standard deviation of axial Mach number
\( \xi, \eta, \zeta \): generalized curvilinear coordinate
\( \psi \): static pressure rise coefficient
\( \tau \): non-dimensional time

1. Introduction

The rotating stall in a compressor is a phenomenon whereby separations in blade passages advance along the blade row in the circumferential direction. It is generally known to be initiated in the operating range between normal flow and surge. These moving separation regions sometimes called stall cells can block the flow in the blade passage, resulting in a reduced operating range. Moreover, the rotating stall adds stress to blades and can break blades due to pressure changes on the surface of blades. Since this deterioration and damage have bad effects on airplane reliability as well as the compressor itself, much attention has been paid to the characteristics of the rotating stall to establish effective active control methods.

Since Emmons et al.1) advanced a theory explaining the propagation of the stall cell in axial compressors, theoretical studies using a lumped parameter approach have been conducted to analyze the characteristics of the rotating stall.2) However, most studies on the rotating stall have been conducted using experiments focused on stall inception.3–9) Generally, axial compressors have better performance near the stall point, but many stability problems arise near stall and surge points. Consequently, to operate compressors safely near the stall point and to control the rotating stall, it is necessary to understand rotating stall precursors. Based on these previous results, the rotating stall follows a pre-stall wave such as modal perturbations (with long-length-scale) or originates from spike-type precursors (with short-length-scale) depending on the compressor operating condition. In addition, it is generally known that the rotating stall in axial compressors is induced in the tip region by interaction between tip leakage flow and other phenomena, such as passage shock and boundary layer on the casing and blade surfaces. Since Epstein et al.10) proposed the active control of the rotating stall, a significant amount of research has sought to control it.11–16) Such research found that both
upstream injection and downstream bleed-off can delay the rotating stall and expand the operating range of axial compressors. In recent years, many numerical studies on the rotating stall have been published.17–24) Hoying et al.20) established a relationship between stall inception and trajectory of the center of the tip leakage flow, hypothesizing that a disturbance is initiated from the vortex motion of the tip leakage flow. Niazi24) conducted both simple numerical simulations and numerical tests of control techniques for the rotating stall. These numerical results showed the stall inception process in detail and gave an intuitive understanding about the rotating stall characteristics.

Based on these experimental and numerical results, it is now recognized that tip leakage flow plays an important role in stall inception. However, little attention has been paid to role of hub-corner-separation on the rotating stall although it is a common flow feature in an axial compressor operating near the stall point, exerting a large effect on internal flows and loss characteristics. Although some researchers25–28) have investigated the structure of corner separation and its impact on internal flows in an axial compressor, only Day25) noted that corner separations occurred throughout his stall tests, and that the size of these separations was changed periodically by the modal wave. However, there is still no definitive explanation of the cause of the rotating stall to date. Therefore, we conducted a 3D computation to investigate whether or not hub-corner-separation causes the rotating stall.

2. Test Configuration

2.1. Geometric specifications

We conducted a numerical study of the role of hub-corner-separation on the rotating stall, using the database from experiments on a low-speed axial compressor tested by Wagner et al.27,28) Because this compressor not only has a rotor without stator and inlet guide vane, but it also rotates slowly at 510 rpm, the maximum pressure ratio between the inlet and outlet is equal to about 1.01. The Reynolds number based on the inlet velocity and blade chord length is about 250,000. This compressor has 28 blades, which have a circular-arc camber line with NACA 65 airfoil as a base profile. Unlike other axial compressors with a constant tip clearance, it has variable tip clearances such as 2.3% of the chord length at the blade leading edge, 1.0% at the mid-chord and 3.3% at the trailing edge. Detailed geometry specifications are summarized in Table 1.

2.2. Measurement positions

By changing the inlet boundary layer thickness on the hub and casing, Wagner et al.27,28) conducted a series of experiments on separations on the blade surface and secondary flows at the downstream of the rotor using the rotor test rig shown in Fig. 1. There are five important measurement stations in the experiment. The boundary layer thickness is adjusted at STA.-1 with various screens of different wire diameter and spacing. The inlet and exit flow conditions such as total pressure, total temperature, density and velocity are measured at STA.1 and STA.2 respectively. To complete the pressure rise curve, the upstream and downstream static pressures are measured on the hub and tip at STA.1 and STA.3. There is a small gap on the hub between the moving and stationary parts at STA.4. The relative measurement positions are summarized with STA.0 as a reference point in Table 2.

3. Numerical Method

3.1. Numerical scheme

Simulations of the 3D flow were conducted using the TFlow flow solver. TFlow has been improved to calculate the internal flow in turbomachinery since its development in the mid-1990s.29) It has been validated through a series of calculations for subsonic and transonic axial compressors and a subsonic axial turbine.30–32) TFlow uses the compressible RANS (Reynolds Averaged Navier-Stokes) equations like Eq. (1), which are written in the normalized form of a generalized curvilinear coordinate system, to describe the viscous flows through a blade row.
The governing equations were discretized by the finite volume method in space. An upwind TVD (Total Variational Diminishing) scheme based on Van Leer’s flux vector splitting method\(^{33}\) was used to discretize the inviscid flux terms and MUSCL (Monotone Upstream Centered Scheme for Conservation Law) technique was used to interpolate flow variables. The second-order central difference method was used to discretize the viscous flux terms. The equation was solved using the Euler implicit time-marching scheme with first-order accuracy to obtain a steady solution and also with second-order accuracy to simulate unsteady flow. The laminar viscosity was calculated by Sutherland’s law and the turbulent viscosity was obtained using the algebraic Baldwin-Lomax model because the flow field was assumed to be fully turbulent. In the unsteady simulation of the rotating stall, the computational domain must have several blade passages in a blade row to show the movement of stall cells. Since an identical grid was used at each passage, each blade passage was assigned to one processor, and the flow variables in contact with other passages were transferred by MPI (Message Passing Interface) libraries.

### 3.2 Computational domain and grid

By using the measurement positions as a guideline, the computational domain was fixed in the region between STA.1 and STA.3, and a multi-block hexahedral mesh was generated using ICEM-CFD.\(^{34}\) To capture the motion of stall cells, four blade passages, one seventh of all blade passages, were used for an unsteady simulation as shown in Fig. 2. Although four blade passages were insufficient to accurately predict the rotating speed and frequency of fully developed rotating stall, they were sufficient to capture where the first asymmetric disturbance triggered the rotating stall. Once a steady solution was obtained using a mesh with about 0.5 million nodes in one passage, the simulation of the rotating stall was conducted using the same mesh with four passages. Each passage consists of 125 nodes in the streamwise direction, 58 nodes in the pitchwise direction, and 73 nodes in the spanwise direction. To capture the tip leakage flow correctly, the region of the tip clearance was filled with the embedded H-type grid, which has 52 nodes from leading edge to trailing edge of the blade, 10 nodes across the blade thickness and 16 nodes from blade top to casing. Therefore, the whole computational domain with four passages has a total of about 2.1 million nodes, where the distance of the first grid point from the wall is set at \(y^+\) to be equal to or less than 5.

### 3.3 Boundary and initial conditions

In simulation of turbomachinery internal flows, there are four types of boundary such as inlet, outlet, wall, and periodic conditions. By using the temperature and pressure at the standard atmosphere and the velocity profile with thick boundary layers on the hub and casing measured by Wagner et al.,\(^{27}\) the total pressure, total temperature and flow angles were fixed at the inlet condition and the upstream-running Riemann invariant was extrapolated from the interior domain. For the outlet condition, the static pressure on the hub was specified and the local static pressures along the span were given by using the SRE (Simplified Radial Equilibrium) equation. Other flow variables such as density and velocities were extrapolated from the interior. On the blade walls and endwalls, the no-slip condition was used to calculate velocity components. The surface pressure and density were obtained using the normal momentum equation and adiabatic wall condition respectively. Since only one seventh of all passages were calculated, it was necessary to implement the periodic condition between the first and fourth passages.

Using the steady simulation result near the stall point \((\phi = 0.65)\) as an initial condition, a time-accurate unsteady simulation was conducted as the back pressure at the outlet condition, \(p_3/p_1\), was set to be 1.008, which is a slightly larger value than the static pressure of \(\phi = 0.65\). However, no artificial asymmetric disturbances were imposed at the inlet condition.

### 4. Numerical Results

#### 4.1 Performance curve

Generally, compressor performance can be characterized by the flow coefficient and static pressure rise coefficient, which are defined as follows. In the experiments, the latter value was calculated using the static pressure increment between STA.1 and STA.3 at both tip and hub.

\[
\phi = V_\tau / U_m, \quad \psi = \frac{p_3 - p_1}{0.5 \rho U_m^2}
\]

As shown in Fig. 3, the static pressure rise curve obtained by the steady numerical computation corresponds to the experimental one within the range between \(\phi = 0.65\) and \(\phi = 0.95\). There is a peak performance point at \(\phi = 0.75\) in both the numerical and experimental results. The static pressure rise coefficients for the unsteady numerical simulation were calculated using the instantaneous flow data, which were saved four times a period. One period here means the time it takes the four blades to traverse the
The computational domain once. The numerical result of the unsteady simulation matches the experimental result until the rotating stall is fully developed. However, there are some discrepancies after development of the rotating stall because only four blade passages were used in the unsteady simulation. The numerical result predicts the stall point relatively well in the static pressure rise curve, and the abrupt drop in performance can be found in the process of stall development between $\phi = 0.58$ and $\phi = 0.55$ although the performance drop is smaller than the experimental one. The static pressure rise coefficients are clustered around the experimental value in the early stage of the unsteady calculation because the asymmetric disturbance is small. The rotating stall initiated at $\phi = 0.62$ scatters the static pressure rise coefficient because of the disturbance in the flow downstream of the rotor caused by small stall cells in the four passages. While there is no large performance drop at stall inception, the performance drops abruptly below $\phi = 0.58$ as the rotating stall develops from the part-span stall to the full-span stall.

### 4.2. Steady flow

To validate the accuracy of the 3D flow solver used in this study, both the pressure coefficients on the blade surface and the flow angles downstream of the rotor obtained from the steady simulation were compared with the experimental results. Unfortunately, validation of unsteady results was impossible because there is no experimental data for validating the operating range below $\phi = 0.65$.

Figure 4 shows the distributions of pressure coefficients on blade surfaces at 4%, 50% and 95% spans from the hub. The pressure coefficient is defined as follows:

$$c_p = \frac{p - p_1}{0.5 \rho V_1^2}$$

Comparison of the computational and experimental results shows good overall agreement from the hub to the tip, but there are small discrepancies on the suction surface. This might be due to the transition region on the suction surface reported in the experiment, which could not be captured by the fully turbulent flow solver. As the flow rate decreases, the maximum load increases and the points of the maximum load move from one-fourth of the chord to the leading edge.

Figure 5 shows the distribution of the exit flow angles along the span, which were area-averaged in the pitchwise direction. Although there is a local maximum point at about 20% span from the hub at a flow coefficient of $\phi = 0.65$, the flow angle distribution shows good agreement with the experimental data. This local maximum point is almost equal to the center of the hub-corner-separation between the suction and hub surfaces, and it is caused by underturning of the flow due to blockage by the separation. When the rotor is operating at $\phi = 0.85$, the flow angle of the computation is also very similar to the experimental one.

It is very important to investigate the flow characteristics in the steady state near the stall point because internal flows at this point have a large effect on the rotating stall. The coefficient of the rotary total pressure was calculated at the downstream of the rotor to investigate the changes in the internal flows such as hub-corner-separation and tip leakage flow in proportion to the load. The rotary total pres-
sure is frequently used to remove the effects of rotation and is defined as:

$$ p_{T, \text{Rot}} = p + \frac{1}{2} \rho (W^2 - U^2) $$

Its coefficient was obtained using both the area-averaged total pressure at the inlet and the rotary total pressure, and it is defined as follows:

$$ c_{pT, \text{Rot}} = \frac{p_{T,1} - p_{T, \text{Rot}}}{0.5 \rho U_m^2} $$

The computational result was compared with the experimental result of Wagner et al. as shown in Fig. 6, revealing good agreement except for the size of the region affected by the tip leakage flow. The small hub-corner-separation found at the design condition ($\phi = 0.85$) becomes large at the junction of the hub and suction surfaces as the load is increased.

To reveal the cause of hub-corner-separation, the diffusion factor was calculated using the inlet and exit flow angles along the span, which were area-averaged in the pitchwise direction. This factor can be obtained by the following equation and the results are shown in Fig. 7.

$$ D = \left( 1 - \frac{\cos \beta_1}{\cos \beta_2} \right) + \frac{s \cos \beta_1}{c} \left( \tan \beta_1 - \tan \beta_2 \right) $$

Separation occurs on the suction surface when this factor is greater than 0.6 (Horlock). The diffusion factor at $\phi = 0.85$ exceeds 0.6 from the hub to 10% span and from 87% span to 94% span, but it is smaller than 0.6 in the core flow region. The separation region, where the diffusion factor is greater than 0.6, grows large as the flow coefficient is reduced. This means that the separation can be easily initiated near the hub by large deflection of the blade. This separation acts as blockage and results in a large loss of total pressure in the steady flow. Moreover, we found that this hub-corner-separation has a large effect on stall inception.

### 4.3 Unsteady flow

Four numerical sensors were installed at 85% span from the hub and 25% of the chord length upstream of the leading edge to evaluate whether the rotating stall originated or not, as shown in Fig. 8. These sensors rotate counter clockwise as the calculation proceeds because the numerical simulation was conducted in the rotating frame, and the sensors read the axial velocity at each position 480 times a period.
Figure 9 shows the time-history of axial velocities measured by each numerical sensor. There is no disturbance at the beginning of the unsteady calculation, but some small disturbances appear in the time-history near 9.0 periods although no artificial asymmetric disturbance was imposed. This disturbance moves at the same speed as the rotor, meaning that it adheres to the blade row. As the flow coefficient is reduced, the disturbance in the axial velocity grows bigger by throttling. The rotating stall is abruptly found in the time-history of the numerical sensors at about 19.5 periods, and the flow coefficient has a value of 0.62 at this moment. Wagner et al.\textsuperscript{27) have stated that the effect of the rotating stall is observed for flow coefficients below 0.57 because the backflow is seen with tufts mounted on the hub and casing between STA.0 and STA.1. However, it is possible that the rotating stall observed in the experiment was the fully developed rotating stall, and it might have been initiated at the higher flow coefficient because the blockage caused by the small stall cell, which was just initiated, was not strong enough to turn the flow direction around the tufts at one chord upstream of the rotor. The rotational speed of the stall cell quickly falls to 79% of the rotor speed and moves in the opposite direction to the rotor blade in the rotating frame. The speed decreases continuously from 79% to 74% of the rotor speed as the size of the stall cell increases after stall inception.

To capture the cause of the pre-stall disturbance, the distributions of the rotary total pressure on the cylindrical surface near the tip are shown in Fig. 10, where the coefficient of the rotary total pressure was calculated using Eq. (5). Numerical sensors cannot detect any signal of disturbance at 3.0 periods because the rotary total pressure in the tip region has similar features in all passages. The front line of the tip leakage flow, A, is located behind the leading edge plane at this moment. Then local disturbances can be observed in the tip leakage flow near 9.0 periods, when the numerical sensors detect some disturbance for the first time, and the rotary total pressure shows some different pattern at each passage as shown in Fig. 10(b). The front line of the tip leakage flow, B, moves upstream in comparison to Fig. 10(a) but is still located slightly behind the leading edge plane. This disturbance is fixed inside the blade passage, rotates with the rotor at the same speed and grows to be a bigger attached stall cell by throttling as shown in Fig. 10(c). When the attached stall cell reaches a critical size, the tip leakage flow locally moves around the leading edge of the next blade and spills into the adjacent flow passage due to blockage of the attached stall cell. The attached stall cell finally changes to a short-length-scale rotating stall through this stall inception process as shown in Fig. 10(d).

Once the rotating stall is generated, it advances to the next blade one by one and grows into a large stall cell in a relatively short time scale as shown in Fig. 10(e, f).
The localized disturbance is caused by interaction of the tip leakage flow and the main flow in a subsonic axial compressor, as Hoying et al.\textsuperscript{20} reported, when the front line of the tip leakage flow propagates forward of the leading edge plane. Hah et al.\textsuperscript{23} showed that interaction between the tip leakage flow and the passage shock causes a local disturbance in a transonic axial compressor. The first localized disturbance in this study was generated before the front line of the tip leakage flow reached the leading edge plane, and the single rotor used in this study cannot have any passage shock because it is a subsonic compressor. Therefore, there must be another mechanism triggering the local disturbance of the tip leakage flow in this case. Figure 11 shows the rotary total pressure distribution at STA.2 at 3.0 periods. There was no disturbance in both the time-history of the axial velocities and the rotary total pressure near the casing as mentioned above, but the hub-corner-separation showed some asymmetric behavior at STA.2 at this moment. This disturbance might have been generated by numerical round-off error or by small changes in flow caused by the pressure increment at the outlet.

To investigate the change in hub-corner-separation at stall inception, the passage-averaged axial Mach number and its standard deviation were calculated using the flow field of four passages. The results are shown in Fig. 12. The passage-averaged value shows that the hub-corner-separation has a large separation region at 3.0 periods, but it is reduced at 15.0 periods because blockage of the tip leakage flow grows continuously as the flow coefficient is reduced. The core flow region, which is excluded from the region affected by the tip leakage flow and hub-corner-separation, has a small standard deviation in the axial Mach number, suggesting that all passages have similar flow phenomena in the core flow region. A large standard deviation occurs near the hub-corner-separation region at 3.0 periods but not in the tip region. The standard deviation in the tip region increases at 9.0 periods because the local disturbance is initially generated in this region, while the value near the hub-corner-separation is reduced. The attached stall cell at 15.0 periods increases the standard deviation near the casing, and the value of the hub-corner-separation is reduced slightly compared to that at 9.0 periods.

This numerical simulation revealed two interesting points. First, the initial disturbance is originated by hub-corner-separation on the suction surface. Then, the disturbance emerges in the tip region when the front line of the tip leakage flow is still located behind the leading edge plane. These results led us to conclude that asymmetric disturbance is transferred to the tip leakage flow from the hub-corner-separation. Second, the attached stall cell is observed in the process of rotating stall development. This localized attached stall cell is different from a modal-wave because it has the same rotational speed as the rotor. This stall development via the attached stall cell may cause the spike-type rotating stall because it is an abrupt process to change the attached stall cell to the rotating stall cell as shown in Fig. 9.

5. Conclusion

A 3D numerical simulation was conducted to study the role of hub-corner-separation on stall inception in a subsonic axial compressor. In the steady simulation, the hub-corner-separation caused by large deflection of the blade increases on the suction surface as the flow coefficient decreases. In the unsteady simulation, the asymmetric disturbance initiated by the hub-corner-separation is transferred to the tip leakage flow although no artificial disturbance is imposed at the inlet condition. This disturbance grows to become an attached stall cell, which adheres to the blade passage in the throttling process. When the attached stall cell near the casing reaches a critical size, this cell moves along the blade row and develops into the rotating stall. This numer-
tical study found that hub-corner-separation on the suction surface is another cause of the rotating stall. In addition, stall development via attached stall cells might cause the spike-type rotating stall because the change from attached stall to rotating stall occurs abruptly in a short time scale.

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