

Horizontal dual-point excitation and fatigue test of full-scale wind turbine blade

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Abstract. In order to meet the requirements of large-blade fatigue testing, a horizontal dual-point linear exciter loading program is developed. In this paper, Lagrange's equation and FEM method are used to simulate the dual-point excitation theory model. The horizontal dual-point excitation characteristics of the blade are validated by the experimental method. The relationship among the dual-point exciting force, cycle time and bending moment during blade fatigue test in flap wise direction is studied. The advantages of horizontal dual-point excitation in blade fatigue test are analysed. Compared with vertical single-point excitation when the equal bending moment is reached, the exciting force and energy consumption required for dual-point excitation are reduced by 60.0% and 96.8%. Resonant frequency of the test system is improved by reducing the dead weight because of dual-point horizontal excitation. The entire blade fatigue test time is shortened by 7.4%. The use of horizontal dual-point excitation could save energy and shorten the entire fatigue test time.

1. Introduction

The reliability of wind turbine blades should be validated through fatigue test^[1-4]. The single-point fatigue test of the large-blade requires high power for the exciter, moreover, bending moment on the blade applied by the single-point exciter could not fit well with target value in the blade long direction^[5-6]. The dead weight is used to adjust the local bending moment on the blade during fatigue test, which will reduce the blade natural frequency and the resonant excitation force that lead to the entire fatigue tests need more energy consumption and test time. In order to meet the requirements of large-blade fatigue test, a horizontal dual-point linear exciter loading program is developed^[7-8]. By changing the position and force of the two exciter, the bending moment of the blade is adjusted to fit the target value. The test equipment fatigue loss and spare parts replacement decrease because of the excitation force of each exciter is reduced by replacing the single exciter cycle loading on the blade with two exciters. Xin Lai's^[9-10] study showed that there is problem of electromechanical coupling that influencing the control of vibration hammers. In this experiment, the resonant loading was carried out by two linear motor exciters. Compared with the vibration hammer, the linear motor has advantage of the orientation of loading force is in one direction, which eliminates the influence of additional torsional force on the coupling between the two linear motor exciters. The loading cycle time precision



of the exciter has a great influence on the blade vibration bending moment. The cycle time deviation of linear motor loading system could control in less than 1ms. At present, the excitation force cycle loading direction during the blade fatigue test in the flap wise direction is perpendicular to the ground^[11-14]. The linear motor's axis force should reduce the gravity of dynamic mass, spring force, friction and others, of which the rest is the exciting force. The gravity of dynamic mass is huge in flap wise direction. If the blade is turn to 90° and exciter is horizontal, the linear exciter's axis force input is direct to exciting force. The efficiency of linear exciter will increase so much. From the current domestic and foreign documents, the way that full-scale blade cycle loading is applied by horizontal dual-point exciters during fatigue test in flap wise direction has not studied yet.

In this paper, the numerical model of fatigue loading system is established on the basis of the horizontal dual-point excitation test method during blade fatigue test in flap wise direction. The vibration characteristics of the system are simulated by finite element method. Horizontal dual-point excitation test is carried out on 40.3m blade mounted on the test bench. The results of the finite element simulation model are validated through the test, which provides a theoretical basis for precise control of dual-point fatigue loading of large-blades and optimal design of blades.

2. Theory modal of horizontal dual-point excitation

2.1. Numerical modal

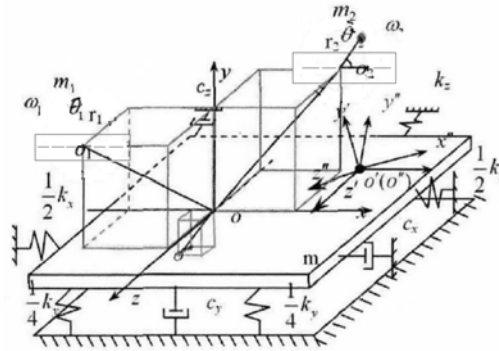


Fig 1. Dynamic model of dual-point horizontal loading system of wind turbine blade fatigue test

$$\begin{aligned}
 & (m + \sum_{i=1}^2 m_i) \ddot{x} + k_x x + c_x \dot{x} = 0 \\
 & (m + \sum_{i=1}^2 m_i) \ddot{y} + \sum_{i=1}^2 m_i \dot{\theta}_i r_i \cos \theta_i + k_y y + c_y \dot{y} = 0 \\
 & (m + \sum_{i=1}^2 m_i) \ddot{z} + k_z z + c_z \dot{z} = 0 \\
 & (ml_{01}^2 + \sum_{i=1}^2 m_i l_{i1}^2) \ddot{\varphi}_1 + \\
 & (ml_{01} l_{02} \sin \beta_{01} \sin \beta_{02} - \sum_{i=1}^2 m_i l_{i1} l_{i2} \cos \beta_{i1} \cos \beta_{i2}) \ddot{\varphi}_2 + k_{\varphi_1} \varphi_1 + c_{\varphi_1} \dot{\varphi}_1 \\
 & = \sum_{i=1}^2 m_i l_{i1} \cos \beta_{i1} (\ddot{\theta}_i r_i \cos \theta_i - \dot{\theta}_i^2 r_i \sin \theta_i) \\
 & (ml_{02}^2 + \sum_{i=1}^2 m_i l_{i2}^2) \ddot{\varphi}_2 + \\
 & (ml_{01} l_{02} \sin \beta_{01} \sin \beta_{02} - \sum_{i=1}^2 m_i l_{i1} l_{i2} \cos \beta_{i1} \cos \beta_{i2}) \ddot{\varphi}_1 + k_{\varphi_2} \varphi_2 + c_{\varphi_2} \dot{\varphi}_2 \\
 & = \sum_{i=1}^2 m_i l_{i2} \cos \beta_{i2} (\ddot{\theta}_i r_i \cos \theta_i - \dot{\theta}_i^2 r_i \sin \theta_i)
 \end{aligned} \tag{1}$$

The dynamic model of the dual-point fatigue loading test system is shown in Fig.1. In order to simplify the system, here it is assumed that the damping force and elastic force of the blade in each direction are the linear functions of velocity and stroke in the direction. As fig.1 shows, $oxyz$ is absolute coordinate system; $o'x'y'z'$ and $o''x''y''z''$ are dynamic coordinate systems; O is the center of mass of entire system (include blade and two exciters); O_1 and O_2 are the center point position of the two masses respectively. m is total mass include blade and two exciters; m_i is the mass of exciter i ; r_i is mass stroke of exciter i ; ω_i is the frequency of exciter i , θ_i is the phase of exciter i ($i = 1, 2$).

The dynamic equation of the vibration system is established according to the Varangian equation, $x, y, z, \phi_i, \theta_i$ ($i = 1, 2$) as generalized coordinates. In equation: ϕ_i is the angle of $oxyz$ and two dynamic coordinate system respectively ($i = 1, 2$); l_{i1} is oo_i projection in the yo_z coordinate plane; l_{i2} is oo_i projection in the xoy coordinate plane; l_{01} and l_{02} is oo'' projection in the yo_z and xoy coordinate plane respectively; β_{i1} is the angle of l_{i1} and z -axis positive direction; β_{i2} is the angle of l_{i2} and x -axis positive direction; β_{0i} is the angle of l_{0i} and y -axis negative direction; $k_x, k_y, k_z, k_{\phi_i}$ and $c_x, c_y, c_z, c_{\phi_i}$ are the blade stiffness and damping coefficient in the direction of x, y, z, ϕ_i , respectively.

2.2. Results of FEM analysis

The stiffness and mass of the blade are assigned to the beam element properties in finite element model, and excitation forces of 1250N and 350N are applied at 16 m and 24 m away from blade root, respectively. The resonant frequency of the blade is 0.610Hz, and the amplitude of the 24m position is 1000mm. It can be seen from the phase-frequency diagram that the phases of difference positions on the blade are substantially coincident and the phase transitions from 0° to 180° at the resonant frequency.

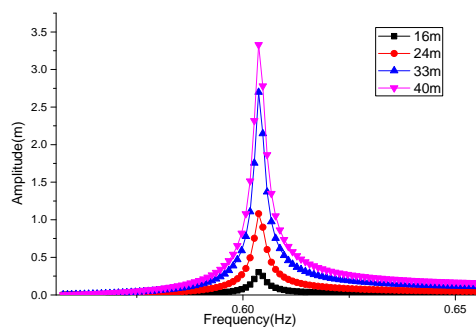


Fig2. Blade Amplitude-frequency curves of dual-point horizontal loading system

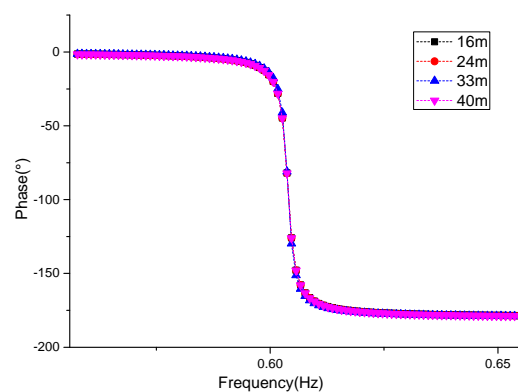


Fig3. Blade phase-frequency curves of dual-point horizontal loading system

3. Experiment results and discussion

The 40m blade with the leading edge facing up and the trailing edge facing down was mounted on the test bench during the fatigue test in the flapwise direction. The two exciters applied a horizontal excitation force on the blades by reciprocating the dynamic mass. Exciter A and B were installed at 16m and 24m away from the root. The clamp weights were 800Kg, 900Kg, 1300Kg and 300Kg at the distance of 12m, 16m, 24m and 32m from blade root, respectively. The exciting force was changed by adjusting the number of dynamic mass and stroke. The stroke ratio of the two exciters was obtained by

separately exciting the blades to the target bending moment with the exciters A and B respectively, and then set the two linear motor exciters in accordance with the ratio of the stroke exciting the blade to the target bending moment.



Fig.4(a) Horizontal dual-point exciter mounted on the blade



Fig.4(b) Vertical single-point exciter mounted on the blade

3.1. Comparison of bending moment

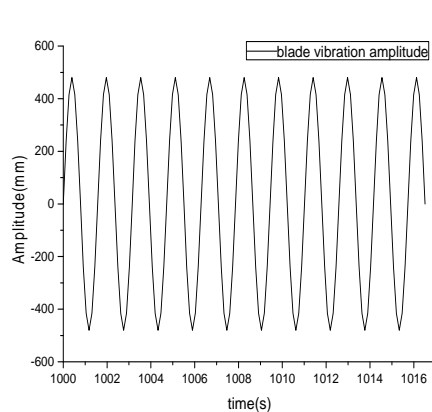


Fig.5 Amplitude-time curves in the blade 24m position

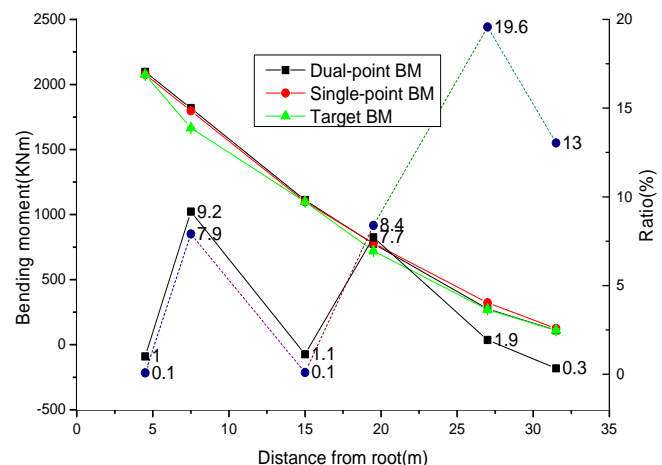


Fig.6 Comparison of single-point, dual-point exciting and target bending moment

The vibration amplitude of the blade at 24m is 960mm when blade is exciting to target bending moment. Compared with the theoretical value of 1000mm, the deviation is 40mm. The stability of the amplitude data among cycles indicates that the two exciters can co-load well. The ratio of strain to bending moment per cross section is obtained by static calibration of the blade, through which the peak-to-peak value obtained by the vibration of the blade is converted into the bending moment. The difference of bending moments among single - point excitation, two - point excitation and target bending moment is compared.

As we can see from Fig.6, the target bending moment can be included both by single-point and dual-point excitation curves. The round dots on the graph represent the difference between the single point and target bending moment, and the square dots represent the difference between the dual-point and target bending moment. The fitting deviation of single-point excitation and target bending moment reach 19.6%, while the dual-point excitation can be controlled within 9.2%. It shows that the bending moment distribution of the dual-point excitation on the blade can fit the target bending moment better.

3.2. Exciting force and energy consumption comparison

When the same bending moment on the blade is reached, the maximum exciting force of dual-point horizontal excitation and single-point vertical excitation are 1250N and 3112N respectively, the excitation force of dual-point is 60.0% less than that of single-point. Reduced excitation force is conducive to improving the life of test equipment and reducing the replacement of spare parts. Fatigue test for large-blade can be done only by adding exciter modules.

The energy consumption of one cycle of the horizontal exciter can be divided into four 1/4 cycles to calculate. The energy in the horizontal direction is the loading force and displacement integral. The energy consumption of the vertical exciter is a displacement integral that overcomes the mass gravity and exciting force. When the exciter is moving upward, force is the gravity of the mass plus the exciting force. When the exciter is moving downward, the force is the gravity of the mass minus the exciting force. In one cycle, the exciting force acts as a neutralizing force in the exciter energy consumption. The energy consumption for the exciter is equal to the integration of the mass gravity and displacement.

$$F = 8\pi sm\omega^2 \quad (2)$$

$$w_h = 4 \int_0^s F ds = 4 \int_0^s 8\pi sm\omega^2 ds = 16\pi m\omega^2 s^2 \quad (3)$$

$$w_v = \int_{-s}^s (F + G) ds + \int_s^{-s} -(G - F) ds = 4GS \quad (4)$$

F is linear motor exciting force; s is the dynamic mass stroke; m is mass; ω is frequency; G is the gravity of mass.

When the blade fatigue test is changed to horizontal dual-point excitation, the energy consumption is only 3.22% of the vertical single-point excitation. The linear motor needn't to work to overcome the mass of gravity in the horizontal direction. So the axial force of linear motor output is almost direct to the excitation force of the blade.

Table1. Comparison between single and dual points exciting

Item No.	Exciting direction	Mass Stroke (mm)	Dynamic mass (kg)	Frequency (Hz)	Exciting forces (N)	Single cycle work (J)	2E6cycles electricity consumption (KWh)
1	Vertical single	210	1860	0.563	3112	15312	8506
2	Horizontal A	168	780	0.616	1250	420	233
	Horizontal A	53	780	0.616	394	42	23

3.3. Resonant frequency and test cycle time

As the use of dual-point excitation, bending moment distribution on the blade can be adjusted through changing two exciting force and position, so you can reduce the use of deadweight. Blade fatigue test includes flapwise and edgewise directions, which need at least 2 million cycle times in each direction. So the resonant frequency directly affects the entire test time. Fatigue test time become a critical bottleneck in test efficiency due to the fact that the blade need to be mounted on the test bench during fatigue testing. The frequency of the system include blade and equipment is 0.616 Hz. Compared with the theoretical value of 0.610Hz, the deviation is 0.006Hz. The dual-point test saves 3 days in one direction compared with the single-point test, which reduces the entire fatigue test cycle time by 7.4%.

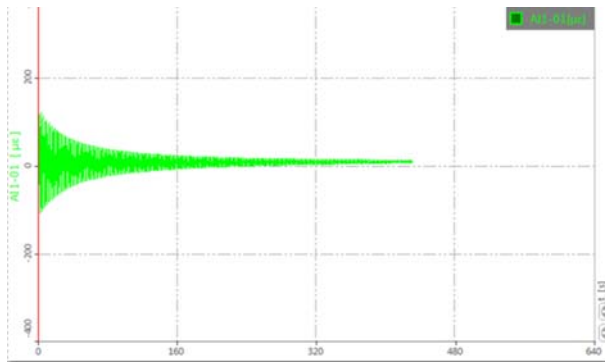


Fig.7(a) Amplitude-time diagram of dual-point exciting

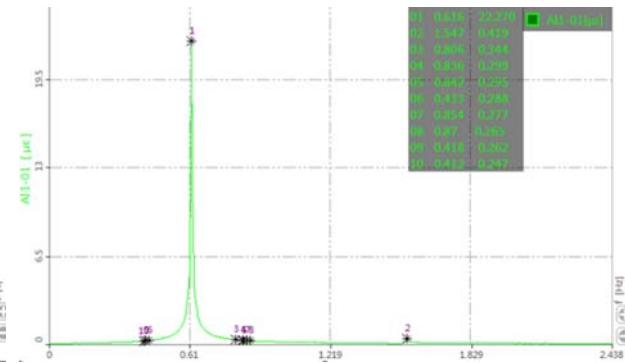


Fig.7(b) First Flap frequency of dual-point exciting

Table2. Cycle time of fatigue test

Item No.	Case	Frequency(Hz)	Cycle Time (s)	2E6cycles test time(day)
1	blade	0.853	1.172	27
2	Single-point	0.563	1.776	41
3	Dual-point	0.616	1.623	38

4 Conclusion

The horizontal dual-point excitation is used to carry out the continuous cyclic loading on the blades. The exciting forces is saved by 60.0% and the energy consumption is reduced by 96.8% during the whole fatigue test period.

The whole blade fatigue test cycle is shortened by 7.4% because of the resonant frequency increases due to the dead weight decrease during the dual-point exciting loading system.

By comparing that the frequency and 24m position amplitude deviation is 0.006Hz and 0.04m respectively, the deviation of experimental results and theoretical calculation is in line with regulation requirement.

Acknowledgments

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