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Method of Modal Analysis for Wind Turbines Based on Air-structure Coupling Vibration Model

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Abstract: In order to explore the modal characteristics of the wind turbine, aiming at 1500 kw variable speed variable pitch double-fed wind turbines which are the mature products in market, established the whole wind turbine dynamic modal equations by Lagrange method and obtained modal parameters through decoupling calculation. Then process on-site testing to obtain measured modal characteristic parameters. The results of both analytical dates show that deviation between theoretical data and measured data is less than 10%. The results show a good consistency, the theoretical model and the subsystem boundary conditions simplified methods are consistent with the actual situation, indicating a high reliability. The theoretical model verifies that natural frequencies of the major structural components for testing wind turbine within the operation speed range can avoid coupled resonance phenomenon which meets GL specification requirements. It is can effectively conduct the wind turbine development, design and optimization.

Key words: Wind turbines, dynamical model, modal analysis, air-structure, optimization

INTRODUCTION

As a new green energy, wind power generation has been received much attention. By the end of 2011, the national total installed capacity has reached to 62.36 GW (Li, 2012; Broggi and Cattani, 2006) which maintains a leading position in the total wind-capacity. However, the operation stability and generating efficiency of domestic wind turbines are not satisfied. For the problems of design accuracy, material quality and rationality of assembly and installation, there is a certain deviation between the design and the actual situation (Arnay *et al.*, 2009; He, 2006). And coupled vibration fault is the great hidden danger of turbines' safe operation.

In order to ensure the safe operation of turbines, the natural and exciting frequency of main components does not cross or overlap according to the requirement of industry vibration standard and GL specification. If cannot avoid, corresponding actions must be taken to ensure that the units quickly pass the intersection and the amplitude is controlled effectively, so that coupling resonance phenomenon is avoided (Li *et al.*, 1999; Ni *et al.*, 2008). But the modal analysis method of components and the recommended value of natural frequency are not given.

The current research on modal characteristics of wind turbines is based on analytic mechanics, multi-body dynamics and finite element analysis (Thomas and Arlo, 1983; Wright *et al.*, 1999; Lee *et al.*, 2002; Gentile and Saisi, 2007). The modal analysis of the unit and main components is completed through the simulation or experimental method (Yao and Zhao, 2009; Yao *et al.*, 2011). However, the wind turbine operates always under atrocious conditions. Its subsystems have complex connecting conditions of boundary and interaction among their operations. What's more, there is lack of measured data and it is difficult to obtain reliable modal parameters of wind turbine only by simulation or experimental methods.

Therefore, the research focused on the megawatt wind turbine in order to explore its modal characteristics. The unit dynamics equation was established using Lagrange method and the modal parameters were obtained by decoupling. The measured modal parameters have been got through the following measurement in the wind field. It was studied whether the boundary simplification of theoretical model and subsystems fits the actual situation. The deviation between theoretical data and measured data and the stability analysis of unit were discussed. Coupled vibration model of wind turbine is shown in Fig. 1.

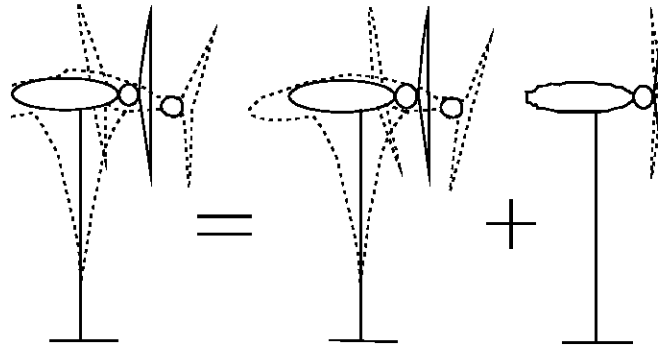


Fig. 1: Coupled vibration model of wind turbine

MODEL AND DEFINITION

Rotor aerodynamic model: To apply the Lagrange method to establish the dynamic equation of wind turbine, firstly the centroid of turbine is defined as the generalized origin of coordinate, its axial direction as X axis, the transverse direction of the unit as Y axis and Z axis is vertically upward. Then the coordinate system of blades, hub and tower is defined as shown in Fig. 2. Finally x_z is defined as a fore-aft displacement of wind wheel system, β as blade deformation angle, γ as blade elastic deformation, y_z as the fore-aft displacement of tower coupled system, w as the speed of wind wheel and F as external wind load. The dynamic model of wind wheel system is established as the Eq. 1-3:

$$\begin{aligned} (3m_1 + m_2)\ddot{x}_z + 3S\ddot{\beta}\cos\alpha + 3m_1\ddot{\gamma}\cos\alpha \\ - 6S\omega\dot{\beta}\cos\alpha - (3S\omega^2\cos\alpha + F\sin\alpha)\dot{\beta} \\ = F\cos\alpha - 3S\omega^2\sin\alpha \end{aligned} \quad (1)$$

$$\begin{aligned} 3S\ddot{x}_z\cos\alpha + 6I\ddot{\beta} + 3S\ddot{\gamma} + \frac{3}{2}m_1\ddot{\gamma} \\ + 3c_n\dot{\beta} + 3k_n\beta + 3S\omega^2\gamma = FL \end{aligned} \quad (2)$$

$$3m_1\ddot{x}_z\cos\alpha + 3S\ddot{\beta} + 3m_1\ddot{\gamma} - \frac{3}{2}m_1\dot{\beta} + 3\frac{k_n}{l}\beta = F \quad (3)$$

Tower system dynamics model: In view of the structure characteristics and the ratio of length to width of blades, wheel, engine room and the tower, the assumption could be made as followed: (1) Blades and tower are elastic beams which mass lumps on the central shaft. (2) Wheel and engine room are regarded as rigid mass concentration. (3) Hub and spindle, nacelle and tower are rigidly connected to the rigid foundation. Tower coupled system dynamics model established as the Eq. 4:

$$\begin{aligned} \frac{1}{12}m_3(l^2 + w^2)\ddot{y}_z + \frac{1}{2}m_4(2r^2 + t^2 - 2rt)\ddot{y}_z + c_t\dot{y}_z + \frac{3\pi E}{256h^3}(2rt - t^2)y_z \\ = gS_z h - \frac{3\pi E}{256h^3}(2rt - t^2)L \end{aligned} \quad (4)$$

Air-structure coupling vibration model: Because the modal equations of the two subsystems are dependant of each moment of inertia, connection stiffness and boundary conditions, simplifying model boundary connecting conditions of coupled system can not only make the issue simpler, but also reflect the coupled movement. Blade, Hub and Tower top coordinate system of wind turbine is shown in Fig. 2.

As the fore-aft displacement of wind wheel and the tower coupled system are same, that means $x_z = y_z$. Solving the simultaneous equations (1-4) obtains the kinetic equation:

$$[K]\{\chi\} + [C]\{\dot{\chi}\} + [M]\{\ddot{\chi}\} = \{F\} \quad (5)$$

Where:

$$[K] = \begin{bmatrix} 3m_1 + m_2 + \frac{1}{12}m_3(l^2 + w^2) + \frac{1}{2}m_4(2r^2 + t^2 - 2rt) & 3S\cos\alpha & 3m_1\cos\alpha \\ 3S\cos\alpha & 6I & 3S \\ 3m_1\cos\alpha & 3S & 3m_1 \end{bmatrix}$$

$$\begin{bmatrix} F\cos\alpha - 3S\omega^2\sin\alpha + gS_z h - \frac{3\pi E}{256h^3}(2rt - t^2)L \\ FL \\ F \end{bmatrix}$$

$$[K] = \begin{bmatrix} \frac{3\pi E}{256h^3}(2rt - t^2) & -3S\omega^2\cos\alpha - F\sin\alpha & 0 \\ 0 & 3k_n & 3S\omega^2 \\ 0 & 0 & 3\frac{k_n}{l} \end{bmatrix} [C] = \begin{bmatrix} c_t & -6S\omega\cos\alpha & 0 \\ 0 & 3c_n & \frac{3}{2}m_1 \\ 0 & -\frac{3}{2}m_1 & \end{bmatrix}$$

$$\{\chi\} = \begin{Bmatrix} x_z \\ \beta \\ \gamma \end{Bmatrix}$$

The natural frequency is inherent characteristic of the system which is dependant of the mass, stiffness and

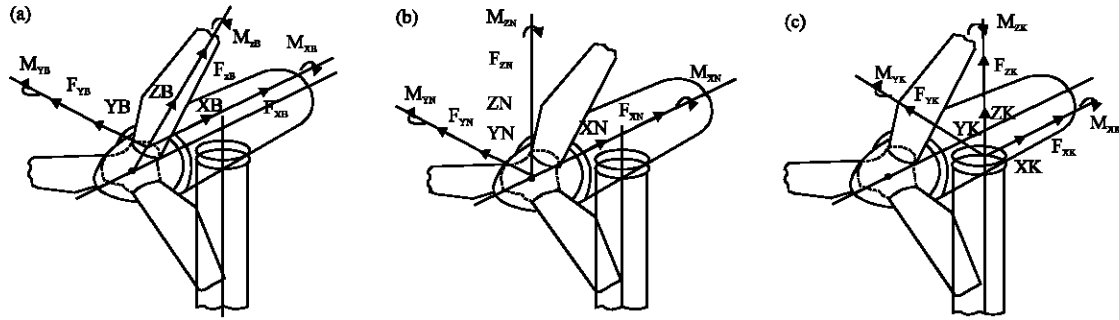


Fig. 2(a-c): Blade, hub and tower top coordinate system of wind turbine

Table 1: Natural frequency of wind turbine

Modal	Natural frequency (Hz)	Main formation
1	0.426	First-order fore-aft motion of tower coupled system
2	0.809	First-order flapwise motion of wind wheel system
3	1.505	First-order edgewise motion of wind wheel system

damping. The influence of air viscous damping of the machine is so small that its effect could be neglected. By setting $[C] = [0]$, $\{F\} = \{0\}$ the Eq. 5 reduces to:

$$[K]\{\chi\} + [M]\{\ddot{\chi}\} = 0 \quad (6)$$

Substitute into this:

$$\{\chi\} = \{A\} \sin(\omega t + \varphi) \quad (7)$$

Rearranging the Eq. 6 yields the main formation matrix $[A]$. Then the decoupling stiffness matrix and mass matrix could be yielded as:

$$[K_r] = [A]^T [K] [A] = \begin{bmatrix} K_1 & 0 & 0 \\ 0 & K_2 & 0 \\ 0 & 0 & K_3 \end{bmatrix}$$

$$[M_r] = [A]^T [M] [A] = \begin{bmatrix} M_1 & 0 & 0 \\ 0 & M_2 & 0 \\ 0 & 0 & M_3 \end{bmatrix}$$

The Eq. 7 reduces to:

$$(K_r - \omega_r^2 M_r) = [0]$$

where, K_r is the r th-order modal stiffness, M_r is the r th-order modal mass and ω_r is the r th-order modal frequency.

As a calculating object of the 1500 kW variable speed pitch doubly-fed models, its hub height is 65 m, the diameter of wind wheel 82 m, length of blades 40.3 m, range of wind wheel speed 9.9-17.4 rpm and rated speed 17.4 rpm. According to the stiffness matrix $[K]$ and

mass matrix $[M]$ of the 1500 kW unit, the natural frequency of wind turbine is found out as shown in Table 1.

TESTING OF WIND TURBINE

Testing of wind wheel system: To obtain the accurate natural frequencies of a wind wheel system under the field conditions, the 55# unit of a 1500 kW wind field in Zhangbei County in China, Hebei province was selected to be tested. ICP acceleration sensors (frequency response range of 0.2-25 kHz) were installed flapwise and edgewise at the position of about 20 meters to the blade root. The response data excited by natural wind was collected respectively by a data sampler (24 bit, AD/102.4 kHz) at zero-degree and 90-degree blade pitch angles. Wheel system test sensor distribution is shown in Fig. 3. The testing is shown in Fig. 4-7.

At zero degree pitch angle, the flapwise dynamic response data was more obvious. As illustrated in Fig. 4 and 5, the main peak frequency was the first-order fore-aft natural frequency of tower coupled system equal to 0.4 Hz and the first-order flapwise frequency of wind wheel system was 0.75 Hz. The torsional frequency of tower coupled system was 1.05 Hz and the first-order edgewise frequency of wind wheel system was 1.325 Hz. At 90 degree pitch angle, the flapwise dynamic response data is prominent. As shown in Fig. 6 and 7, the main peak frequency as the first-order fore-aft natural frequency of tower coupled system was 0.4 Hz and the first-order flapwise frequency of wind wheel system was 0.8 Hz. The torsional frequency of tower coupled system was 1.075 Hz and the first-order edgewise frequency of wind wheel system was 1.375 Hz.

Testing of tower coupled system: The dynamic model is established on the assumption that the foundation is rigid, while the stiffness of turbines' foundation is affected by soil, perfusion, reinforced concrete structure and etc. (Tony and Xin, 2007). In order to obtain the accurate natural frequency of tower coupled system under

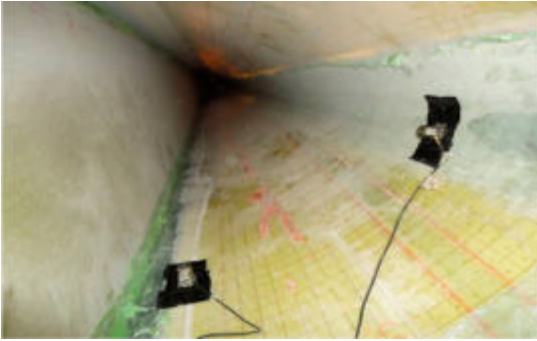


Fig. 3: Wheel system test sensor distribution

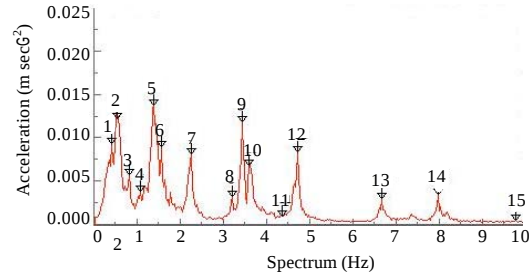


Fig. 6: Spectrum diagram of 55# wind turbine blade flapwise at 90 degree pitch angle

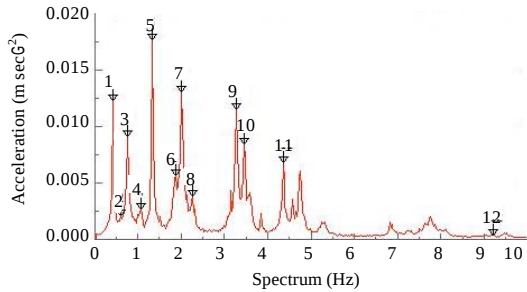


Fig. 4: Spectrum diagram of 55# wind turbine blade flapwise at 0 degree pitch

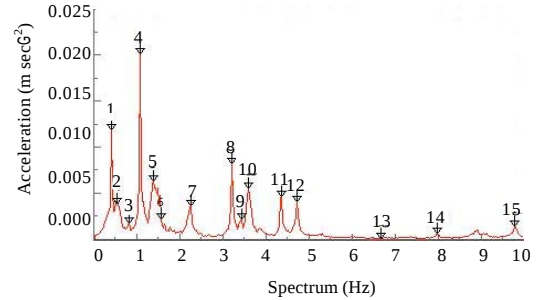


Fig. 7: Spectrum diagram of 55# wind turbine blade edgewise at 90 degree pitch angle

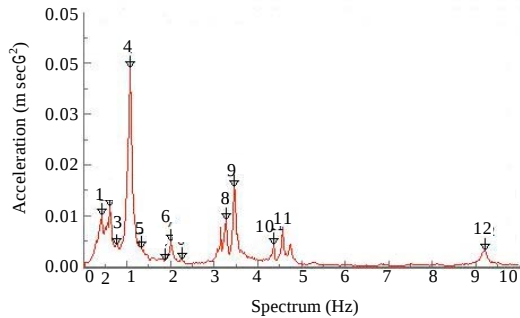


Fig. 5: Spectrum diagram of 55# wind turbine blade edgewise at 0 degree pitch

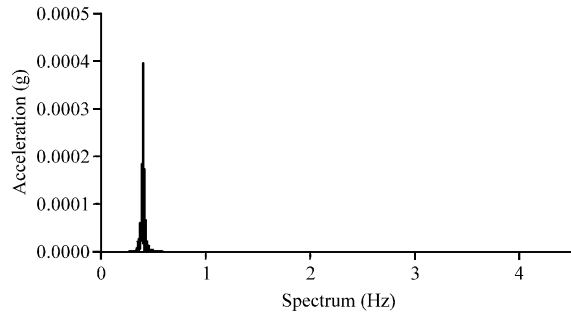


Fig. 8: Spectrum of 55# wind turbine fore-aft direction

the field conditions, ICP acceleration sensors (frequency response range of 0.2-3000 Hz) were installed at the position of about 0.5 m to the tower top, respectively in zero-degree and 90-degree directions of each ladder. Vibration data during operation and shutdown was acquired by using DEWERON 24-bit-AD-conversion-card data sampler. The result was shown in Fig. 8 and 9. The fore-aft and side-side first-order natural frequency of tower coupled system was about 0.42 Hz.

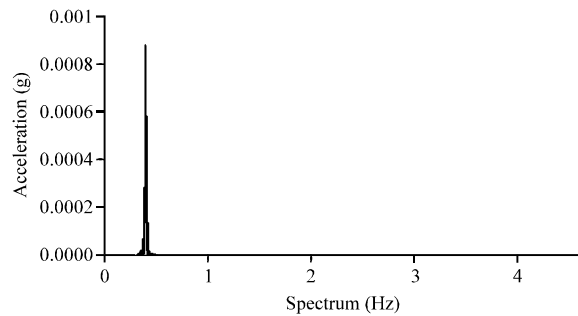


Fig. 9: Spectrum of 55# wind turbine side-side direction

DATA COMPARISON AND ANALYSIS

With the development of wind motor assembling machine capacity and scale more and more big, wind turbine fault rate is higher and higher also and wind turbine monitoring and fault diagnosis is more important. According to statistics of the failure of the wind farm in Sweden, Finland and Germany, the electrical system, blade and gear up to the site of the wind power system failure, gearbox failure site of the longest downtime in wind turbine failure (Tavner *et al.*, 2007; Ribrant and Bertling, 2007). According to the characteristics of complicated structure as well as fault probabilities and mechanism of the bottom and intermediate events in Wind turbine gear-box transmission system, fuzzy theory was introduced into fault tree analysis method, the fault tree analysis method of Wind turbine gear-box transmission system was proposed based on T-S fuzzy fault tree analysis (Wang *et al.*, 2013; Song *et al.*, 2005).

Data comparison: According to the calculating results of dynamic model, the first-order modal vibration of turbines mainly expressed fore-aft direction pitching motion of the tower coupled system. The second-order modal vibration mainly showed the flapwise motion of wind wheel system (outside of rotating plane). The third-order modal vibration mainly expressed edgewise motion of wind wheel system (inside of rotating plane).

As indicated as Table 2, the deviation of the dynamics calculation and the measured data of model are within 10% and there is a good agreement within both of them. That means that the simplified method of the dynamic model and the boundary condition is accordance with the actual situation which has high reliability.

The overall stability analysis: According to the measured data of wind turbine, the rotation frequency of 1P wind wheel is 0.165-0.29 Hz and 3P is 0.495-0.87 Hz. The first-order fore-aft and side-side natural frequency of tower coupled system is about 0.42 Hz. The first-order flapwise frequency of wind wheel system is about 0.75 Hz and the first order array frequency of wind wheel systems is about 1.375 Hz.

There is a large distance and no cross or overlap phenomenon among the first-order natural frequency of tower coupled system. The campbell chart of wind turbine is shown in Fig. 10. When the speed of wind wheel is 15 rpm, the 3P wind wheel and the first-order flapwise frequency of wind wheel has a point of intersection. Because the rated speed of the turbine is 17.4 and 15 rpm is only a node during the accelerating section, control strategy on this node is the combination of increasing bull chain damping and quick pass. By this, the vibration energy through the node is reduced effectively to ensure the safe and stable operation of the unit which is in accordance with the engineering requirements of design and operation.

Table 2: Comparison of theoretical data and measured data

Modal	Name	Theoretical data (Hz)	Measured data (Hz)	Error (%)
1	First-order fore-aft motion of Tower coupled system	0.426	0.420	1.4
2	First-order flapwise motion of wind wheel system	0.809	0.750	7.3
3	First-order edgewise motion of wind wheel system	1.505	1.375	8.6

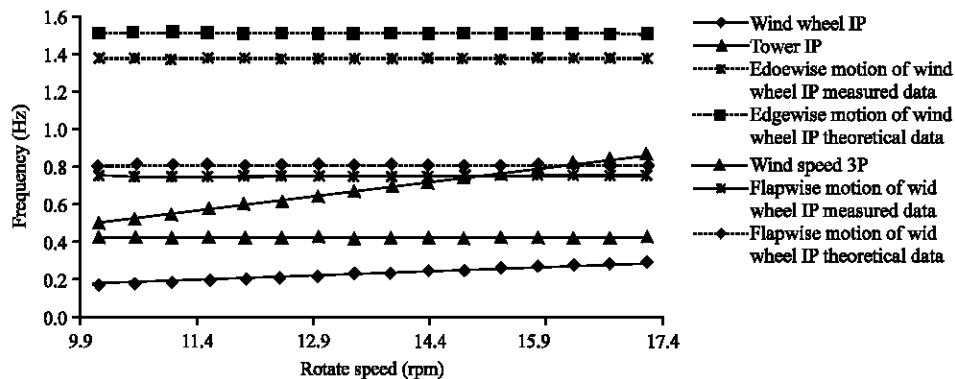


Fig. 10: Campbell chart of wind turbine

CONCLUSION

As megawatt wind turbines develop towards large scale and tower height increases, the overall stiffness and boundary connection stiffness of each subsystem decrease. Therefore, the phenomenon of unit coupled resonance may be occurred, affected by the excitation of the wind rotation frequency of 1P and 3P, the flapwise frequency of wind wheel system and the edgewise frequency of tower coupled system. By the means of establishment of wind turbines' dynamic model and field measurement in this paper, conclusions were drawn as following:

- The error between dynamics calculations and the measurement of model is within 10%. Both of them have a good agreement. That indicates that the simplified method of theoretical model and boundary condition of subsystems is accordance with the actual situation which has high reliability and can provide the theoretical foundation for development, design and optimization of turbines
- Using the theoretical model, the measured wind turbine is calculated and analyzed. It is testified that there exists no intersection and coincidence among the first-order natural frequency of tower coupled system, 1P and 3P wind rotation frequency and the first-order edgewise natural frequency of wind wheel system. The intersection between the rotation frequency of 3P wind wheel and the first-order natural frequency of the wind wheel system exists only if the speed of wind wheel is 15rpm. Because of this node in the accelerating section, the combination of increasing the bull chain damping and quick pass is applied as the control strategy to reduce the vibration energy through the node effectively which ensures the safe and stable operation of wind turbine and is in accordance with engineering requirements of design and operation

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