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Analysis of blade vibration response induced by rotating stall in axial compressor

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Abstract: An experimental and numerical study was conducted to investigate the forced response of blade vibration induced by rotating stall in a low speed axial compressor. Measurements have been made of the transient stalling process in a low speed axial compressor stage. The CFD study was performed using solution of 3-dimensional Navier-Stokes equations, coupled with structure finite element models for the blades to identify modal shapes and structural deformations simultaneously. Interactions between fluid and structure were managed in a coupled manner, based on the interface information exchange until convergence in each time step. Based on the rotating stall measurement data obtained from a low speed axial compressor, the blade aeroelastic response induced by the rotating stall flow field was analyzed to study the vibration characteristics and the correlation between the phenomena. With this approach, good agreement between the numerical results and the experimental data was observed. The flow phenomena were well captured, and the results indicate that the rotating field stall plays a significant role in the blade vibration and stress affected by the flow excitation.

Key words: compressor; rotating stall; blade forced response; CFD; fluid-structure interaction

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Modern turbofan engines are highly developed complex systems that provide a large amount of power with relatively low weight. With the demand for higher aerodynamic performance, the blades are more likely to react to the unsteady aerodynamic loading. Fatigue failures of compressor rotor blades have been a major problem in the development of the axial flow compressor. One of the sources of many failures has been rotor blade vibration excited by rotating stall.

The ability of a compression system to deliver the required pressure ratio over a wide range of settings is essentially governed by the performance map of the system. As the mass

flow rate through the compression system decreases, the pressure rise across the system increases. However, the system stalls as below a certain mass flow. The rotating stall, a typical dynamic instability for performance, has been long concerned in turbomachinery. The rotating stall process has attracted a number of attentions over the past several decades. Stall in compressors can be associated with initiation of several types of fluid dynamic instabilities. These instabilities and the different phenomena, surge and rotating stall, which result from them were discussed in the paper of Greitzer^[1].

Numerous studies have shown that the fluid dynamics mechanism of stall inception is identi-

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cal regardless how the compressor stalls. It is closely related to the tip clearance flow. Hoying et al^[2] examined the influence of three-dimensional flow structures within a compressor blade passage to determine the mechanism in rotating stall inception. They found that the stall inception depends on the stability of tip clearance leakage vortex. Vo et al^[3] studied the impact of blade passage flow on the compressor stability. He summarized two criteria for the subsonic compressor stall inception: tip clearance flow spillage below blade tip at the leading edge and backflow at the trailing edge blade tip. Stall was interpreted to occur as a result of a critical momentum balance between the approach fluid and the tip-leakage flow which is investigated by Matthew^[4]. Therefore, the blockage of tip passage caused by the leakage vortex movement and the backflow at the trailing edge are the principle causes of stall inception^[5-7].

Rotating stall is a local phenomenon in which a circumferentially uniform flow pattern is disturbed, and some of the blades stall. The stalled regions move from blade to blade; and appear to rotate about the shaft axis, at an angular velocity which is 1/3 to 1/2 that of the angular velocity. Rotating stall not only brings aerodynamic drawbacks but also may lead to vibrations and fatigue of the compressor blades^[8-10]. Forced vibration can occur due to the excitation produced by passing through rotating stall cells. Forced vibration is critical when the excitation frequency coincides with a natural frequency of the blade. When pressure fluctuation occurs, it acts as a periodic force on the blades. This produces blade vibrations of significant magnitude. A rotating stall and blade vibration survey was conducted on a modified production turbojet engine by Howard^[11]. It was found that the maximum vibratory stresses occurred at the first and second stages. These vibrations were excited by the second harmonic of the three-zone relative stall frequency, which was not in resonance with the natural bending frequency of the blade. Engeda^[12] examined the basic knowledge in both

rotating stall and blade vibration, took an effort to correlate them. Despite the progress in design, there is still no known direct analytical correlation between rotating stall and blade vibration. As the need for higher pressure ratio and wide operational range, the solution to correlate rotating stall and blade vibration is highly needed.

With the great improvement of CFD method, it becomes possible that fluid-structure coupled simulations of aeroelastic phenomena in the time domain can be carried out in practice. Few of the current studies in this field touched on the blade vibration induced by rotating stall in CFD coupled method. In this present work, an experimental measurement is conducted to obtain the transient stalling data, and a time domain numerical approach is carried out to enhance the understanding of three dimensional blade row aeroelastic characteristics under the parallel computation. With a good agreement between the numerical results and the experimental data, the blade aeroelastic response induced by the rotating stall flow field is analyzed to study the vibration characteristics and the correlation between the phenomena.

1 Numerical method

1.1 Fluid dynamics

A finite-volume based method is applied to solve the conservation laws for unsteady compressible flow. The Navier-Stokes equations in integral form can be written as

$$\iiint_V \frac{\partial \mathbf{U}}{\partial t} dV + \iint_S \mathbf{F} \cdot d\mathbf{S} = \iiint_V \mathbf{F}_{st} dV \quad (1)$$

where V is the control volume with boundary surface S . The vector of state variables $\mathbf{U}$ is given by

$$\mathbf{U} = (\rho, \rho \mathbf{u}, \rho \mathbf{v}, \rho \mathbf{w}, \rho E, \rho k, \rho \epsilon)^T \quad (2)$$

where ρ is the density, $\mathbf{u}$, $\mathbf{v}$, $\mathbf{w}$ are the velocity vectors, and E is the total energy of the flow. The eddy viscosity is obtained from the $k-\epsilon$ model. The flux tensor $\mathbf{F}$ consists of the inviscid and viscous parts. And $\mathbf{F}_{st}$ stands for the effects of gravity and other body forces.

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{U}) = 0 \quad (3)$$

This equation states the conservation of mass for an unsteady flow with boundaries. And the momentum equation, total energy equation can be written as

$$\begin{aligned} & \frac{\partial(\rho \mathbf{U})}{\partial t} + \nabla \cdot (\rho \mathbf{U} \times \mathbf{U}) = \\ & \nabla \cdot [-p\delta + \mu(\nabla \mathbf{U} + (\nabla \mathbf{U})^T)] + \mathbf{S}_m \quad (4) \\ & \frac{\partial(\rho H)}{\partial t} - \frac{\partial p}{\partial t} + \nabla \cdot (\rho \mathbf{U} H) = \\ & \nabla(\lambda \nabla T) + \nabla \left(\mu \nabla \mathbf{U} + \nabla \mathbf{U}^T - \frac{2}{3} \nabla \cdot \mathbf{U} \delta \mathbf{U} \right) + \mathbf{S}_e \end{aligned} \quad (5)$$

Where total enthalpy H , the dynamic viscosity μ , the thermal conductivity λ , momentum source $\mathbf{S}_m$, energy source $\mathbf{S}_e$.

In this study, the k - ϵ turbulence model is used for the wall treatment. The values of k and ϵ come directly from the differential transport equations for the turbulence kinetic energy and turbulence dissipation rate

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho u_j k) = \frac{\partial}{\partial x_j} \left(\frac{u_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + G - \rho \epsilon \quad (6)$$

$$\begin{aligned} & \frac{\partial}{\partial t}(\rho \epsilon) + \frac{\partial}{\partial x_j}(\rho u_j \epsilon) = \\ & \frac{\partial}{\partial x_j} \left(\frac{u_t}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial x_j} \right) + C_{\epsilon 1} G \frac{\epsilon}{k} - \rho C_{\epsilon 2} \frac{\epsilon^2}{k} \end{aligned} \quad (7)$$

Where the constant values $C_\mu, \sigma_k, \sigma_\epsilon, C_{\epsilon 1}, C_{\epsilon 2}$ the turbulence viscosity $\mu_t = \rho C_\mu \frac{k^2}{\epsilon}$, the viscous force

$$G = \mu_t \left(\frac{\partial \mu_i}{\partial x_j} + \frac{\partial \mu_j}{\partial x_i} \right) \frac{\partial \mu_i}{\partial x_j}.$$

1.2 Structural dynamics

The structural equations for a mechanical system with a finite number of degrees of freedom are given by

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F} \quad (8)$$

where the mass matrix $\mathbf{M}$, the damping matrix $\mathbf{C}$, the stiffness matrix $\mathbf{K}$, the vector of displacements $\mathbf{x}$, and the forcing vector $\mathbf{F}$.

1.3 Fluid-structure interface

Although the fluid and the structure meshes do not mix, there exists a fluid-structure interface as a two-dimensional manifold in three-dimensional

space. Within each time step, the flow equations and the structural equations are solved simultaneously, exchanging information on the fluid-structure interface. This procedure is repeated until the flow and displacements are converged, before proceeding to the next time step. A distributed pressure load, therefore, must be first transferred into equivalent nodal forces. Such a transformation must satisfy two requirements as conservation of loads and energy.

$$\begin{aligned} \sum F_i &= \int p_j ds \\ \sum F_i u_i &= \iint p_j ds du \\ v_{fi} &= v_{si} \end{aligned} \quad (9)$$

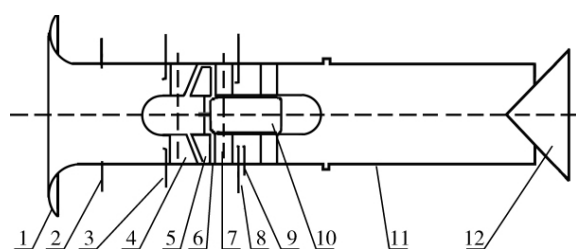
The first one is that the nodal forces must yield the same net forces as the original distributed pressure loads. For the evaluation of the energy, the definition of virtual work is conveniently used to ensure conservativeness. Equating the virtual work performed by the nodal forces F_i acting on a virtual nodal displacement u_i with that done by the original distributed surface pressure p_j moving through the equivalent distributed virtual displacement du , this requirement is given as the second one.

In the aeroelastic applications, the computational domain may be enclosed by moving boundaries. When the boundaries are moving, there must be at least some part of the computational mesh which moves along the boundary motion. This requires the computation of some geometric quantities that include the grid velocities (the fluid interface mesh node velocity v_{fi} , the structure interface mesh node velocity v_{si}), and the geometric conservation equation can be written as the third one.

According to the load transfer and deformation tracking system of Doi et al^[13], ZHANG et al^[14], this mismatch of mesh points on the interface transfers the pressure distribution on the fluid surface mesh to a corresponding nodal load distribution on the structure and also the transfer of the displacements of the structural nodes to corresponding perturbations of fluid mesh using a numerical approximation.

2 Experimental tests of rotating stall in a compressor

The research object of this paper is the low-speed single stage compressor of Beijing University of Aeronautics and Astronautics (BUAA), as shown in Fig. 1. It consists of 19 rotor blades and 13 stator blades. The design mass flow rate is $2.40 \text{ m}^3/\text{s}$ at a rotational speed of 3000 r/min , providing a total pressure rise of 1500 Pa . The rotor tip radius is 225 mm and the hub/casing ratio is 0.75 , with a rotor tip clearance of 0.25 mm . The Reynolds number is about 2×10^5 based on rotor tip chord. Details of the compressor geometry, operating conditions, and experimental data can be found in reference[15]. Some information concerning the compressor stage is given in Table 1.



- 1 Flow gathering ware;
 2 Flow rate static pressure holes;
 3 Inlet total pressure comb; 4 Inlet guide vane;
 5 Rotor blade; 6 Kulite dynamic pressure probe;
 7 Stator blade; 8 Exit total pressure comb;
 9 Exit total pressure rake; 10 Motor;
 11 Rear casing; 12 Throttle plug.

Fig. 1 Compressor test rig

Table 1 Design parameters of axial compressor

Parameters	Value
Rotor revolutions per minute at 100% speed	3000
Tip speed/(m/s)	70.7
Hub-Tip radius ratio	0.75
Number of rotor blades	19
Number of stator blades	13

Detailed measurements have been made of the transient stalling process in an axial compressor stage. Static pressure fluctuations were measured with five Kulite dynamic sensors proposed at the shroud of rotor blade leading edge with differences of 21° in the rotating direction. The stall signals were obtained by varying the position of the throttle and thereby the mass flow. One of the static pressure measurement samples is shown as the following Fig. 2. It indicates a significant fluctuation as the stall occurs.

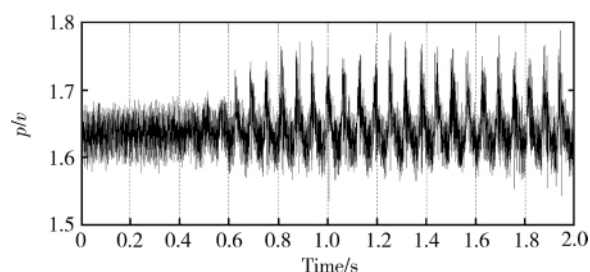


Fig. 2 Pressure measurement samples for one Kulite trace at 100% speed

The main focus of this analysis method was to locate the rotating wave frequency. In order to do this, spectra analysis on stall signal history are carried out to analyze the frequency in detail. For the 19 blades of the compressor rotor, the blade passing frequency can be conducted as:

$$f_{\text{bpf}} = \frac{n}{60} \times z_r = \frac{3000}{60} \times 19 = 950 \text{ Hz} \quad (10)$$

where the blade passing frequency f_{bpf} , the rotor speed at $n \text{ r/min}$, the blade number z_r .

From the Fig. 3, it can be found clearly the

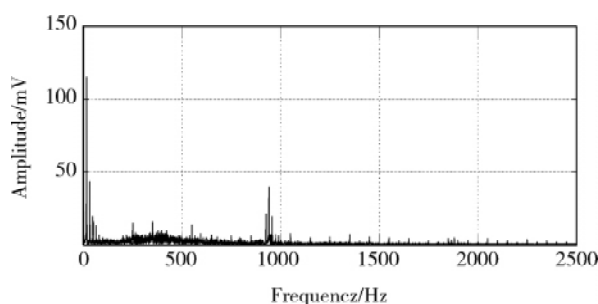


Fig. 3 Amplitude spectrum in a wide range

blade passing frequency (950 Hz). And also from the results obtained by the amplitude spectra of the stall signals, a low frequency appears with high amplitude in the Fig. 4. These experimental results indicate that one stalled region appears to rotate about the eigenfrequency of 17 Hz, at an angular velocity which is 1/3 to that of the angular velocity in the rotating direction (Fig. 5).

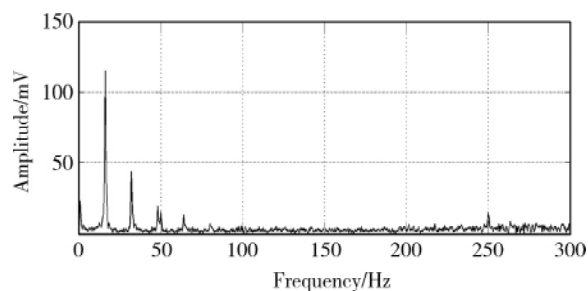


Fig. 4 Amplitude spectrum in a low range

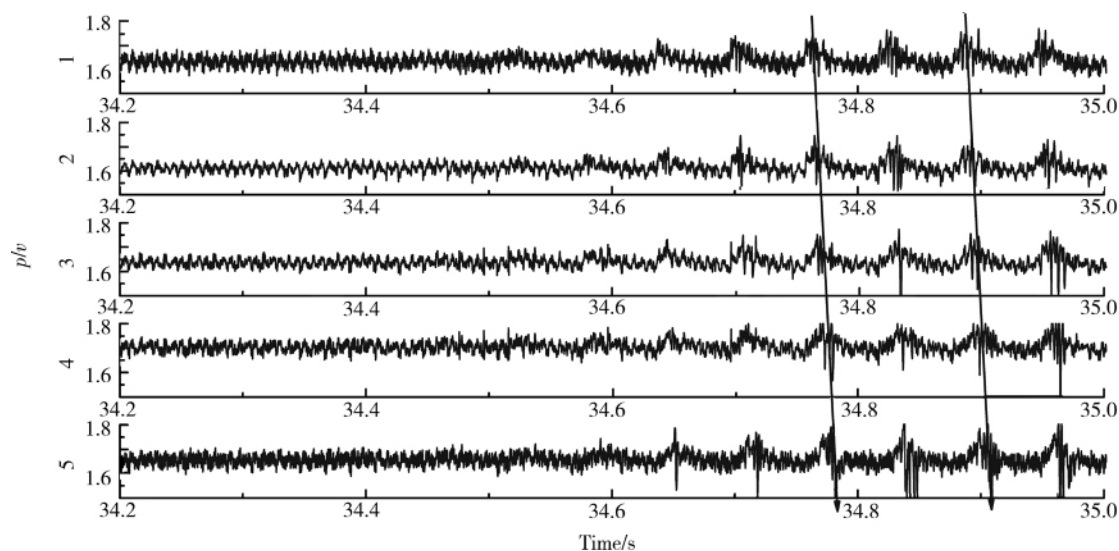


Fig. 5 Pressure measurement samples for five Kulite traces at 100% speed

3 Computation results and discussions

3.1 Computation models

The fluid-structure interaction method described above has been used successfully on a compressor. The unsteady flow field of rotor is studied using numerical solution of three-dimensional Navier-Stokes equations, coupled with structure finite element models for the blades to identify modal shapes and the structural deformations simultaneously.

Interactions between fluid and structure are dealt with in a coupled manner, based on the interface information exchange until convergence in each time step. The experimental measurements section at the inlet of compressor is selected for the calculation field boundary in the simulation. The extension of the inlet and outlet is about 2 times blade chord length in the middle. Fig. 6 shows the CFD model used in Navier-Stokes computations. And the blade first natural

flexural mode shape is shown in the Fig. 7.

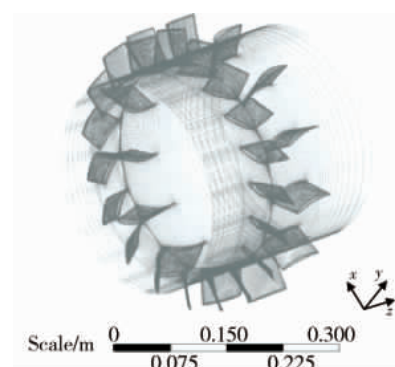


Fig. 6 Computation models

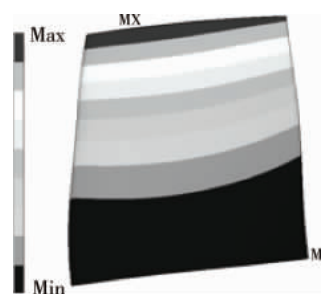


Fig. 7 Blade first natural flexural mode shape

3.2 Computation results

In the following part of the paper, based on the rotating stall measurement data obtained from a low speed axial compressor, the blade aeroelastic responses induced by the rotating stall flow field are performed to analyze the vibration characteristics and the correlation between the phenomena. The part performance of the compressor calculated by the coupled method is shown in Fig. 8 compared with experimental data (EXP). And the characteristic lines calculated by the coupled method agree well with the experiment result at all operating conditions with normalized mass flow Φ .

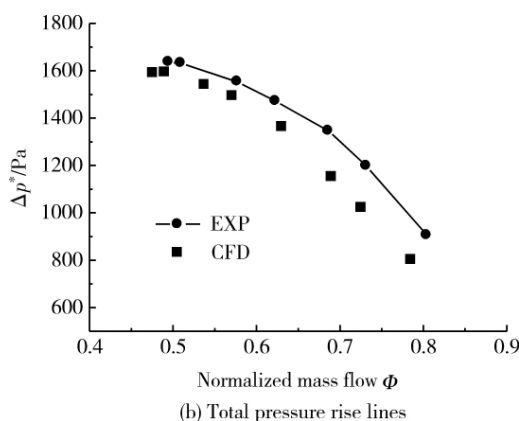
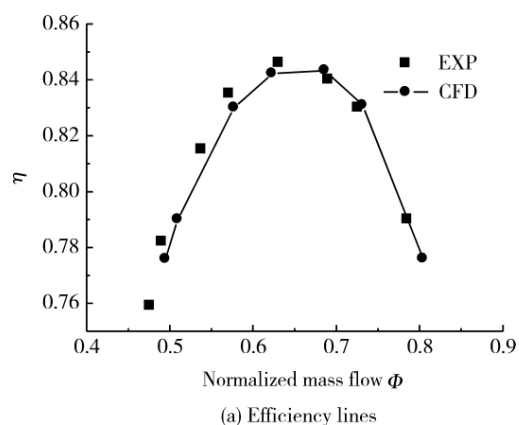


Fig. 8 Comparison of compressor characteristics between CFD and experimental results

As shown the spanwise distribution of total pressure near the design point in Fig. 9, the data obtained by the numerical simulation are in accordance with the experimental result. The above analysis shows that unsteady coupled simulation results are very close to the experiment

measured results. In other words, the unsteady numerical calculation can reproduce the experimental results quantitatively. Therefore, it is reasonable and reliable to predict the flow in the compressor using full annular unsteady simulation during the stall process.

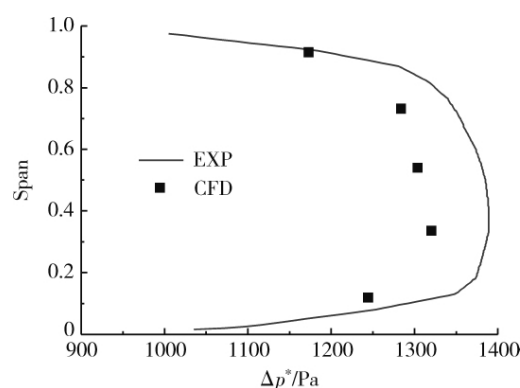


Fig. 9 Spanwise distribution of total pressure near design point

In order to simulate the stall inception-recovery process, two steps are taken. Static pressure is set as the outlet boundary condition to simulate different operating points of the compressor. Firstly unsteady simulation was carried out using constant static pressure as the outlet boundary condition. When the unsteady simulation was converged, the variable static pressure changing with time was applied as outlet boundary condition, as shown in Fig. 10. The static pressure firstly increases and then decreases as time released, the rotating compressor experiences the whole process from entering stall to recovering during the time.

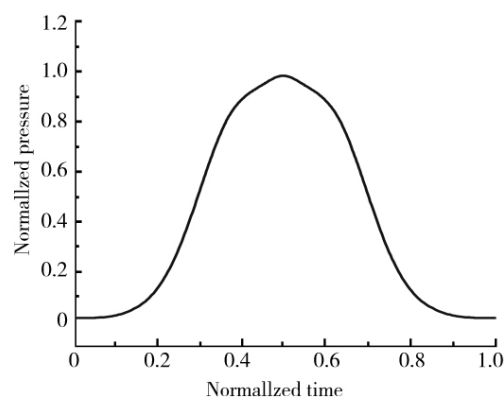


Fig. 10 Variable static pressure at outlet

In order to analyze the stall inception in detail, 17 static pressure measuring points were placed on the shroud of the rotor blade leading edge in the CFD process with differences of 21 degree in the circumference as the same positions in the experiment. Because the static pressure measuring points are fixed on the rotating frame, these points are moving with the rotor.

It shows the static pressure measurement samples of rotor tip on the rotating frame in the time domain with Fig. 11. The abscissa axis stands for the simulation time, and the ordinate axis is nondimensional static pressure. The stall cell appeared at a certain state, and spread in circumferential direction. Because of the higher rotor rotating speed than the stall speed, the stall region rotating direction is opposite to the rotor rotating direction in the rotating frame.

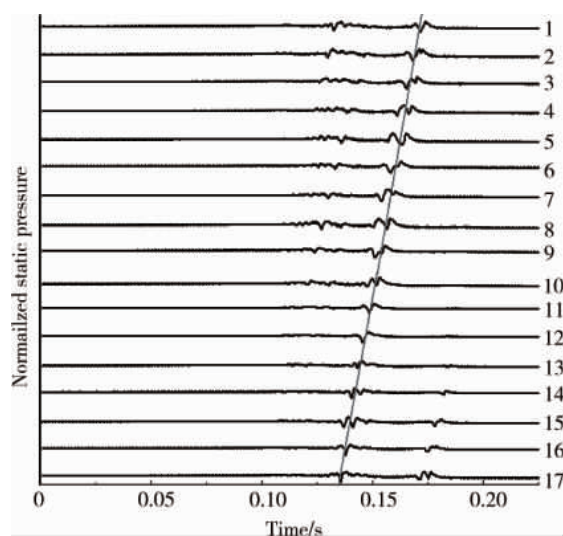


Fig. 11 Static pressure measured by 17 points on rotating frame

Then spectra analysis is carried out to analyze the frequency on the stall signal time history. It is found that a low frequency (32.47 Hz) appears with high amplitude in the amplitude spectrum of Fig. 12, which means the speed of the stall cell is about 65% of rotor rotating speed in the opposite direction. As the rotor rotating at 3000 r/min, the frequency of measuring point can be calculated to be 50 Hz. So in the stationary frame, one stall cell appears to rotate about the eigenfrequency of 17.53 Hz, at an angular

velocity of 35% to that of the angular velocity in the rotating direction (Fig. 13). It is in good agreement with the experiment results.

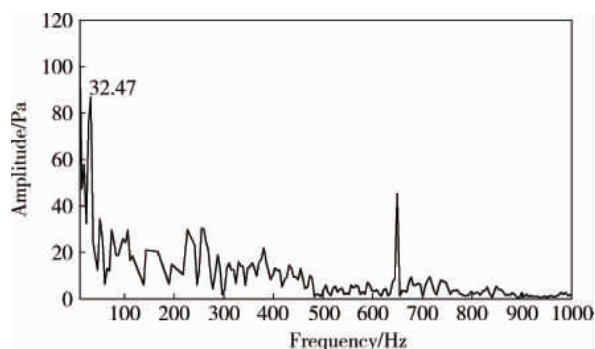


Fig. 12 Amplitude spectrum

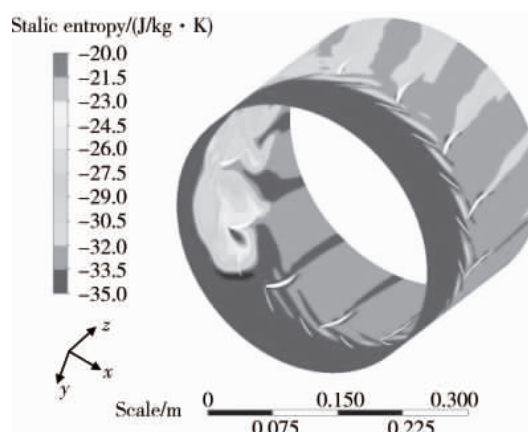


Fig. 13 Stall cell in the compressor

Analysis on the blade deformation chart (Fig. 14) obtained by the coupled method at state D shows that the blade vibration mode is similar to the first bending mode (Fig. 7). The node of maximum deformation on the blade tip leading edge is selected according to the characteristics of different states of the rotor, on which

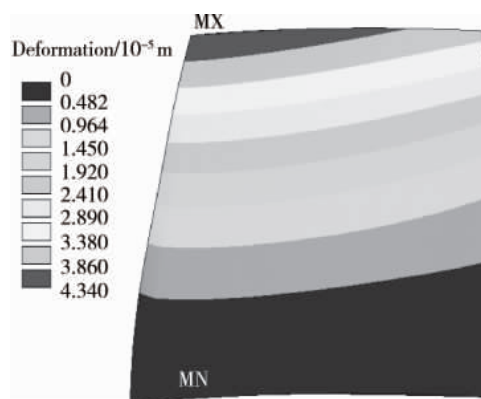


Fig. 14 Blade deformation chart

basic characteristics of spatial and temporal vibration are analyzed. The curve of the blade tip vibration at different mass flow states is shown in Fig. 15.

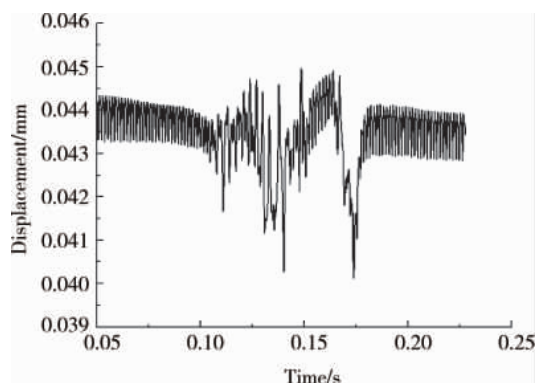


Fig. 15 Blade tip vibration curve

Rotating stall is a local phenomenon in which a circumferentially uniform flow pattern is disturbed. Stall may produce large spatial periodic flow fluctuations within the compressor. Forced vibration can arise from the excitation produced by passing through rotating stall cells. And if the excitation frequency of the rotating stall were to be equal to or some fraction of the natural frequency of any of the blades, a resonant vibration would occur.

Although there is no crossing point between the stall rotating frequency and the blade first natural flexural (1F) or torsion (1T) frequency in the Campbell diagram (Fig. 16), the influence on the blade forced response by the rotating stall can not be ignored. When pressure fluctuation occurs, it acts as a periodic force on the

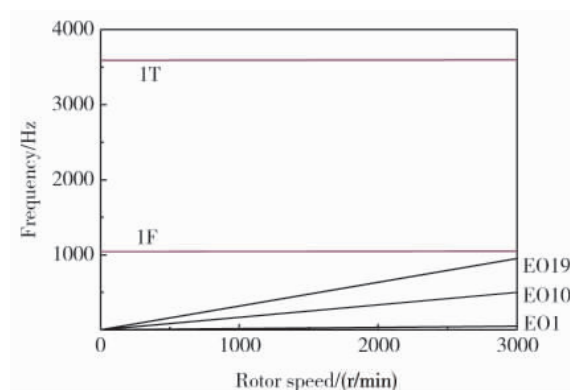


Fig. 16 Campbell diagram (EO:engine order)

blades. The amplitude of the resulting vibratory stress would, of course, depends on the strength of the excitation force due to rotating stall and the amount of damping in the blading. This produces blade vibrations of significant magnitude. Analysis on the stress at different operating states of the rotor (Fig. 17) by selecting a key point of maximum stress at blade root (Fig. 18) shows the same phenomenon that the rotating field stall plays a significant role on the blade vibration and stress influenced by the flow excitation.

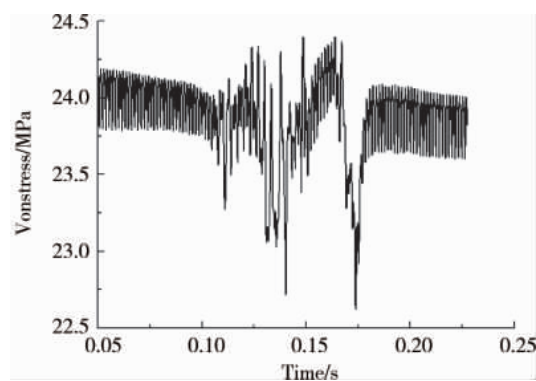


Fig. 17 Stress change of a key point at blade root

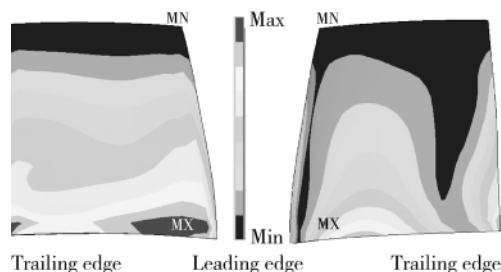


Fig. 18 Blade stress chart
(left: pressure side; right: suction side)

4 Conclusions

This paper takes the single stage and low speed axial compressor as the research object, presents a fluid-structure coupled aeroelastic method on blade forced response in turbomachinery. Compared with experimental results, the current study demonstrates the capabilities of the coupled computational approach for investigating blade aeroelastic response induced by the rotating stall flow. The conclusions can be summarized as follows.

1) For the research compressor of this paper, the experimental results indicate that one stalled region appears to rotate about the eigen-frequency of 17 Hz, at an angular velocity which is 1/3 to that of the angular velocity, and the spread direction is the same with the rotor rotating direction.

2) Rotating stall is a local phenomenon in which a circumferentially uniform flow pattern is disturbed, it acts as a periodic force on the blades. The excitation induced by passing through rotating stall cells can cause the blade forced response, producing blade vibrations of significant magnitude. The results indicate that the rotating field stall plays a significant role on the blade vibration and stress influenced by the flow excitation.

3) Rotating Stall is known to cause dangerous compressor and blade vibrations. While the pressure, velocity and other aerodynamic signals are used to established relationship with the rotating stall phenomenon, the magnitudes of blade vibration and stress may tend to be closer corresponding to the stall inception process. These are worth an effort to be taken on in the following works.

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