The Study of Wind Turbine Blade Fatigue Test

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Abstract. The fatigue test of wind turbine blade is an important means to verify the fatigue life of wind turbine blade. This paper analyses the problems existing in the fatigue test of wind turbine blade, focuses on the excitation mode, the relationship between excitation amplitude and vibration frequency and the vibration principle, and puts forward feasible solutions in practical operation.

1 Introduction

The design life of wind turbine blade is generally 20 years. How to verify its fatigue life in the blade design stage has been the focus and difficulty of blade design. It is well known that fatigue damages are one of the main causes of blade damage, which seriously affects the service life and safe operation of wind turbine blades. Since the theoretical method of blade fatigue life analysis is not yet mature, the results of theoretical calculation alone are often quite different from the actual results. Therefore, the fatigue test of wind turbine blade is a key link to test its life and quality. At the same time, the measured data are compared with the design data, which can effectively to verify the reliability of the theoretical analysis[1].

This paper emphasis on the theoretical analysis of the vibration principle of wind turbine blade fatigue tests, providing theoretical basis and practical guidance for testers engaged in blade fatigue tests[2].

2 Mode of excitation

2.1 Single point excitation

Single point excitation refers to the excitation mode with only one loading point. The excitation system is installed on a certain section of the blade. The centrifugal force generated by the rotation of the eccentric wheel of the excitation system imposes alternating constant amplitude loads onto the blade. By adjusting the rotational speed of the eccentric wheel and the mass of the eccentric wheel, the rotational frequency is close to the natural frequency of the blade, and then the resonance is generated to achieve the excitation effect. The single point excitation is shown in Fig.1. Single-point excitation method has low requirements on test equipment and test cost, which is the main method for fatigue test for wind turbine blades in China[3].

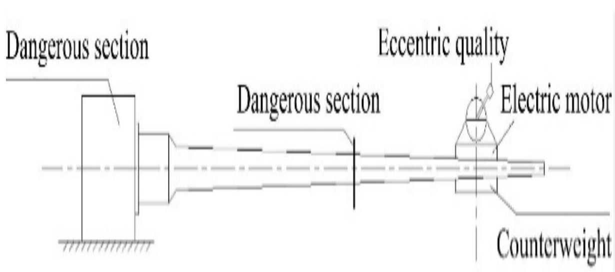

Figure 1. Diagram of single point excitation

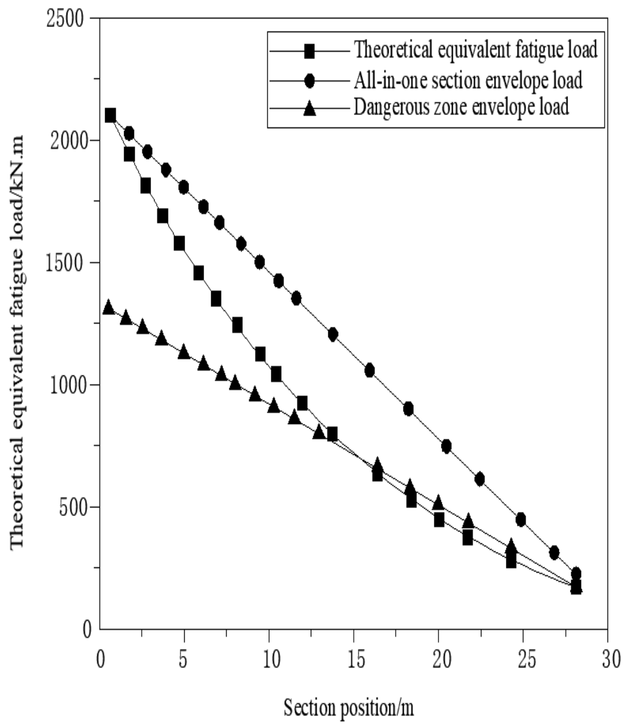

Figure 2. Load curves of each section under single point excitation

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Since there is only one loading point, the test load of each section is approximately linear with the section position, and the equivalent fatigue load of each section of the blade is concave curve with the section position. It can be seen from Fig. 2. that the difference between the test load of each section and the equivalent fatigue load is large. If the test load is enveloped from the root of the blade, the test load of the other sections will significantly exceed the equivalent fatigue load, and there is a risk that the service life of the blade cannot be effectively verified. Therefore, single-point excitation is generally used to conduct targeted tests on dangerous areas with small fatigue safety factors, or to verify the fatigue life of the blade in sections.

2.2 Multi-point excitation

Multi-point excitation refers to the test method of loading and exciting vibration through multiple loading points, mainly through hydraulic lifting equipment, and simultaneous loading and unloading of multiple sections of the blade at the same time to achieve the excitation effect. The schematic diagram is shown in Figure. 3. The multi-point excitation method requires high test equipment and test cost. At present, only a few foreign laboratories can carry out multi-point loading.

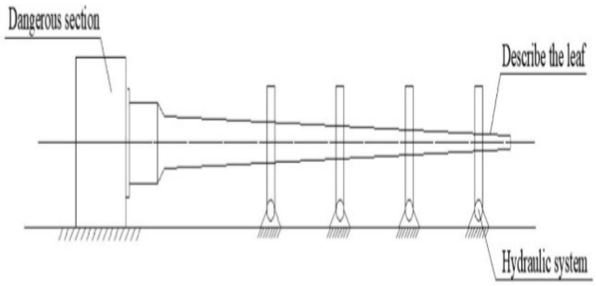

Figure 3. Schematic diagram of multi-point excitation

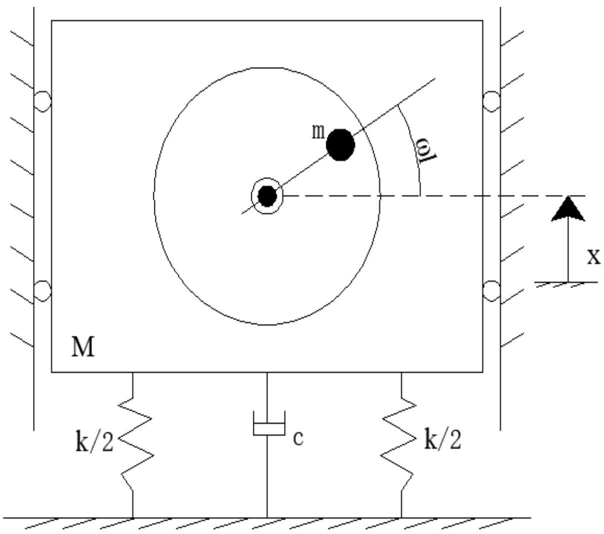

Figure 4. Mechanical model diagram of blade vibration

Because there are multiple loading points working together, and the position, load and vibration frequency of each loading point can be accurately adjusted, the test load of each section of the blade can be approximated to the equivalent fatigue load. Therefore, the multi-point excitation method is generally used for the equivalent fatigue load envelope test of the full section of the blade.

Multi-point excitation requires high accuracy on the test site and equipment, and it takes a long time to debug before the test. According to statistics, multi-point excitation tests requires at least one year for foreign experienced testing institutions, while single-point excitation only needs 1 year. The test can be completed within half a year from the preparation time of 2 months. This article analyses and studies the single-point excitation method commonly used in China.

3 Relationship between amplitude and number of vibrations

According to the principle of equal damage, using the S / N curves formula in GL2010:

$$N=\left[\frac{R_{k,t}+|R_{k,c}|-|2\gamma_{Ma}S_{k,M}-R_{k,t}+|R_{k,c}||}{2(\gamma_{Mb}/C_{1b})S_{k,A}}\right]^{10}$$

$$D=\sum\frac{n}{N}$$

After the transformation:

$$S_{KA}=\left[\frac{R_{k,t}+|R_{k,c}|-2\gamma_{Ma}S_{k,M}-R_{k,t}+|R_{k,c}||}{[2(\gamma_{Mb})]N^{\frac{1}{10}}}\right]$$

Where "n" is the number of test vibrations; "N" is the number of vibrations allowed; "D" is fatigue damage; S_{KA} is the amplitude stress of dangerous section; $S_{k,M}$ is the mean stress of dangerous section; $R_{k,t}$ is the material partial coefficients. So S_{KA}、$S_{k,A}$、$S_{k,M}$ $R_{k,t}$、$R_{k,c}$ are constants.

The relation between amplitude and vibration times is obtained by conversion as follows.

$$S_{KA}{}^{10}=\frac{c}{n} \tag{1}$$

It can be seen from the above equation that if the test amplitude is small, the vibration number increases exponentially. If the test amplitude cannot meet the requirements, the test time will be increased, which seriously affects the test progress and greatly increases the test cost. Therefore, how to effectively control the amplitude to reach the design value is the premise of whether the fatigue test can proceed smoothly.

4 Principle of amplitude regulation

According to the amplitude control principle of vibration mechanics analysis, the mechanical model of blade vibration is established by assuming that the fixed mass of the blade and the surge system is "M", the mass of the eccentric wheel is "m", the eccentricity is "e", the rotational angular velocity of the rotor is "ω", and the blade is regarded as two parallel springs with stiffness

"$k/2$". The damping in the vibration process is simplified as the effect of the damper with damping coefficient "c" in parallel with the spring on the system. As shown in Fig.5.

According to the literature.

Frequency-ratio is.

$$\lambda = \frac{\omega}{\omega_n} \quad (2)$$

The natural frequency of the system without damping are.

$$\omega_n = \sqrt{\frac{k}{M}} \quad (3)$$

The relative damping coefficient is.

$$\xi = \frac{c}{2\omega_n M} \quad (4)$$

Phase difference is.

$$\phi = tan^{-1}\frac{2\xi\lambda}{1-\lambda^2} \quad (5)$$

The amplitude of steady-state vibration is. $B = \frac{me}{M} \cdot \frac{\lambda^2}{\sqrt{(1-\lambda^2)^2+(2\xi\lambda)^2}}$ (6)

The curve families of phase difference "ϕ" and frequency "$\frac{MB}{me}$" ratio under different damping coefficients "ξ" drawn by Formulas are called phase frequency response curve and amplitude frequency response curve.

According to the amplitude-frequency response curve, when $\lambda = 1$ reaches resonance, the maximum amplitude are.

$$B = \frac{me}{2\xi M} = \frac{me\sqrt{k}}{c\sqrt{M}} \quad (7)$$

According to the phase-frequency response curve, the phase difference at resonance is 90° . Relationship between Blade Position and Rotary Arm Position is shown in Fig. 5.

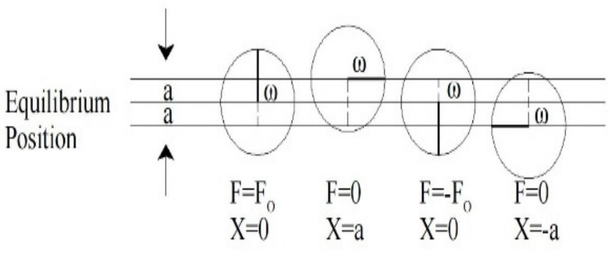

Figure 5. Relationship between Blade Position and Rotary Arm Position

Based on the above analysis, conclusions can be drawn.

When the excitation point position is constant, namely, "k" is constant, the maximum amplitude of the blade is proportional to the mass "m" of the eccentric block, proportional to the cantilever length of the eccentric wheel "e", inversely proportional to the mass square $\sqrt{M}$ of blade and excitation system, and inversely proportional to the damping coefficient "c".

When the frequency of the motor is close to the natural frequency of the blade and the excitation system, the amplitude changes sharply.

When the blade reaches the maximum amplitude, the arm position is horizontal, and the blade reaches resonance.

5 Amplitude controls method

According to the above analysis and combined with the engineering practice, the following measures are taken. Adjust to increase the amplitude:

Adopt high-power motor, increase the mass m of eccentric block.

Under the premise of constant power, light motor is used to reduce the fixed mass M.

Saw off the tip part, reduce the fixed mass Y, and reduce the damping C.

High precision frequency converter is used to get closer to the resonant frequency.

The practice shows that the above methods can effectively adjust the blade amplitude and meet the requirement of theoretical amplitude.

6 Conclusion

Multi-point excitation is mainly used for the test of the full section envelope theory load, while single-point excitation is mainly used for the targeted test of the dangerous area, and has low requirements on the test equipment and cost. Therefore, it is the main method for the fatigue test of wind turbine blades in China[4].

The number of vibration is inversely proportional to the amplitude of 10 times. The premise of the fatigue test is to effectively adjust the amplitude adjustment.

According to the analysis results of vibration mechanics, the maximum vibration amplitude of the single-point excitation system on the blade is $\boldsymbol{B = \frac{me}{2\xi M} = \frac{me\sqrt{k}}{c\sqrt{M}}}$. The fixed mass M and damping C can be adjusted by changing the motor power, counterweight mass, sawing off the blade tip, etc. At the same time, the high precision frequency converter is used to effectively control the amplitude, reaching the ideal level of fatigue tests[5].

Acknowledgement

This work was supported by the research of Wuhan Polytechnic University student scientific research project 2021(Funds, xsky2021138) and the research project of Wuhan Polytechnic University (NO 2021Y44)

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