Numerical Simulation and Modal Analysis of External Flow Field of Large Wind Turbine

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Abstract. Aiming at the problems of complex and slow optimization process of large-scale wind turbine blade design and the influence of wind turbine flutter, etc. The numerical simulation analysis and modal analysis of the external flow field of the wind turbine under the rated wind speed are carried out by the hybrid method. A fluid simulation model based on CFD was established. The method of rotating reference frame and SST k-omega turbulence model are used to simulate the external flow of fan wind turbine in FLUENT. The first 6 natural modes of the wind wheel are calculated in Hypermesh/Optistruct. The results show that the hybrid simulation method can quickly obtain the impeller flow field information, and the flow of the tip portion is extremely unstable. In each natural mode, the blade deformation position is basically from the maximum chord length to the tip position. The closer to the tip, the more obvious the blade vibration deformation is. At the same time, the first-order natural frequency of the wind wheel is 0.919Hz, which is greater than 0.382Hz when the rated speed is 12m/s, thus effectively avoiding resonance phenomenon.

1. Introduction

The development of new energy sources such as wind, nuclear, solar and geothermal energy has become a hot issue in the world today. Compared with other energy sources, wind energy has obvious advantages. It has a large amount of reserves, 10 times that of water energy, and it is widely distributed and never exhausted. It is especially important for islands and remote areas with inconvenient transportation and far away from the main power grid. The most common form of utilization of wind energy is wind power [1-2]. There are two ideas for wind power generation, horizontal axis fans and vertical axis fans. At present, horizontal axis fans are widely used and are the mainstream models of wind power generation [3].

The blade is the main component of the wind turbine energy conversion, and the wind mill composed of the blade and the hub is an energy capture mechanism for converting wind energy into mechanical energy. Blade performance determines wind turbine efficiency, load characteristics and noise levels [4-5]. At present, the traditional design method for wind turbine blades has a large workload, low analysis efficiency, and long design period. The CFD method is widely used in the study of the aerodynamic performance of the blade. The flow field change through the wind wheel and the aerodynamic force acting on the blade are calculated to provide a theoretical basis for the optimal design of the blade [6]. For a wind wheel composed of a plurality of blades, in addition to vibration generated by a single blade, interaction occurs between the blades, resulting in coupled vibration [7-9]. The modal analysis of the wind wheel avoids the calculation error caused by studying the single blade...
only and not fully understanding the overall shape of the wind wheel vibration. At the same time, some aeroelastic instability phenomena can be found in the simulation process, and the safety hazards existing in the wind turbine blades are eliminated in time.

Taking a 1.5MW large-scale wind turbine as the research object, the aerodynamic characteristics of the blade were simulated by CFD and rotating coordinate system method to quickly and intuitively understand the changes of rotating fluid flow state and surface pressure around the blade under the action of rated wind speed. By analyzing the first 6 natural modes of the wind wheel, the natural frequencies and corresponding vibration modes are obtained to prevent fatigue damage caused by stiffness problems from affecting the blade life.

2. Establishment of model
The wind wheel model includes a three-dimensional solid model and a calculation model.

The technical parameters of the designed 1.5MW horizontal axis wind turbine are shown in Table 1.

| parameter               | Numerical value | parameter               | Numerical value |
|-------------------------|-----------------|-------------------------|-----------------|
| Cutting into the wind   | 4               | Cut out the wind velocity| 25              |
| velocity (ms\(^{-1}\))  |                 | (ms\(^{-1}\))          |                 |
| Rated wind velocity     | 12              | rated power (MW)        | 1.5             |
| (ms\(^{-1}\))          |                 |                         |                 |
| Wind wheel diameter     | 70              | Blade length (m)        | 34              |
| (m)                     |                 |                         |                 |
| Number of blades        | 3               |                         |                 |
| (slice)                 |                 |                         |                 |

The shape of wind turbine blade is a complex curved surface, which requires high modeling accuracy. Based on the blade element theory [10], the theory assumes that a wind turbine blade of finite length is divided into many micro-segments, and the blade element unit itself is small enough to be equivalent to an airfoil[11]. The blade surface created by CATIA is shown in Figure 1, and the solid model is shown in Figure 2.

![Figure 1. Blade surface](image1)

![Figure 2. Wind wheel solid model](image2)

The calculation domain is divided into rotation domain and static domain, and the dynamic domain that closely clings to the blade surface and rotates with the blade is defined as rotation domain. The size of the rotation area is a cylindrical area with a radius of 40 meters and a height of 10 meters, the rotation axis is the Y axis, and the whole impeller is placed therein. The stationary region is a cylindrical region with a radius of 105 meters and a height of 280 meters, which covers the rotating region and the impeller. The entrance to the stationary domain is 70 meters from the blade rotation center and the exit is 210 meters from the blade rotation center. Based on the Moving Reference Frame method, the flow field information of the two regions is transmitted through the interface. The stationary domain and the rotating domain are all divided by tetrahedral unstructured grids. There are 3 million units in the stationary domain and 5 million units in the rotating domain. The specific
3. Flow field analysis

Characteristics and flow field analysis of blades operating under rated conditions. When the incoming wind speed is 12m/s, the blade flow field information.

3.1. Calculation condition setting

The specific settings of the calculation domain boundary conditions are shown in Table 2.

| Computation domain boundary | Numerical valu                                                                 |
|-----------------------------|-------------------------------------------------------------------------------|
| Inlet                       | Velocity inlet, $u=12\text{ms}^{-1}$; Turbulence intensity, $I=10\%$; Turbulent length scale, $L=210\text{m}$ |
| Outlet                      | Pressure outlet, relative pressure, $P=0$; Turbulence intensity, $I=10\%$; Turbulent length scale, $L=210\text{m}$ |
| Wind wheel surface          | No-slip rotation wall boundary                                                 |
| Interface                   | INTERFACE surface, appearing in pairs                                          |
| Other wall                  | No-slip wall boundary                                                          |

For the working environment of the horizontal axis wind turbine, the governing equation of the incompressible viscous fluid flow can be used:

The continuity equation:

$$\nabla \cdot u = 0$$

(1)

The momentum equation:

$$\frac{\partial u}{\partial t} + (u \cdot \nabla)u = -\nabla p + \frac{1}{Re} \nabla^2 u$$

(2)

where:

$u$ represents the velocity vector; $p$ is the fluid pressure; $Re$ is the flow Reynolds number.

In combination with the characteristics of the wind wheel and the rotating flow field of the wind turbine, the flow separation is involved, and the boundary layer needs to be solved with high precision. Therefore, the standard k-epsilon model is chosen first in the calculation. This is a very robust turbulence model and is widely used. The model is used to perform initial iteration and parameter research. It can quickly converge and check the parameter settings and the grid model for the calculation. It can play an initial role in the study of the aerodynamic characteristics of wind turbines and speed up the research process. The parameters are set as shown in Table 3.

| Setting item      | Parameter setting                                                   |
|-------------------|---------------------------------------------------------------------|
| Turbulence model  | Standard k-epsilon model and SST k-omega model                      |
3.2. Flow field analysis results
The performance of wind turbine under rated wind speed is analyzed, and the variation laws of performance parameters such as blade surface pressure and speed are studied. The blade pressure surface velocity cloud diagram is shown in Figure 5. The static pressure cloud diagram of the blade pressure surface is shown in Figure 6.

From the pressure surface velocity nephogram, it can be seen that there is separation flow below the middle of the blade, and there is complete attachment flow from above the middle of the blade to the tip region, forming a good two-dimensional flow. On the whole, the airflow speed on the blade surface increases from the blade root to the blade tip. The maximum airflow speed appears at the blade tip, and the airflow speed at the blade tip is extremely unstable. This unstable airflow speed will generate aerodynamic noise.

From the static pressure nephogram of the blade pressure surface, it can be seen that the pressure surface is positive pressure, and the pressure is relatively high from the middle part to the tip part of the blade along the spanwise direction. This phenomenon is due to the rotation of the blade. Under the action of centrifugal force, the airflow moves along the blade to the tip direction and continuously presses towards the tip direction, thus increasing the pressure at the tip. Under the combined action of centrifugal force and additional Coriolis force, the airflow on the blade flows from the leading edge to the trailing edge along the chord direction, making the leading edge pressure small and the trailing edge pressure strong, causing the pressure near the tip trailing edge of the rotating blade to be larger in the complex flow field.

The blades studied are lift-type blades. The higher the lift, the better the aerodynamic performance of the blades, and the higher the power of the blades. Therefore, the pressure on the suction side of the blade is also important for the study. The static pressure cloud diagram of the suction surface of the blade is shown in Figure 7.

By comparing the static pressure nephogram of the pressure surface and suction surface, it can be seen that the overall pressure surface of the blade is positive pressure, and the closer to the tip of the blade, the greater the pressure is. The overall pressure on the suction surface is negative pressure, reaching the tip section at about 70% of the blade span. The middle part of the whole blade has the lowest negative pressure. The pressure difference between the blade root and the blade tip gradually increases, so that the blade can provide more lift force to the blade tip at 70%. From the root of the blade to 70% of the blade span, the overall aerodynamic performance of the blade is good on the...
The premise of ensuring the strength. The wind wheel meets the design requirements for aerodynamic performance.

![Static pressure map of suction surface](image1)

![Wind wheel finite element model](image2)

**Figure 7. Static pressure map of suction surface**

**Figure 8. Wind wheel finite element model**

### 4. Modal analysis

In addition to meeting the aerodynamic performance requirements, the wind wheel must also increase the natural frequency of the blade to prevent the blade from resonating. Resonance will cause obvious gear box front and rear swaying, strong shaking of the left and right sides of the nacelle, and yaw braking position, which will seriously damage the components inside the engine room. And the blade breaks. Therefore, it is necessary to study the vibration situation, that is modal analysis.

#### 4.1. Modal theory

Assuming that the wind wheel is under external force, the equation of motion of the system is

$$ M\ddot{x} + C\dot{x} + Kx = 0 $$

(3)

- $M$ is the mass matrix of the structure;
- $C$ is the damping matrix of the structure;
- $K$ is the stiffness matrix of the structure;
- $\ddot{x}$ is the acceleration vector;
- $\dot{x}$ is the velocity vector;
- $x$ is the displacement vector.

When no external load is applied to the system, the solution of the equation reflects the natural frequency and mode shape of the structure. Engineering discussion of the inherent characteristics of the blade usually ignores the damping effect, and equation (3) becomes:

$$ M\ddot{x} + Kx = 0 $$

(4)

The analytical formula of equation (4) is

$$ x = L \sin \omega t $$

(5)

$L$ is the modal shape; $\omega$ is the natural frequency (Hz); $t$ is the time (s). Bring formula (5) to (4):

$$ \left(K - \omega^2 M\right)L = 0 $$

(6)

Letting $\lambda = \omega^2$, the condition that the formula (6) has a non-zero solution is

$$ \det \left(K - \lambda M\right) = 0 $$

(7)

Equation (7) is a generalized characteristic equation, which is the n-order algebraic equation of $\lambda$.

Thus,

$$ \det \left(K - \lambda_i M\right) = 0 $$

(8)

where:

- $i=1,2,3…n$; solving the characteristic equation (8), the modal shape $L$ and the natural frequency $\omega_i$ can be obtained.

#### 4.2. Modal analysis result

The finite element model of the built impeller 3D model was established in Hypermesh, and the modal analysis was performed using Optistruct. The blade is fixed to the hub, and the TRIAS unit is used to draw the grid. The model has 101,236 units, and the finite element model of the wind wheel is shown in Figure 8.
In the actual working process of the wind wheel, the external wind force has few high-order excitations on the blade, mainly the low-order mode is excited, so it is necessary to study the low-order mode shape. The 6th-order mode of the wind wheel is extracted and analyzed. The frequency of the wind wheel is shown in Table 4.

Table 4. The first 6 natural frequencies of the wind wheel

| Modal order | Frequency (Hz) |
|-------------|---------------|
| 1           | 0.919         |
| 2           | 0.922         |
| 3           | 1.263         |
| 4           | 2.586         |
| 5           | 2.588         |
| 6           | 3.495         |

The formula for calculating the natural frequency of a wind turbine at rated wind speed is

\[ f = \frac{\lambda V}{\pi D} \]  

where:

\( \lambda \) is the tip speed ratio, \( \lambda = 7 \), \( V \) is the rated wind speed, \( V = 12 \text{ ms}^{-1} \), \( D \) is the diameter of the wind wheel, \( D = 70 \text{ m} \).

Calculated to get \( f = 0.382 \text{ Hz} \). The first-order natural frequency of the wind wheel is 0.919 Hz, which is greater than 0.382 Hz, and is also greater than 20% of the rotation frequency of 0.764 Hz, so that the resonance phenomenon can be effectively avoided.

The modal shape patterns of the first to sixth orders of the wind wheel are shown in Figure 9.

According to the first 6 natural frequencies and vibration patterns of the wind wheel, it can be known that the first vibration pattern is waving vibration, the second vibration pattern is waving vibration, the third vibration pattern is swinging vibration, the fourth vibration pattern is waving vibration, the fifth vibration pattern is waving vibration, and the sixth vibration pattern is waving pendulum vibration composite vibration. With the increase of external excitation, the mode of vibration of the blade becomes more and more complex and the amplitude of the blade becomes larger and larger. The influence positions are basically from the maximum chord length of the blade to the tip position, and the closer to the tip, the greater the deformation. Therefore, it is necessary to pay attention to the stiffness from the maximum chord length of the wind turbine blade to the tip position. In order to prevent fatigue damage during work, the stiffness can be strengthened to prolong the service life of the wind turbine blade.

5. Conclusion

(1) The traditional design and calculation method for wind turbine blades has large workload and low analysis efficiency. CFD method can be used to obtain wind rotation field information quickly within certain accuracy requirements. The scheme can be adjusted in time during the blade design process and reduce the blade design period.
(2) Blade mode of vibration is mainly waved and shimmied, with strong anti-torsion ability. In actual production, the stiffness from the maximum chord length to the tip of the blade can be properly strengthened to prevent fatigue damage and prolong the service life of the wind turbine blade.

(3) The natural frequency of the blade at the rated speed is only related to the speed of the wind wheel. In the actual speed adjustment process, full consideration should be given to avoid resonance.

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