Modal Characteristics of Novel Wind Turbine Rotors with Hinged Structures

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Abstract. The vibration problems of the wind turbine rotors have drawn public attention as the size of wind turbine has increased incredibly. Although various factors may cause the vibration problems, the flexibility is a big threat among them. Therefore, ensuring the high stiffness of the rotors by adopting novel techniques becomes a necessity. The study was a further investigation of several novel designs regarding the dynamic behaviour and the influencing mechanism. The modal testing experiments were conducted on a traditional blade and an isolated blade with the hinged rods mounted close to the root. The results showed that the rod increased both the modal frequency and the damping of the blade. More studies were done on the rods’ impact on the wind turbine rotor with a numerical model, where dimensionless parameters were defined to describe the configuration of the interveined and the bisymmetrical rods. Their influences on the modal frequencies of the rotor were analyzed and discussed.

1. Introduction

The flexibility of large wind turbine rotors may lead to the vibration instability, such as the flutter problem, which is caused by the coupled vibration of flap and torsion [1, 2]. While the lightweight technique and advanced manufacturing process brought in large amounts of benefits, they resulted in a decreased inherent torsion frequency of the wind turbine rotor [2]. The decreased frequency is usually an indirect factor for the coupled vibration in the different directions. During the design of the rotors, the stiffness should be guaranteed to avoid the occurrence of the flutter vibration. In addition, the edgewise vibration instability is an urgent problem when the size of the rotors grows significantly, since the edgewise damping is usually smaller than that in the flap direction [3]. Therefore, the structural design of the wind turbine rotor should give consideration to the load bearing capability, weight, stiffness, damping, etc. at the same time. The stiffness of the rotor is the focus in this paper.

In the traditional design, the blade comprises an aerodynamic shell and the main spar. This spar usually has a box or “D” shape made of composite plies and a sandwich material to ensure bending stiffness. [4] Other designs adopted a new airfoil to ensure the aerodynamic behavior and the stiffness requirement, such as the flat-back foil by cutting part of the trailing edge, blunt trailing edge by continuously increasing the thickness of shell. [5] In addition, Roth-Johnson et al. proposed a biplane root-type blade and numerically studied the mechanical behavior of this structure.[6, 7] They showed that a biplane structure has a smaller tip displacement, higher lift–drag ratio. [6, 7]
The modal frequency reflects the dynamic stiffness of the structure. Frequency maximization [8] incorporates the optimized constraints of the segmented spar length to satisfy both the weight and stiffness requirements. [9] Tarfaoui et al. used the finite element method to study the influence of the spar configuration on the modal frequency. [10] The finite element method is effective for examining both the static deformation and vibration behavior of a structure and verifying a blade design. Modal testing is an experimental technique for obtaining the natural frequency of a structure. [11]

In the previous study, several designs with the different configurations of the hinged structure and rods were analyzed mainly regarding their load bearing capability and modal frequencies. Two schemes performed well in terms of the reduced moment and the increased frequencies. [12] The configuration of these two schemes was shown in Figure 1. [12] However, the results were not validated by the experiments and the rods’ influencing mechanism on the modal frequencies was not clearly demonstrated. Herein, the study will focus on the two kinds of rotors’ dynamic behavior and the influencing factors. Based on both the experiments and numerical study, the paper will show the discipline of the modal frequency and the damping with the change of the configuration of the rods.

![Figure 1. Illustrations of two novel wind turbine rotors, scheme 3 and scheme 5. [12]](image)

2. Modal testing experiments
Modal testing experiments were completed on a platform developed in-house (Figure 2) to examine the rod’s influence on the stiffness of the two blades [13]. The length of the tested blade was 1.35 m. The blade was made of fiberglass reinforced plastics and weighted 2.8 kg. The chord distribution of the blade was shown in Figure 2(c). To obtain clear and reliable signals, two INV 9823 acceleration sensors were mounted on gauge points close to the blade tip to record the blade response. The sensors and hammers were directly connected with an eight-channel acquisition card. Thirteen gauge points were stroked by the hammer in the chord-wise direction in order, and the excitation force and free vibration response were recorded. Data acquisition and signal processing (DASP) method was adopted to extract the modal frequency, damping ratio and mode shape. Each rod, made of aluminium alloy, was composed of two parts. The two separated parts were connected by threads. In the tested model, the rods connected with blade at No.6 gauged point, where is 47 % span along the blade length.

The modal frequency, damping ratio and mode shape of the cantilevered blade with and without rods were obtained in modal testing (Figure 3). Because the two acceleration sensors were mounted on the blade perpendicularly, vibration information in both the flap and edgewise directions were acquired (f: modal frequency in these two directions, Nm: mode number).

Because of the low stiffness in the flap direction compared with the edgewise direction, the lower modes were mostly flap modes; the third mode exhibited edgewise vibration. After the hinged rods were mounted in the rotating plane of the blade (Figure 2), the edgewise frequency rose significantly (Figure 3(a)). The hinged rod improves the in-plane stiffness of the blade. The frequency of the flap modes did not increase after the hinged rods were introduced. Each group of modal testing
experiments was conducted three times, and the results seemed similar. The reduced frequency of the higher modes may be related with the slight change in stiffness due to the in-plane rods.

Figure 2. Modal testing experiments: (a) cantilevered blade; (b) cantilevered blade with hinged rods; (c) chord distribution of the tested blade.

Figure 3(b) illustrates the mode shapes and corresponding modal damping of the two blades in Figure 2. The upper part of Figure 3(b) shows the previous six modes for the blade with the hinged rods, while the lower part of Figure 3(b) shows the results of the cantilevered blade. The two blades had the same mode shapes for the previous four modes. However, because the rod added a DOF to the system, the blade with the in-plane rods had one more edgewise mode. Furthermore, introducing the hinged rods reduced the frequency of the higher modes: the flap and torsion coupling modes (sixth mode in upper Figure 3(b) and fifth mode in lower Figure 3(b). The hinged rods elevated the system’s damping ratio, especially for the edgewise mode. This improved the reliability of the blade because a large structural damping ratio helps with the vibration attenuation and reduces the blade instability. The experiments demonstrated that the hinged structure and the rods effectively improve the blade’s mechanical performance. Further study was needed to reveal the mechanism of the rods’ influences on the modal frequencies of the rotor.

3. Factors influencing the rotor stiffness.

Different rod configurations enhanced the stiffness in schemes 3 and 5 of Figure 1. [12] To study the underlying principles, two dimensionless parameters were defined. For scheme 3, the normalized distances between the two rod locations and the blade root over the blade length were defined as \(x_1\) and \(x_2\) (Figure 4(a)). The three values of \(x_1\) were kept constant, and \(x_2\) was moved along the blade to obtain its influence (Figure 4(b) and (c)). The collective mode frequency \(f_{c1}\) and whirling mode frequencies \(f_{c2,3}\) were normalized by the first edgewise mode frequency \(f_{e0}\).

The intervened rods clearly had different effects on the collective mode and whirling modes. Looking at the curves’ valleys in Figure 4(b), when \(x_2 = x_1\), a local minimum of \(f_{c1}\) was obtained. When \(x_1 = 0.1659\), the rotor stiffness could not be improved when \(x_2 > 0.7187\). When \(x_1 = 0.3285\), the rotor stiffness could not be improved when \(x_2 > 0.6537\). The optimal location of \(x_2\) to reach the maximum frequency was about 0.6 ~ 0.7 of the distance to the root. When \(x_1 = 0.4586\), the optimal location of \(f_{c2}\) was about 0.1667. This could be derived from the curve’s peaks. When \(x_1 = 0.4586\), mounting the other rod end at locations away from \(x_2 > 0.3285\) could not improve \(f_{c1}\). These characteristics are distinct from the other two cases. When \(x_1\) was close to the root, the curves had two peak values (Figure 4(b)).

Figure 4(c) shows the influence of the intervened rods on the second and third edgewise modal frequencies. The three curves with different values of \(x_1\) had similar trends. When the rod location \(x_2\)
was moved forward to the blade tip, $f_{2,3}$ increased significantly and then decreased. The frequency was improved the most when $x_2 = 0.6–0.7$. When $x_1= 0.4586$, $f_{2,3}$ was markedly improved. Thus, the whirling modes were more affected by the interveined rods than the collective mode.

![Graph showing modal frequencies and damping ratios](image)

**Figure 3.** Comparison between the cantilevered blade with and without hinged rods: (a) modal frequencies; (b) damping ratios and modes.

In scheme 5, the rod configuration affected the frequency change in a different pattern (Figure 5). Two parameters were defined to describe the locations of the supporting rods and hinged rods: $x$ denotes the normalized rod-mounting location on the blade, while $R$ denotes the distance of the joint of the supporting rod and the tension rod away from the hub center. $R$ was divided by the hub radius $R_0$ to get the dimensionless parameter $R/R_0$.

Figure 5(b) shows the influence of the rod-mounting location on the frequencies of the first edgewise mode group. The whirling mode frequencies were clearly improved more than those of the
collective mode. A different influence principle was found. When \( R/R_0 \) was set to a small value (e.g., 2.5), \( f_{e1} \) slightly increased. However, it decreased as the joint location \( R/R_0 \) moved away from the hub. When \( x = 0.1659 \) and 0.3285, once \( R/R_0 > 2 \), \( f_{e1} \) could be improved. However, for the whirling modes, when the joint location moved in the range of [2, 5], the frequencies increased quite quickly until they reached a steady value.

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\text{Figure 4. Influence of the locations of the interveined rods on the modal frequency of the rotor: (a) defined parameters; (b) frequency of the collective mode; (c) frequency of the whirling modes.}
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\text{Figure 5(c) summarizes the pattern of how the tension rods’ location affecting the frequencies. Both of the frequencies of the collective mode and whirling modes first increased and then decreased. This impact was more obvious for the whirling modes than the collective mode. } f_{e1} \text{ was most improved when the tension rod was located at } x \sim 0.6 \text{ from the blade root. } f_{e2,3} \text{ were most improved when the tension rod was located at } x \sim 0.7 \text{ from the blade root. As shown in Figure 5 (c), when the tension rod was located too far from the blade root (e.g., } R/R_0 \sim 10.25\text{), } f_{e1} \text{ showed almost no improvement.}
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4. Conclusions

In this study, the dynamic behavior of the cantilevered blades with the interveined rods and the single hinged blades with the bisymmetrical rods were investigated. The conclusions were drawn as follows:

1) The modal testing results showed the potential benefits of the hinged rods to enhancing both the blade stiffness and damping of the edgewise modes.

2) The dimensionless parameters \( x_1, x_2, x \) and \( R/R_0 \) were defined to reveal the principles of the edgewise modal frequencies for scheme 3 and 5. The influencing curves of the rods’ location on the frequencies showed large differences with the maximum frequency occurring at the different locations.
Figure 5. Influences of the locations of the bisymmetrical rods on the edgewise frequency of the rotor: (a) defined factors; (b) the hinged locations’ influences; (c) the tension rods’ location’s influences.

Acknowledgments
The study was supported by the National Natural Science Foundation of China (No. 51575296). The authors gratefully acknowledge this support.

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