Design and Analysis of Wind Turbine Rotors Using Hinged Structures and Rods

Hongya Lu\textsuperscript{1}, Pan Zeng\textsuperscript{2,\ast} and Liping Lei\textsuperscript{2}

\textsuperscript{1}Beijing Institute of Mechanical Equipment, No.50 Yongding Road, Haidian District, Beijing 100039, China
\textsuperscript{2}Key Laboratory for Advanced Material Processing Technology of MOE, Department of Mechanical Engineering, Tsinghua University, Beijing 100084, China

\ast zengp@mail.tsinghua.edu.cn

Abstract. Light weight and high stiffness are key design factors in ensuring cost effectiveness and reliability of wind turbines, especially for the inboard region of the rotor blades. In this study, several novel designs were developed to improve the mechanical performance of the rotor. Experiments were performed on an isolated blade incorporating the new features of a hinged structure and rods. The results validated the effectiveness of these features at alleviating the root-bending moment of the blade under varying wind loads and enhancing the stiffness of the blade. A numerical investigation was carried out to further examine the bending moment distribution, shear and axial force, and rod tension of these novel rotor designs under uniform loads. Longitudinal geometrical variations of the blade were considered in the model. Results showed that two designs realized a favorable bending moment distribution and improved the modal frequencies of the edgewise modes: bisymmetrical rods on a single-hinged structure and interveined symmetrical rods on a cantilevered structure. However, these designs have different deformation mechanisms. In addition, the first group of edgewise modal frequencies of these two designs were improved compared with the traditional rotor design. Their potential values in the application to the design of a lightweight, high-stiffness, and reliable wind turbine rotor were discussed.

1. Introduction

The increasing size of wind turbine rotors has posed new demands for wind turbine blade designs \cite{1} in order to ensure both a light weight and high stiffness and a low cost and high reliability. \cite{2} The loads on the blade are the key factor that determine the structural design.\cite{3} In-plane loads including gravity, the centrifugal force and the aerodynamic load for the large blades increase enormously and become not negligible.\cite{1,4}

There have been several studies on realizing lightweight and stiff wind turbine rotors by using advanced materials. For instance, carbon fiber reinforced composite laminates have been used to enhance the stiffness and mass ratios of the blade.\cite{5,6} However, the high cost of carbon fiber material and the difficulty with designing a gradual transition between carbon fiber and ordinary glass fiber have limited the application of this approach. Other designs include adopting a new airfoil to meet the stiffness requirement. The flat-back foil with thick or blunt trailing edge \cite{7,8} have been utilized at the blade root to enhance the cross-sectional area and bending stiffness. SNL 100-03 includes a flat-back trailing-edge foil to realize a lower moment at the blade root and adopts a slender foil combined...
with the optimized structural and material configuration to realize a 16% reduction in weight compared with the SNL 100-02 blade. The present authors [9] aimed to reduce the moment of the blade at the root by introducing a rod design. A combined numerical and experimental investigation into the structure indicated that the weight of an NREL 5 MW blade is reduced by approximately one-third.

A novel design of the blade root with a small bending load and high stiffness is necessary to obtain a lightweight and reliable design of the rotor. The conventional rotor has a cantilevered beam structure, which does not reduce the loads at the root. Gourieres and Shi [10] presented a design that releases a degree of freedom (DOF) of the blade and supporting structure to reduce the load. The present authors previously considered the concept of releasing a DOF[9] to develop a specific rotor design.

This paper presents a more elaborate investigation of several rotor designs that considers both the load-bearing characteristics and stiffness. The design features of a hinged structure and tension rods were considered. Experiments and the finite element method were utilized to reveal the principles behind the mechanical behavior of these designs. The differences between these designs were considered. Two schemes were selected owing to their good performance, and the factors that influenced their enhanced stiffness were compared.

2. Novel wind turbine rotor design schemes

Scheme 1 (Figure 1) shows the structure of a conventional wind turbine rotor: three blades are fixed at the hub to form three cantilevered beams. A hinged structure and tension rods were used to design seven novel structures with the aim of improving the moment distribution on the blade by changing the configuration of the cantilevered beam.

![Figure 1. Illustrations of novel wind turbine rotors featuring a hinged structure and rods.](image-url)
In scheme 2, three rods are mounted symmetrically around the hub between the blades. The three blades are still fixed to the hub, and each rod is located at the same position on the blade. Scheme 3 reorganizes the rods between the blades; the two ends of each rod are located at different positions on the blades (i.e., interveined). In scheme 4, the three blades are connected with the hub by a hinged structure. Because a DOF of the blade in the rotor rotating plane is released, three central symmetrical rods are introduced to support the structure. This structure is actually unstable. When one of the blades deflects slightly, the whole rotor rotates around the hub to arrive at a new balanced state. Scheme 5 upgrades scheme 4 to connect another three rods, called supporting rods, with the tension rods by hinged joints. The present authors previously investigated scheme 5 with regard to the bending moment and tension on the rods under a concentrated force. Scheme 6 adopts a double-hinged joint similar to that applied in aircraft design to connect the blades with the hub. This guarantees the stability of the structure without rods. Schemes 7 and 8 respectively introduce central and interveined symmetrical rods to scheme 6.

The hinged structure and rod arrangement style change the boundary conditions and stiffness distribution of the blade. The natural frequency and load distribution may differ from that of the conventional cantilevered blade structure. In addition, the tradeoff between a light weight and high stiffness may unbalance a design and make it unable to satisfy both requirements. Therefore, the influence of the hinged structure and rods on the mechanical behavior was investigated. All of the above designs were examined in terms of the rotating plane; thus, out-of-plane loads were not considered.

### 3. Experimental tests

First, the load-bearing characteristics with the hinged structure and rods were investigated. The method, used by H. Lu, was utilized to record the force and moment at the blade root. Two kinds of structures were tested under wind velocities of 1–12 m/s: a hinged isolated blade with supporting rods and a conventional cantilevered blade, as shown in Figure 2. The truss root of the novel blade (left one) made it convenient to mount the rods. The rods were connected with the blade at the location of 15% span along the blade length. To prevent the vibrations of the wind tunnel platform affecting the results, the tested structures and sensors were all placed on a plate directly connected with the ground and separate from the bottom of the wind tunnel. When the chord of the blade tip was parallel to the wind direction, the angle of attack (AOA) was set as 0°. The experiments were conducted at AOAs of 15° and 20° for at least 30 s at each wind velocity.

![Figure 2](image)
The blade with hinged rods had load-bearing characteristics distinct from those of the cantilevered blade. As Figure 3(a) shown, at wind velocities of 1–12 m/s, the bending moment $M_x$ at the blade root stayed approximately 0. The overall lift $F_y$ tended to increase with the wind velocity. The hinged structure clearly reduced $M_x$, even at a wind velocity of 12 m/s. The tension in the rods demonstrated that they helped support the loads acting on the blade (Figure 3(b)). One rod bore the tensile load, while the other rod bore the compressive load; they were distributed almost symmetrically. The experimental results at AOAs of 15° and 20° demonstrated similar performances. The rod and a part of the blade below the location where the hinged rods meet the blade becomes an “amplified” hub, which makes it possible to transfer part of the load on the blade to the rods. In this manner, the bending moment on the inboard blade is reduced, which decreases the material needed for manufacture.

![Figure 3](image)

Figure 3. Experimental results, (a) the cantilevered blade at wind velocities of 1–12 m/s wind velocity: lift and bending moment; (b) the blade with rods at wind velocities of 3–12 m/s.

4. Numerical simulation of wind turbine rotor

A finite element model was established to study the mechanical behavior of rotors incorporating three blades, a hinged structure and tension rods (Figure 1). The bending moments, shear forces, axial forces, tension on rods and edgewise frequency were evaluated. ANSYS 14.5 were used for the models, which had exactly the same blades and rods, identical geometries and material properties under identical uniform loads.

4.1. Finite element model of the wind turbine rotor

Figure 4 shows the coordinate system. The rotor was located on the XOY plane. Uniform loads were distributed on perpendicular to each blade. Because the “box” of the main spar bears most of the loads on the blade, the blade was assumed to be segmented, and each segment had the same cross-section of a rectangular hollow tube. For both simplicity and effectiveness, the beam 188 element was used to describe the blade. The size of the element cross-section was derived by scaling the area and stiffness distribution of a NREL 5 MW blade. [12] Then, all available parameters were normalized by the data of the root cross-section. The obtained normalized edgewise stiffness distribution was compared with that of the NREL blade (Figure 4 (b)). The blade was scaled to a length of 2 m and divided into 17 elements. The nodes were numbered 1–18 from the root to the blade tip parallel to the $Y$-axis. The effective elastic modulus varies along the blade and the density was assumed to be 1800 kg/m$^3$. All of the rods were modeled as link 180 elements made of aluminum 6061. The double-hinged joint comprised an aluminum 6061 herringbone bracket with an angle between the two struts of 120°. When the blade was connected with the hub by the hinged joint, a corresponding rotation DOF was released. Otherwise, the blades were fixed at the root nodes. For all of the central symmetrical rods, the two rod
ends were located at node 6 on the blades, which is 32.85% span along the blade length. For all of the interveined symmetrical rods, the two rod ends were located at nodes 4 and 8 on the blade, which are 16.59% and 45.86% span along the blade length separately. Static structural analysis was then conducted.

![Finite element model of the rotor: (a) beam shape along the longitudinal axis and applied loads; (b) normalized edgewise stiffness distribution along the blade.](image)

**Figure 4.** Finite element model of the rotor: (a) beam shape along the longitudinal axis and applied loads; (b) normalized edgewise stiffness distribution along the blade.

### 4.2. Static load-bearing characteristics

Figure 5(a) and (b) illustrate the in-plane bending moment and shear force when uniform loads act on a rotor with cantilevered blades. The shear force on the blade was linear, while the bending moment was quadratic from the root to tip. In scheme 2, the bending moment and shear force distribution were the same as in scheme 1, and the tension on the rods was zero. This can be explained by taking the blade along the Y-axis as an example. The two rods cause equal forces that are opposite in direction on the blade, so they cancel. In the steady state, the three rods bore no loads. Thus, the configuration of central symmetrical rods could not improve the moment distribution of the blade and had no effect from a load-bearing perspective. Scheme 3 changed the shear force distribution and bending moment distribution compared to that of the cantilevered blade (Figure 5(c) and (d)). The bending moment was decreased where the rod was mounted to the blade root. The bending moment reached a local minimum when the rod was close to the root. The maximum bending moment was still at the root but smaller.

Although the three rods had identical tensions, the ends of each rod were connected at different locations on the blade. This applied an axial force on the blade. The segment closest to the root had the largest axial force. The axial force helped reduce the bending moment below the rod-mounting location. Thus, the interveined symmetrical rods changed the blade deformation type. In this case, the blade tip experienced pure bending deformation, while the bottom part experienced both bending and compression deformations. Therefore, scheme 3 requires ensuring the axial compressive strength of the material. The hinged structure reduced the bending moment at the blade root to zero in the experiment.

Figure 6 in the Appendix shows the performance of scheme 5. The maximum bending moment was at the tension rod location and was the same as that of scheme 1, cantilevered. The tension in the supporting and tension rods acted together to share the load and reduce the bending moment on the inboard part of the blade. The influence of the rod locations on the moment distribution is discussed in detail in our previous report. [9]

Figure 7 in the Appendix illustrates the load distribution in scheme 6. The double-hinged structure changed the continuity of the shear force. The bending moment direction in the hinged structure became opposite that in the blade. Because the blade was identical with the previous cases, the
maximum bending moment was still at the blade root. The bending moment and shear force in the herringbone bracket were symmetrically distributed. Because the bending moment at the joint of the blade and hinged structure is large, this part needs a detailed design to prevent stress concentration.

Figure 5. In-plane loads distribution, (a) scheme 1: bending moment (N-mm); (b) scheme 1: shear force (N); (c) scheme 3: bending moment (N-mm); (d) scheme 3: tension in rods and axial force in blades (N).

Figure 6. In-plane load distribution in scheme 5: (a) bending moment (N-mm); (b) tension in rods (N).

Figure 8 in the Appendix illustrates the influence of scheme 7. The bending moment kept the same distribution as scheme 6, and the rods bore no loads. The double-hinged structure was subjected to
axial forces, where one side was tensile and the other was compressive. Scheme 8 combined the bending moment distribution and tension in rods in schemes 3 and 6 (Figure 9). The joint connecting the blade and double-hinged structure bore the largest bending moment. The local maxima bending moments occurred when the interveined rods were mounted at 16.59% span and 45.86% span along the blade length separately.

**Figure 7.** In-plane load distribution in scheme 6: (a) bending moment (N-mm); (b) shear force (N).

**Figure 8.** In-plane load distribution in scheme 7: (a) bending moment (N-mm); (b) tension in rods (N).

**Figure 9.** In-plane load distribution in scheme 8: (a) bending moment (N-mm); (b) tension in rods (N).
In scheme 8, the interveined symmetrical rods changed the axial force in the double-hinged structure. The load was greater in the brace bearing the compression deformation and was 99.50 N. The tensile load on the other side was 10.29 N. When schemes 3 and 8 were compared, the axial force below 45.86% span along the blade length in the double-hinged structure slightly exceeded that in the cantilevered blade. In scheme 8, the axial forces on the blade were 63.48 and 36.08 N. The tension in the three rods was identical at 37.67 N each.

The maximum bending moment and axial force of the above schemes were extracted (Figure 10). Scheme 5 achieved the largest reduction in bending moment and did not introduce an axial force. In addition, schemes 3 and 8, which both employed interveined symmetrical rods, decreased the maximum bending moment compared to the cantilevered blade because they introduced an axial force that bore part of the load.

**Figure 10.** Maximum bending moment and axial forces of all schemes under a uniform load; N is the scheme number.

### 4.3. Modal analysis of the novel rotors

Modal analysis was performed based on the same structural models given above, and the first group of edgewise modes was extracted. Figure 11(a) shows the first group of edgewise modes in sequence. Figure 11(b) uses the central symmetrical rods in scheme 2 to represent the typical influence of the rods on the mode shapes. In the first group of edgewise modes, the three blades vibrated in sequence under an identical frequency. However, introducing the rods broke the isolation of the three blades. In the first edgewise mode, the three blades had the same deflections in the same direction in a centrosymmetric pattern. This was the collective mode of the rotating rotor. The flexibility of the interveined rods decreased the modal frequency of this mode. The other two mode shapes behaved like the whirling mode of the rotating rotor. When one blade reached the location of the largest deflection, the other blades arrived at different states but had the same frequency.

Other schemes with different rod configurations had similar mode shapes for the edgewise mode groups (Figure 11(b)). The modal frequency of the first edgewise mode group of the cantilevered rotor in scheme 1 is denoted as $f_0$. Modal frequencies of the first edgewise mode group in the other schemes are denoted as $f$. All of the frequencies derived from the numerical simulation were normalized by $f_0$ (Figure 11(c)). Schemes 1 and 6, which had no rods, had an identical frequency in the first edgewise mode group. Once the rods were incorporated into the rotor, the three frequencies were grouped into two. The lower one was the first edgewise mode (i.e., collective mode), while the two identical higher frequencies were the whirling modes. The interveined symmetrical rods in schemes 3 and 8 increased all three frequencies of the first edgewise mode group. The central symmetrical rods only increased the two frequencies of the whirling modes but lowered the frequency of the collective mode. Scheme 5 improved all of the frequencies of the first group of edgewise modes. Therefore, schemes 3, 5 and 7 were effective at enhancing the rotor stiffness and ensuring reliability.
4.4. Potential application on the enhanced performance

In the study, the total weight of the blade was about 8.1 kg. For scheme 3, the weight of the added rods was 1.65 kg, which accounted for 20% of the blade weight. If the total amount of material kept constant, it meant that the weight of each blade was reduced to 80% of the original blade. Implementing this process by reducing the size of the spar, and repeating the numerical simulation, the mechanical performance of scheme 3 was obtained. The maximum moment at the root was 94.2% of the root bending moment on the cantilevered blades in scheme 1. Compared with scheme 1, the moment along the blades was smaller. The tension on the rods were 35.10 N.

Furthermore, the two schemes were compared regarding the natural frequencies. In scheme 3 with the same rotor weight as the scheme 1, the first group of edgewise frequencies were 45.675, 50.537
and 50.537 Hz. While in scheme 1, the first group of edgewise frequencies were identical, which were 42.505, 42.505 and 42.505 Hz. The comparison between these two schemes proved that the edgewise modal frequencies can be improved and the moments can be reduced by adopting the interveined rods in scheme 3.

The similar manipulation was operated on scheme 5 to keep the same rotor weight as scheme 1. The obtained moments were obviously smaller than that of scheme 1. The first group of edgewise frequencies were 42.339, 47.755 and 47.755 Hz. In the first edgewise modes group, the collective modal frequency kept almost the same while the whirling modal frequencies were improved.

According to the above analysis, when scheme 3 and scheme 5 kept the same weight with scheme 1, the blades endured less loads and became stiffer. Thus, utilizing schemes 3 and 5 to design a wind turbine rotor can reduce the material requirements.

5. Conclusions
In this study, a series of novel wind turbine rotors was designed featuring a hinged structure and tension rods. The mechanical behavior of these designs, including the static load-bearing performance and modal frequency, were investigated experimentally and numerically. Several conclusions were drawn.

1) The experiments in the wind tunnel platform demonstrated that the hinged structure can relieve the bending moment at the blade root under different wind velocities.
2) The ANSYS simulation illustrated the load-bearing characteristics of the design schemes, which helped in selecting designs with a relatively small bending moment and large stiffness: bisymmetrical rods on a single hinged structure (scheme 5) and interveined symmetrical rods on a cantilevered structure (scheme 3).

Compared with the conventional cantilevered structure, utilizing schemes 3 and 5 to design a wind turbine rotor can reduce the material requirements while ensuring stiffness and guarantee performance reliability. Since the novel designs were located close to the root of the blades, their influences on the aerodynamics and energy production can be limited with optimized design. More study should be done to make a thorough understanding of the aerodynamic performance of scheme 3 and 5.

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