ABSTRACT

Rotating stall in the high speed compressor can lead to violent disruption of flow, damage to the blade structures and, eventually, engine shutdown. To investigate the mechanism of stall inception and stall propagation in multi-stage compressor, the onset and transient behavior of rotating stall in a transonic 2-stage compressor are simulated with full-annulus unsteady method, based on the in-house software ASPAC. The choked throttle model is adopted. The results indicate that, in the 2-stage compressor, stall cells exist in rotors of both compressor stages, which propagate at 43% and 25% of the shaft rotational speed respectively. Meanwhile, the rotating stall disturbance can be suppressed by stator of the first stage, which will weaken the influence of rotating stall on the compressor stage downstream.

INTRODUCTION

Aerodynamic stability of compressor is one of the key factors influencing performance, stability, and reliability of aero-engine. The control strategy for suppressing aerodynamic instability in compressor can be optimized through exploring the mechanism of aerodynamic stability. As a result, the stability boundary of compressor will be expanded, the performance of compressor will be improved.

Rotating stall is the main forms of aerodynamic instability in compressors. However, the flow mechanism of rotating stall is not fully understood[1]. However, experimental test and comprehensive measurement during stall and surge cycles, especially for multi-stage high speed machines, remain a challenging task[2]. Simulating the whole process of rotating stall development based on CFD is an effective way to uncovering the flow mechanism.

3D unsteady Reynolds-averaged Navier-Stokes (URANS) methods have been successfully applied to simulation of rotating stall phenomenon in compressors[3]–[12], and most of the studies concentrated on rotors or single-stage compressors. It is pointed by Wu[12] that, flow characteristics of rotating stall in single-row environment are different from that in multi-row environment. However, very limited work can be found on multi-stage compressors[2].

This paper presents an investigation into the onset and transient behaviour of rotating stall in a 2-stage high speed axial compressor with full-annulus unsteady method, based on the in-house software ASPAC.

METHODOLOGY

The simulation is conducted based on the in-house code ASPAC, which had been validated for various turbomachinery flows. The 3-D compressible reynold averaged Navier-Stokes equations are solved by fully-implicit scheme with a cell-centered finite volume method. The inviscid flux is evaluated by Roe Scheme with the Van Albada limiter. The viscous flux is determined in a central differencing manner with Gauss’s theorem. The second-order
backward difference is applied to the temporal derivative and the inner iteration is conducted at each time step. The parallel is conducted based on MPI.

Taking Rotor 37[13] as an example, comparison between simulational results of ASPAC and experiment results is shown in Fig. 1. ASPAC_RANS means the steady reynold averaged Navier-Stokes equations are solved, ASPAC_URANS means the unsteady reynold averaged Navier-Stokes equations are solved. The results of ASPAC match well with experimental results.

![Fig. 1 Rotor 37 speedline](image)

Taking Stage 35[14] as an example, comparison between simulational results of ASPAC and experiment results is shown in Fig. 2. In terms of ASPAC_RANS, the mixing plane method is applied. In terms of ASPAC_URANS, the dynamic overlapped grid technique is applied at rotor/stator interface. The results of ASPAC match well with experimental results.

![Fig. 2 Stage 35 speedline](image)

RESULTS AND DISCUSSION

Design parameters of the 2-stage compressor are listed in Table 1, where rotori denoted rotor of i\textsuperscript{th} stage and statori denoted stator of i\textsuperscript{th} stage.

<table>
<thead>
<tr>
<th>Design parameters of 2-stage compressor</th>
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<tbody>
<tr>
<td>Rotor Speed at 100% Speed</td>
</tr>
<tr>
<td>Blade number of rotor1</td>
</tr>
<tr>
<td>Blade number of stator1</td>
</tr>
<tr>
<td>Blade number of rotor2</td>
</tr>
<tr>
<td>Blade number of stator2</td>
</tr>
</tbody>
</table>

Numerical model of the 2-stage compressor is shown in Fig. 3.
Full-annulus grid of the compressor is shown in Fig. 4. The space distance of the first layer of grid is 3e-6m and $y^+$ is 1. Cell number of the full-annulus grid is 83 million.

When simulating speedline of the compressor, the static pressure at hub of outlet is set and the radial equilibrium equation is applied. In near-stall condition, the "choked" throttle model that specifies corrected mass flow at the exit. This boundary condition allows variation of exit static pressure to match the compressor exit mass corrected to the exit total condition. The formula is as follows:

$$P_c = P_a + \frac{m_r^2}{k_t}$$  \hspace{1cm} (1)

Where $P_c$ is static pressure at outlet, $P_a$ is the environment pressure which is set as 101325Pa in current study, $m_r$ is mass flow at outlet, $K_t$ is a parameter to control the throttle area.

The full-annulus simulation is conducted using 256 ARM V64 CPU cores. A temporal discretization level of 2000 physical time steps per rotor revolution and 50 inner iteration steps are used. The CFL number is set to 5.

The speedline of the compressor is simulated via mixing plane method(ASPAC_RANS). the result is shown as the red line in Fig .5. When it comes to near stall condition, the steady flow is used as the initial flow field for unsteady computation(ASPAC_URANS) and the choked throttle model is used. When applying the choked throttle model, $K_t$ is calculated at the first time, which is then multiplied by 0.95 and stay unchanged during the whole process of unsteady simulation. The onset and development of rotating stall in the compressor is shown as the green line in Fig .5. After fully developed, the rotating stall is marked in balck circle as shown in Fig. 5.
Variation of mass flow during stall development is shown in Fig. 5. At about 28.5th revolution, the throttle is applied, the total pressure at outlet increases sharply, the mass flow decreases sharply. At about 36th revolution, the mass flow drops to the lowest point. The mass flow then increases gradually, after 6 revolutions, the mass flow reaches to the rotating stall condition as marked in Fig. 4.

The numerical probes are placed at shroud near the middle chord position of blade tip. 23 probes are placed at 23 blade passages of rotor1 with the opposite direction as rotational direction. 29 probes are placed at 29 blade passages of rotor2 with the opposite direction as rotational direction. The instantaneous static pressure on numerical probes are shown in Fig. 6. To show the regulation more clearly, the pressure of numerical probe at i\textsuperscript{th} passage is transformed as follows:

\[ P_i = 2 \cdot i + \frac{P_i - 78000}{40000} \]  \hspace{1cm} (2)

The red line on the left denotes the propagation of stall cells, which rotates at 43% of shaft speed at rotor1 and at 25% of shaft speed at rotor2. Which means that rotating stall performs differently at different stages in multi-stage environment. The characteristics of rotating stall are decided by geometry characteristics and aerodynamic characteristics of different stages. The stall cell in rotor2 occurs about 0.3 revolution earlier than that in rotor1, which means that the stall disturbance originates from rotor2, and rotor1 is then influenced.
Fig. 6 Variation of static pressure on the shroud in different blade passage

The instantaneous entropy distributions of s3 surface at different blade rows are shown in Fig. 7~Fig. 10. The flow fields before stall onset(7.5th revolution) and after rotating stall is fully developed(41.2th revolution) are displayed. As the entropy increase greatly, the value intervals before and after stall onset are set differently.

In terms of rotor 1, before emergence of rotating stall, high entropy region only exist abound blade and blade tip. When rotating stall is fully developed, top half span at full-annulus are influenced by stall cells, as shown in Fig. 7. When it comes to stator1, the stall cells are cut by stator blades, flow in some blade passages returns to normal, as shown in Fig. 8(b). However, the stall cells spread from top half span to region near hub. As a result, the stall cells show different characteristics in rotor2 compared with rotor1, as shown in Fig. 9(b). In rotor2, part of the annulus are occupied by stall cells and the full span of blade passage is influenced.

Fig. 7 Instantaneous entropy distribution on s3 surface of rotor1
Fig. 8 Instantaneous entropy distribution on s3 surface of stator1

(a) 7.5\textsuperscript{th} revolution  
(b) 41.2\textsuperscript{th} revolution

Fig. 9 Instantaneous entropy distribution on s3 surface of rotor2

(a) 7.5\textsuperscript{th} revolution  
(b) 41.2\textsuperscript{th} revolution
CONCLUSIONS

The in-house CFD software ASPAC is validated on Rotor 37 and Stage 35. Which is then applied to simulation of onset and development of rotating stall in 2-stage compressor. Some conclusions are drawed as follows:

1. The results of steady and unsteady simulation based on ASPAC match well with experiments.
2. The stall cells display different characteristics in rotors of the two compressor stages. The rotational speed of stall cells in rotor1 and rotor2 is different. In rotor1, top half span at full-annulus is influenced by stall cells. The stall cells are splitted by blades of stator1. As a result, in rotor2, part of the annulus is occupied by stall cells and the full span of blade passage is influenced.

REFERENCES


