

Effects of aerodynamic fairing on full scale blade fatigue test

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Abstract. The reliability of large blades should be verified by means of full scale fatigue test. In order to solve the problem of lack of exciting force during fatigue test in the flap wise direction, the program that aerodynamic fairing is installed in the tip of blade to reduce the air resistance is proposed. The numerical model of blade vibration and damping ratio calculation is established. The relationship between damping ratio, exciting force and amplitude is constructed by finite element method respectively. The difference of the exciting bending moment of blade and the damping ratio before and after the installation of aerodynamic fairing is compared respectively. The results show that damping ratio decreased by 27.9%. When the vibration of the blade reaches the target bending moment, the exciting force of the equipment decreases by 45.4%. It is an effective way to reduce the exciting force.

1. Introduction

In accordance with the wind turbine certification requirements, the reliability of blade should be verified through fatigue test of full scale blade^[1]. The cycle of dynamic load applied through fatigue test is to monitor the blade operating state. Defects in the blade during the process of production would be exposed through fatigue test, such as wrinkle, lace of adhesive paste, rich resin^[2-6]. The state-of-the-art testing method for rotor blades in industry is based on resonance excitation where typically a dynamic mass excites the blade close to its first natural frequency^[7-10]. Due to lack of exciting force during fatigue test in flap wise direction, the blade is difficult to achieve the target bending moment. The air resistance has great impact on the damping ratio during blade fatigue test in the flap wise direction^[1]. Fabio^[11] et al. studied that macro-composites with star-shaped inclusions for vibration damping in wind turbine blades. Zhang^[12-13] et al. investigated that tuned liquid dampers (TLDs) for vibration control and damping edgewise vibrations of wind turbine blade. In this paper, in order to solve the problem of insufficient exciting force, the program that aerodynamic fairing installed in the tip to reduce the air resistance during the blade vibration is proposed.

The relationship between damping ratio, exciting force and amplitude is established by means of the finite element model in this paper, which is the theoretical basis to adjust the excitation force. Differences of bending moment, damping ratio and exciting force are compared between before and after the installation of the aerodynamic fairing by means of the fatigue test of full scale blade. This study provides an effective way to reduce the exciting force and fatigue loss of testing equipment.



2. Numerical model of Resonance and damping ratio calculation

2.1. Vibration equation of blade

It is assumed that the wind turbine blade test system is composed of the object of mass m and light spring. The test system is subjected to the elastic force of the spring ($F = -kx$), the air resistance ($R = -cv$) which is proportional to the speed. The periodic exciting force ($H = F_0 \cos \omega t$, $F_0 = ml\omega^2$) under the action of forced vibration. According to Newton's second law to get the test system's motion equation.

$$m\ddot{x} + c\dot{x} + kx = F_0 \cos \omega t \quad (1)$$

Set $\omega_n^2 = \frac{k}{m}$, $2\xi = \frac{c}{\sqrt{mk}} = \frac{c}{\omega_n m}$, substituting the above equation,

$$\frac{d^2 x}{dt^2} + 2\xi\omega_n \frac{dx}{dt} + \omega_n^2 x = \frac{F_0}{m} \cos \omega t \quad (2)$$

The general solution of the second order constant coefficient linear nonhomogeneous differential equation is,

$$x = \frac{F_0}{\omega_n^2} \frac{1}{2\xi} \cos\left(\omega t - \frac{\pi}{2}\right) \quad (3)$$

The amplitude is proportional to the exciting force and inversely proportional to the damping ratio.

2.2. Damping ratio solving equation

From the vibration response signal of the blade, the modal parameters reflecting the blade are obtained. Damping ratio defines the characteristic that the energy gradually reduced impacted by the resistance of the blade in the course of movement. The blade amplitude decays exponentially during the free vibration attenuation. The attenuation response can be expressed as,

$$x(t) = Ae^{-\zeta\omega_n t} \left[\frac{1-2\zeta^2}{1-\zeta^2} \sin(\omega_d t + \varphi) + \frac{2\zeta}{\sqrt{1-\zeta^2}} \cos(\omega_d t + \varphi) \right] \quad (4)$$

$$\omega_d = \omega_n \sqrt{1-\zeta^2}$$

Set the time before and after the adjacent vibration periods T are t_1 and t_2 , respectively. The ζ is tiny, so we can derive from eq. (8)

$$\frac{x_1(t)}{x_2(t)} = \frac{Ae^{-\zeta\omega_n t_1 \sin(\omega_d t_1 + \varphi)}}{Ae^{-\zeta\omega_n t_2 \sin(\omega_d t_2 + \varphi)}} \quad (5)$$

$$\text{Set } t_2 = t_1 + T, T = \frac{2\pi}{\omega_d}$$

$$\sin(\omega_d t_1 + \varphi) = \sin(\omega_d t_2 + \varphi) \quad (6)$$

The ratio of the amplitude of the vibration of two adjacent periods is called the attenuation coefficient.

$$\eta = \frac{x_1(t)}{x_2(t)} = \frac{x_n(t)}{x_{n+1}(t)} = \frac{Ae^{-\zeta\omega_n t_1}}{Ae^{-\zeta\omega_n(t_1+T)}} = e^{\zeta\omega_n T} \quad (7)$$

In Eq. (11), $x_k(t)$, $x_{k+n}(t)$ is the ratio of the amplitudes of adjacent n periods, damping ratio can be expressed as,

$$\zeta = \frac{1}{2n\pi} \ln \frac{x_k(t)}{x_{k+n}(t)} \quad (8)$$

The damping ratio of the blade can be determined from the amplitude attenuation of multiple cycles.

3. Results of FEM

3.1. The target amplitude converted from the blade target bending moment.

The length of blade is 56.6m. The blade tip is to be cut off at a radius of 48m in order to reduce the air resistance. Figure 1 shows the deflection of blade applied by gravity, dead weight and fatigue target bending moment, respectively. When the blade reaches the target bending moment, the tip deflection is 6.3m. The deflection which calculated from the FEM applied by target bending moment of fatigue is set as target amplitude of blade vibration.

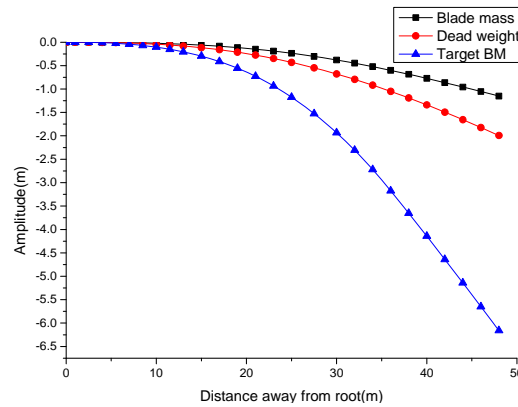


Figure 1. Deflection of blade in different cases

3.2. Effects of exciting force and damping ratio on vibration amplitude of blade

Figure 2 shows the linear relationship with a slope of $1.9\text{E-}3$ between the different exciting forces and blade amplitudes at the damping ratio of 0.217% during fatigue test in the flap wise direction. When the tip amplitude reaches 6.3m , the exciting force is 3300N . The phase transitions from 0 To 180 in the resonant frequency of 0.485 Hz .

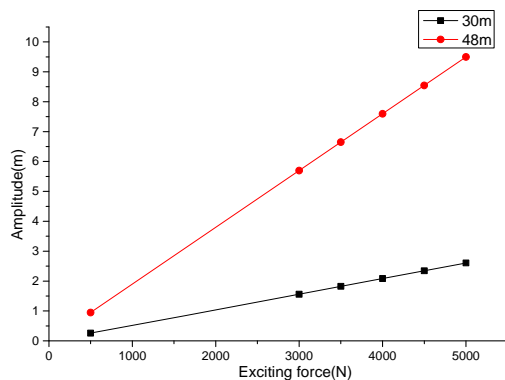


Figure 2. Amplitude vary exciting force curves

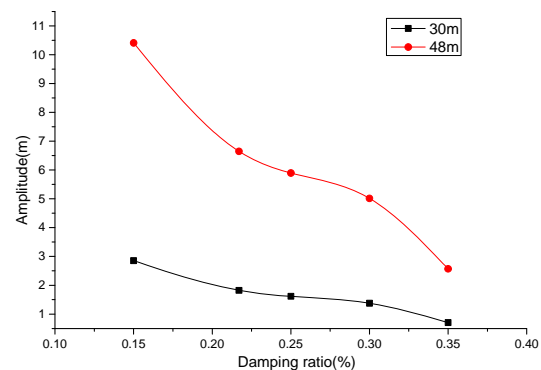


Figure 3. Amplitude vary damping ratio curves

The aerodynamic fairing installed in the tip of blade to reduce air resistance has great impact on the damping ratio during blade fatigue test in the flapwise direction. The stiffness and mass distribution of blade, shape of aerofoil, air temperature and humidity have a great influence on damping ratio, which will change during the test cycle. The relationship between the vibration amplitude of the blade under different damping ratio conditions is established at the exciting force of 3500N in figure 3, which provides theoretical basis for the adjustment of the exciting force and ensures that the blades are loaded cyclically on the target bending moment. The amplitude of the tip is 6.3m when the exciting force is 3500N and the damping ratio is 0.230% .

4. Results of experiments

The effects of aerodynamic fairing on the vibration bending moment are verified by fatigue test of full scale blade. The blade is mounted in the flapwise direction with the pressure side facing up and the suction side facing down. Exciter installed in the 25m away from the root. The aerodynamic fairing is installed in the tip from 38m to 48m areas. The strain gauges are bonded to the center of spar cap of each section of the blade. The force is applied at the tip of blade to produce bending moments in the each blade section. The coefficient that is the ratio of the bending moment to the strain in the blade section, by which the dynamic strain is transformed into the bending moment.

4.1. Vibration bending moment

The vibration bending moment in flap wise direction is the mean of the bending moment converted from strains on spar cap in the pressure side and the suction side. The target bending moments are enveloped by the vibration bending moments of the blade through adjusting the exciting force and the dead weight. Comparing the vibration bending moments before and after the installation of the aerodynamic fairing with the target bending moments in the area of the blade 4m to 36m fatigue test verification respectively, both which can envelop the target bending moments and meet the fatigue test specification requirements. Because of the weight of aerodynamic fairing, the ratio of vibration bending moments of the blade to target bending moments in the area of tip is higher than before, while in the area of root is lower than before.



Figure 4. Aerodynamic fairing and blade mounted in flapwise direction

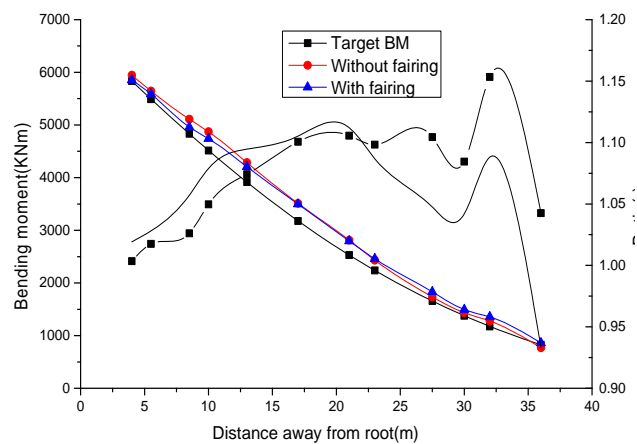


Figure 5. blade resonant bending moment with target value

4.2. Damping ratio and exciting force before and after installation of aerodynamic fairing

The damping of the blade is test in the flapwise direction, which include the structural damping and aerodynamic damping. The tip is released suddenly when the blade reach a certain amplitude, and then let the blade free vibration attenuation to within 1mm amplitude. The amplitude of the blade is acquired at a sampling frequency of 50 Hz. The damping ratio is obtained by the theoretical calculation formula (8).

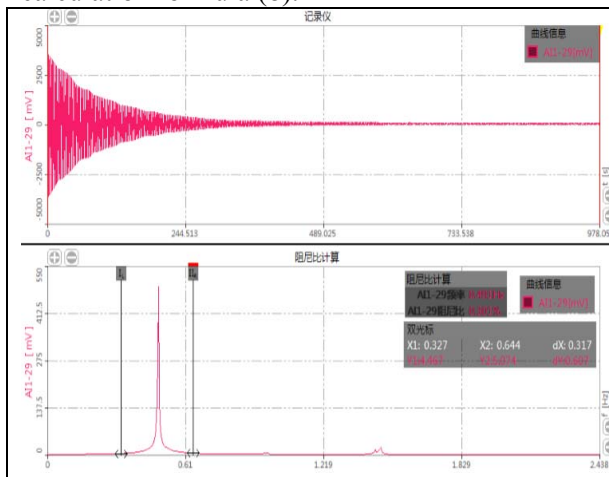


Figure 6.(a)damping ratio before installation of aerodynamic fairing

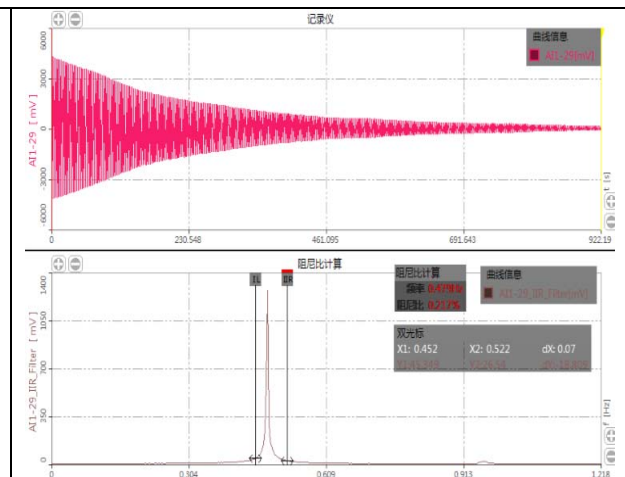


Figure 6.(b)damping ratio after installation of aerodynamic fairing

Table 1. Frequency and exciting force in flap wise direction

Case	Frequency (Hz)	Damping (%)	Exciting Position (m)	Mass(Kg)	Stroke (mm)	Exciting force (N)	Theory prediction (N)
1	0.491	0.300%	25	1936	330	6077	4185
2	0.479	0.217%	25	1890	195	3316	3300

Case 1 and Case 2 in Table 1 are the damping ratio change of the blade and the exciting force required to achieve the target bending moments before and after the installation of the aerodynamic fairing, respectively. Compared with the theoretical calculation, the excitation frequency of the test system is 0.491Hz before the installation of the aerodynamic fairing, and the exciting force is proportional to the square of the frequency. The dynamic mass stroke of the exciter is 330 mm, which is the maximum stroke the exciter can achieve. The friction that exciter has to overcome increases rapidly, so the vibration efficiency of the equipment dropped significantly. These two factors cause the actual exciting force higher than the theoretical excitation force.

The theoretical exciting force 3300N and the actual 3316N are basically consistent after the installation of the aerodynamic fairing. The mass of blade in the tip increases because of the weight of fairing is very sensitive to frequency, which lead to a decrease in blade frequency from 0.491 Hz to 0.479 Hz. Compared with the theoretical calculation of 0.485Hz, the deviation is 1.25%. The damping ratio decreased from 0.301% to 0.217%, the damping ratio decreased by 27.9%, While the exciting force decreased by 45.4%. The exciting force of the blade test system is not simply the output force of the exciter, but the result of the coupling of the exciting force of the equipment and the motion of the blade. The decrease of damping ratio after installation of the aerodynamic fairing can effectively reduce the exciting force at the same bending moment.

5. Conclusion

The vibration bending moments of the blade after the installation of the aerodynamic fairing can envelop the target bending moments, which are in the range of 100% to 115% compared with the target, and meet the requirements of the test specification.

The aerodynamic fairing does not change the structure damping of the blade, which reduces the exciting force by decreasing the vibration of the aerodynamic resistance. The damping ratio of the blade after installation of the aerodynamic fairing is reduced by 27.9%. The exciting force can be reduced by 43.3% when the target bending moments are reached. It is an effective way to reduce the exciting force.

The exciting force of the equipment and damping ratio of test system after the installation of the aerodynamic fairing are basically the same as the theoretical calculation.

Acknowledgments

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