Numerical study on air-structure coupling dynamic characteristics of the axial fan blade

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Abstract: In order to understand the dynamic characteristics of the axial-flow fan blade due to the effect of rotating stress and the action of unsteady aerodynamic forces caused by the airflow, a numerical simulation method for air-structure coupling in an axial-flow fan with fixed rear guide blades was performed. The dynamic characteristics of an axial-flow fan rotating blade were studied by using the two-way air-structure coupling method. Based on the standard k-ε turbulence model, and using weak coupling method, the preceding six orders modal parameters of the rotating blade were obtained, and the distributions of stress and strain on the rotating blade were presented. The results show that the modal frequency from the first to the sixth order is 3Hz higher than the modal frequency without considering air-structure coupling interaction; the maximum stress and the maximum strain are all occurred in the vicinity of root area of the blade no matter the air-structure coupling is considered or not, thus, the blade root is the dangerous location subjected to fatigue break; the position of maximum deformation is at the blade tip, so the vibration of the blade tip is significant. This study can provide theoretical references for the further study on the strength analysis and mechanical optimal design.

1. Introduction
Fan is a kind of common fluid machinery and has been widely used in mine ventilation, petrochemical industry, metallurgy, energy and power, transportation and many other fields. The running of fans has great effects on production safety and efficiency of the enterprise, the stability problem of the blade that is closely related to the safe operation of fan has become an important research subject in recent years.

The structure vibration, stress and deformation of the blade that were resulted from air-structure coupling will influence the blade strength and its working stability and safety, especially large axial flow fan blades. The coupling vibration is a nonlinear vibration problem that exists in all turbo-machinery[1].

Systematic study about the dynamic characteristics of fan blade is still very few at home and abroad[2]. Research content of this paper is a frontier and hotspot issue in the field of the turbo-machinery, and its research progress will help people understand the air-structure coupling deeply in the turbo-mechanical flow and the mechanism of blade flutter.

2. Air-structure coupling principle and vibration control equation
Interaction of the structure and the air is the air-coupling \cite{3}, which makes their respective forms, mass, momentum and energy change. Air-structure coupling principle is to combine the Navier - Stokes equations of gas with the principle of solid-virtual work and to calculate their changes \cite{4}. 

Vibration differential equation of the general expression: 

$$M\ddot{q} + C\dot{q} + Kq = F$$  \hspace{1cm} (1)$$

Where $M$ is structural mass matrix, $q$ is structural displacement vector, $C$ is the damping coefficient matrix, $K$ is the structural stiffness matrix, and $F$ (centrifugal force and aerodynamic force, etc.) is external load vector.

The fan blades rotating at high speed will produce large pre-stress; and the mode and frequency of the rotor blades are different from that in static state. For the effect of rotating effect of coordinate system of blade modal, the incremental potential energy principle studying of the Lagrange coordinate system was adopted in this paper. The finite element formula for deformation was established based on the FEM deformation theory of geometrically non-linear problems, then these factors were put into stiffness matrix of the blade structure and in turn stacked them to the stiffness matrix of the blade structure. Thus blade dynamic equation of the flow field force and rotational speed considered was forming \cite{5}.

$$
\begin{bmatrix}
M_s & 0 \\
\rho_f B & M_f
\end{bmatrix}
\begin{bmatrix}
\ddot{u} \\
\dot{p}
\end{bmatrix} + 
\begin{bmatrix}
C_s & 0 \\
0 & C_f
\end{bmatrix}
\begin{bmatrix}
\dot{u} \\
\dot{p}
\end{bmatrix} + 
\begin{bmatrix}
K_s & 0 \\
0 & K_f
\end{bmatrix}
\begin{bmatrix}
{u} \\
{p}
\end{bmatrix} = 
\begin{bmatrix}
{R}_s \\
0
\end{bmatrix}
$$  \hspace{1cm} (2)$$

Where $M_s$ is structural mass matrix; $K_s$ is the stiffness matrix; $C_s$ is damping matrix, $M_f$ is fluid mass matrix, $K_f$ is fluid stiffness matrix, $C_f$ is the fluid damping matrix, $B$ is the fluid-structure coupling matrix, $\rho_f$ is the fluid’s density, $u$ is the structural vibration displacement; $p$ is the fluid pressure; $R_s$ is the vector loading on the structure.

3. Model building and meshing

Due to large amount of calculation in full channel, so the flow model was simplified to explore the calculation method and reduce the amount of calculation in this paper, and a single channel model was used for calculating \cite{6}. In the flow field calculation, the periodic boundary conditions were adopted to simulate the flow field of the fan, while the effect of tip clearance was ignored. Calculation model is shown in figure 1.

Unstructured tetrahedral volumetric unit was used to the discretization of the solution domain. The grid independence verification has been performed in order to get relatively reasonable number of grid, finally the grid number was determined \cite{1}. The minimum size of grid is 1.7 mm, the maximum size is 338 mm and the grid node number is 187345. Dense grid nodes were arranged in the vicinity of moving blade and guide vane surfaces and large curvature regions. Mesh generation for the solution domain is shown in figure 2.
4. The numerical calculation and the result analysis

4.1. Numerical computation method
Fluid-structure interaction mechanics is a branch of mechanics generated by fluid mechanics and solid mechanics crossed. It is mainly study the various behaviors of solids, solid flow field deformation and the influence of movement in the flow field[8].

The choice of numerical simulation method is crucial for flow field simulation accuracy and efficiency. Turbulent Numerical Simulation method is divided into two categories: Direct Numerical Simulation method (the DNS for short) and the Non-direct Numerical Simulation method. The Direct Numerical Simulation method is directly used to solve instantaneous turbulent flow control equation, while Non-direct Numerical Simulation method is not directly adapted to calculation of turbulent with pulsation characteristics. Turbulence will be made an approximate or simplified.

Reynolds average method adopted in this paper is one of Non-direct Numerical Simulation methods. Calculation model by using two equation models: the $k$-$\varepsilon$ model. It can well handle the high strain rate and larger flow streamline curvature degree. So the standard - model has good adaptability to calculation of rotating machinery fluid[9]. Two-way gas-solid coupled numerical calculation method was adopted[10], the blade's speed is 600r/min, air flow is 130m$^3$/s, static pressure is 2500Pa, and installation angle is 38deg. The material of the blade is aluminium alloy, the material properties are shown in table 1.

| Table 1. Material properties of aluminium alloy |
|-----------------------------------------------|
| Density (kg/m$^3$) | Young's modulus (Pa) | Poisson's ratio |
| 2770              | 7.1×1010              | 0.33           |

4.2. The result analysis
The root of the blade was set as a fixed constraint. The first six orders modal frequencies were calculated through modal analysis, respectively, for the static and rotating blade, as shown in table 2.

| Table 2. The first six orders natural frequencies comparison between static and rotating blade |
|-----------------------------------------------|
| Order | Static modal frequency(Hz) | Rotating modal frequency (Hz) | Increase coefficient |
|-------|-----------------------------|-------------------------------|----------------------|
| 1     | 123.67                      | 126.93                        | 2.64%                |
| 2     | 462.94                      | 463.54                        | 0.13%                |
| 3     | 579.84                      | 586.12                        | 1.1%                 |
| 4     | 921.09                      | 930.13                        | 0.98%                |
| 5     | 1163.6                      | 1165.1                        | 0.13%                |
| 6     | 1485.4                      | 1488                          | 0.18%                |
When both the effects of rotating pre-stress and air-structure coupling are considered, the rotating modal frequencies from the 1st to the 6th order are higher than the corresponding static modal frequencies among 0.6~9Hz, but the rate of increase is relatively small.

Table 2 shows the maximum deformation as the blade is rotating and static, respectively. Compared with the maximum deformations of the stationary blade for the six orders, the maximum deformations of rotating blade increase with varying degrees as the rotating pre-stress of the blade is considered.

| Order | First order | Second order | Third order | Forth order | Fifth order | Sixth order |
|-------|-------------|--------------|-------------|-------------|-------------|-------------|
| Maximum deformation of static blade (mm) | 10.256 | 19.913 | 14.132 | 10.694 | 37.025 | 48.117 |
| Maximum deformation of rotating blade (mm) | 17.363 | 33.954 | 23.192 | 18.931 | 60.807 | 82.130 |
| Increase coefficient | 69.3% | 70.5% | 64.1% | 77.0% | 85.8% | 70.7% |

When the air-structure interaction is considered, the deformation of the blade is shown in figure 3.

**Figure 3.** The deformation of blade as $Q=130\text{m}^3/\text{s}$

The changing curve of the moving blade’s deformation with the time is shown in figure 4. It can be seen that displacement of the moving blade decreases gradually to a steady value, in other words, the blade is in a normal working state, and the flutter would not occur.

The FFT (Fast Fourier Transformation) and spectral analysis are used to the vibration displacement response of blade tip, and the displacement response spectrum gained is shows in figure 5.

**Figure 4.** The deformation curve over time of the moving blade when $Q=130\text{m}^3/\text{s}$  
**Figure 5.** The displacement response at the blade tip spectrum
It can be seen from the figure 5 that the blade oscillates most gravely in 138.9Hz when the air flow is 130m$^3$/s. It can be known from the previous simulation calculation that the static and rotating modal frequency of the first order are, respectively, 123.67Hz and 126.93Hz. Therefore, sympathetic vibration would not occur in this working condition.

The stress and strain distributions of the moving blade are shown in figure 6 and figure 7.

Figure 6. The stress distribution of the moving blade as $Q=130m^3/s$

Figure 7. The strain distribution of the moving blade as $Q=130m^3/s$

It can be seen that the maximum stress and the maximum strain are all occurred in the vicinity of the blade root, i.e. the root area of the blade is the dangerous location where fatigue failure is much more apt to occur.

5. Conclusions
The two-way air-structure coupling method has been applied to the simulation of the transient dynamic characteristics of the axial-flow fan blade. And its working condition was that rotate speed was 600r/min, air flow was 130m$^3$/s, static pressure was 2500Pa, and its established angle was 38deg. The main results were listed as following:

1. When both the effects of rotating pre-stress and air-structure coupling are considered, the rotating modal frequencies from the 1st to the 6th order are higher than the corresponding static modal
frequencies in the range of 0.6~9Hz and the rate of increase is small. While the deformation is considerably increased for the rotating blade.

(2) The maximum deformation is occurred in the vicinity of blade tip. The most serious frequency here is 138.9Hz. Resonance and flutter will not occur for working conditions in the study.

(3) The root of the blade is the dangerous location where fatigue failure is much more apt to occur.

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Acknowledgments
The authors are grateful to professor Y J Zhang, the head of Fan Science and Technology Research Institute of SDUST, for his technical expertise throughout this research. The research reported here is financially supported by the Special Funds of "Taishan Scholars" Project approved by the Government of Shandong Province, and the Instructive Project on Scientific and Technological Research approved by China National Coal Association under No. MTKJ 2011-366.