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Research on the Influence of Blade Tip Trailing-Edge Serrated Structure on Wind Turbine Noise Reduction and Performance

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Abstract. The environmental impact of wind turbine aerodynamic noise has drawn growing attention. This study investigates the effects of serrated structures on blade aerodynamic noise, structural loads and power performance. At the Xu Chang site in China, serrated structures have been installed on the wind turbine prototype, and field measurements of both power performance and noise levels have been collected. This prototype is a horizontal-axis wind turbine, featuring three 76-meter blades and a 100-meter tower. Load and performance simulations were conducted by using Bladed under specific site conditions, including a characteristic turbulence intensity of 0.096 and an average wind speed of 5.78 ms^{-1}. Analysis of the field measurements demonstrated that the serrated design yielded a quantifiable reduction in aerodynamic noise, with a maximum decrease of up to 3.9 dB. Simulation results however showed critical loads on blade root, tower base and hub increase by more than 2% and there was a 1% reduction in power, equivalent to a loss of 25 hours annually. The results indicate that, although serrated structure can effectively mitigate the noise impact on noise-sensitive environment, the application requires careful consideration of the component load capacity and the cost increase with reduced power generation.

1 Introduction

In alignment with the international dual objectives that carbon emissions peak before 2030 and carbon neutrality is achieved by 2060, renewable energy deployment has accelerated markedly. Among these technologies, wind energy has emerged as a leading clean energy source, characterized by increasing turbine sizes, expanding distributed deployments, and the construction of large-scale wind farm bases. As wind farms are increasingly situated near residential areas, noise pollution from turbine rotors and nacelles has become a pressing concern. Mitigating any disturbance to nearby communities present a significant challenge to the industry. To address the environmental impact, new standards for noise emissions have been implemented. Consequently, noise emission levels are now regarded as one of the key performance indicators in wind turbine evaluation. For large-scale wind turbines, noise primarily originates from two sources: mechanical noise and blade aerodynamic noise. Mechanical noise, which is typically produced within the nacelle, can be mitigated through targeted

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design and insulation strategies. In contrast, aerodynamic noise, generated by the interaction between blades and airflows, remains the dominant and less easily controlled source of noise.

With the development of large-scale wind turbines, blade lengths have increased significantly. Consequently, rotor swept area has risen, resulting in higher levels of aerodynamic noise. Research has demonstrated that incorporating serrated structures at the trailing edge of wind turbine blades can effectively reduce far-field radiated noise. Liu, through biomimetic blade design using reverse engineering methods, found that trailing edge serrations alter the distribution of peak noise in the flow field and effectively suppress noise peaks across most frequency ranges. Wang investigated the influence of serrated structures on the aerodynamic noise of the DU91-W2-250 aerofoil, commonly used in wind turbines. The experiments were conducted in a low-speed, fully anechoic wind tunnel, focusing on the effects of different serration geometries under varying angles of attack and the influence on both far-field aerodynamic noise and surface pressure distribution. The results indicated that the dipole distribution characteristics of the far-field sound pressure level were more obvious, and the overall sound pressure level was 5 to 10 dB lower than that of the near-field.

Most existing studies on serrated blade structures focus on noise reduction and aerodynamic performance separately. Hower, comprehensive experimental studies examining the trade-offs between noise mitigation, power generation efficiency, and structural loads remain limited. There is a notable lack of experimental research attempting to validate noise reductions in the field, at full scale. Windey have equipped a full-scale turbine with noise reducing, serrated structures and completed a measurement campaign including neighbouring turbines without the devices to act as a baseline. This paper presents the results from these field tests together with accompanying simulation results to form an integrated study on the noise, power generation, and load performance of blades with serrated construction.

2 Evaluation methods

2.1 Noise assessment method

For each wind speed bin, the apparent A-weighted sound power level $L_{W\,A,i,k}$ in each $1/_3$-octave frequency band is determined from the corresponding background-corrected sound pressure level $L_{v,c,i,k}$ measured in the same band. The relationship is expressed by the following equation:

$$L_{WA,i,k} = L_{v,c,i,k} - 6 + 10\log\left[{4\pi R_1^2}/{S_0}\right], \tag{1}$$

where R_1 denotes the distance between the sound source and the microphone, and S_0 is the reference surface area, typically set to 1 m^2. The constant term -6 dB is an empirical correction factor that accounts for hemispherical sound radiation over a reflecting plane. This equation enables the conversion from measured sound pressure levels to apparent sound power levels under standardized conditions.

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The A-weighted apparent sound power level for a given wind speed interval k, denoted as $L_{W\,A,k}$, is calculated by summing the energy contributions of the A-weighted sound power levels across all one-third octave bands. The calculation follows the logarithmic energy summation method defined in international acoustical standards, and is expressed as:

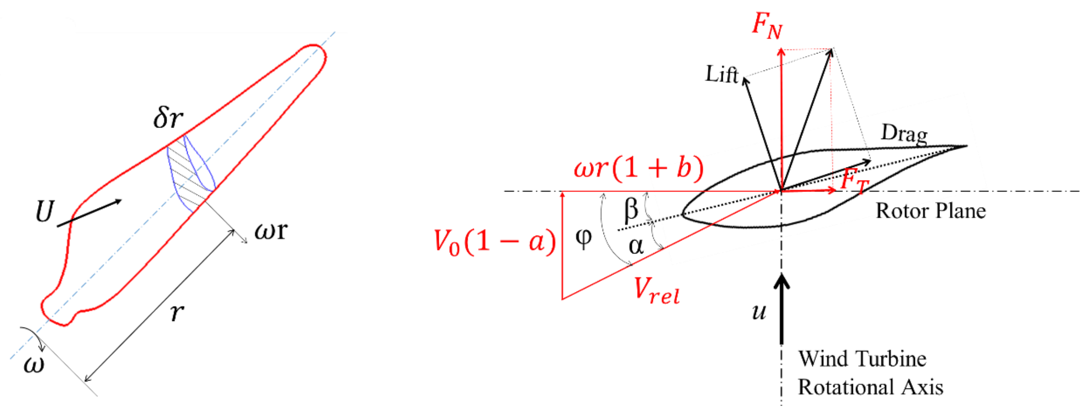
$$L_{WA,k} = 10 \log \sum_{i=1}^{28} 10^{\frac{L_{WA,i,k}}{10}} \tag{2}$$

This approach converts each sound power level from decibels to its linear energy equivalent, performs the summation, and converts the total back to decibels. The logarithmic summation is performed in the energy domain to ensure an accurate aggregation of broadband noise across the full frequency spectrum, in accordance with standard acoustic practices.

This method ensures that broadband noise emitted by the wind turbine is accurately represented by accounting for contributions from all relevant frequency bands. It is in line with standard procedures specified in IEC 61400-11 and ISO 9613-1, which are widely adopted for acoustic measurement and environmental sound evaluation in wind turbine applications.

2.2 Blade Element theory

The Blade element theory (BEM) is a widely used mathematical model. The physical model of BEM divides the blade into a series of infinitely blade micro-segments. Blade element is shown in Figure 1.



a. Blade element model **b. Blade element parameters**

Figure 1: Blade Element schematic diagram

The projections of the lift L and resistance D on the wind rotor plane F_T and normal to this F_N shown in equations:

$$F_N = L\cos\phi + D\sin\phi \tag{3}$$

$$F_T = L\sin\phi - D\cos\phi, \tag{4}$$

The momentum theory mainly calculates the induced speed in the axial and tangential directions. Afterwards, the effect of the induced speed on the inflow parameters is analysed. Finally, the aerodynamics of the blade element can be solved. BEM theory can be obtained through simultaneous solution of the momentum theory formulas and blade element theory formulas. Blade element is rotating with wind rotor. $V, V_{\text{rot}}, V_{rel}$, represent the axial velocity, tangential velocity and resultant velocity,

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respectively. Axial and circumferential induction factors are represented as a and b. Therefore, from the speed triangle, the relative velocity of the blade element can be expressed as follows:

$$V_{rel} = \sqrt{V^2 + V_{rot}^2} = \sqrt{V_0^2(1-a)^2 + \omega^2 r^2 (1+a')^2} \tag{5}$$

Using the relationship between V_0 and V_{rel}, the relationship between the axial velocity factor a and the circumferential velocity factor b are as follows:

$$a = \frac{BcC_N}{8\pi r \sin^2\phi + BcC_N} \tag{6}$$

$$b = \frac{BCC_T}{8\pi r \sin\phi \cos\phi - BcC_T} \tag{7}$$

In the above formula, r is the distance to the hub; c is the chord length; B is the number of the wind turbine blades; C_N is the coefficient of normal force; C_T is the coefficient of tangential force.

3 Test model and measurement equipment

3.1 Test model and site condition

This study is based on a wind turbine prototype with a rated power of 3000 kW, equipped with 76-meter-long blades and mounted on a 100-meter tower. Simulation were run using the commercial software DNV Bladed to evaluate the power output and structure loads before and after the implementation of blade trailing-edge serrations. To ensure consistency between experimental and simulation conditions, site-specific wind parameters were used in the simulations, aligning with the conditions of the noise testing site. The simulations were performed using a characteristic turbulence intensity of 0.095, an average wind speed of 5.87 ms^{-1}, and an air density of 1.196kg/m^3.

This study integrates experimental investigation of serrated trailing-edge blade noise with numerical analysis of wind turbine load and performance. By addressing the current lack of experimental data on the power generation and load characteristics associated with noise-reduction structures, this research contributes a more holistic understanding of the aerodynamic and structural impacts of blade serrations. The findings are intended to provide a valuable reference for future research in the field of wind turbine noise control and structural optimization. The wind turbine model studied in this paper belongs to the variable speed and pitch horizontal axis wind turbine, which has a good power characteristic for capturing the optimal Cp value. The safety system shutdown pitch rate is 2deg/s and the fine pitch angle is uniformly set to 0 deg. The basic fault triggering is all achieved through external controller.

The wind turbine model adopts the WD156-3000 model developed by Windy. The blade model simulates the characteristics of serrated blades but modifying the lift and drag coefficients for the sections near the blade tip (0 m, 10 m, and 20 m), obtained by the relevant blade manufactures through wind tunnel tests.

The schematic diagram of the serrated structure is shown in Figure 2, and the dimension is shown in Table 1.

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Figure 2: Serration structure schematic diagram

Table 1: Serration dimension

Parameters (mm)	Serration 1	Serration 2	Serration 3	Serration 4
Length (L)	221±2	324±3	410±3	405±3
Height (H)	101±1	140±1	178±2	235±2
Bonding Weigth (W0)	50±0.8	50±0.8	60±1.0	70±1.2

3.2 Noise measurement equipment

The test equipment includes test software, microphones, tuners and other writing and data acquisition modules. All the equipment used in the test met the relevant requirements of the standards IEC 61400-11 Ed.3 and GB/T22516-2015, ensuring a high level of accuracy in the test. The entire acoustic testing system was verified using standard sound sources before and after the test. The test unit is located on a plain, with no tall trees around. The test position is ideal. Considering the influence of the surrounding environment on the test location, it is ensured that the test position is in the downwind direction of the wind turbine. The microphone placement was determined based on the site-specific test conditions and the turbine position, in accordance with the requirements specified in the relevant standards. The distance from test wind turbine to the measurement position was measured to be 218 meters, and height of the microphone relative to the foundation of the test wind turbine was measured to be 0 meters.

3.3 Measurement system

The noise testing of the prototype was conducted at a wind farm in Xuchang city, Henan province, China. Sound pressure levels were measured using a microphone mounted on an acoustically reflective hard board, as illustrated in Figure 3.

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Figure 3: Noise test system

The equivalent continuous A-weighted sound level (LAeq) was obtained using an accoutrement and recorded by a central data acquisition system at one-second intervals. The sampling frequency for non-acoustic parameters was also set at 1 Hz. During the measurement process, periods affected by transient external noise, such as passing vehicles or overhead aircraft, were clearly marked in real time and subsequently excluded from data analysis. Remaining data were validated and corrected based on auditory comparison during post-processing to ensure measurement accuracy.

The test site was located on flat terrain and care was taken to ensure that no large reflective objects were positioned near microphone that could significantly influence acoustic measurements. The reflective board was positioned to meet free-field measurement conditions.

4 Comparison of measurement and simulation data

4.1 Noise measurement data and analysis

The noise was measured respectively for the units without serrated blades and those with serrated blades. The test results are shown in Table 2 and Table 3 respectively.

Table 2 Test results of blades without serration structure

Wind speed at hub height $[ms^{-1}]$	Measured power output [kW]	Measured rotational speed [rpm]	Apparent sound power level [dB]
5.5	870	7.63	106.6
6.0	1206	8.7	106.9
6.5	1483	9.1	107.3
7.0	1961	9.9	108.8
7.5	2309	10.4	110.3
8.0	2644	10.5	110.5

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8.5	2910	10.5	110.3
9.0	3032	10.5	110.2
9.5	3074	10.5	110.0

Table 3 Test results of blades without serration structure

Wind speed at hub height $[ms^{-1}]$	Measured power value [kW]	Measure the rotational speed [r/min]	Apparent sound power level [dB]
6.0	1250	9.02	110.6
6.5	1547	9.3	111.2
7.0	2027	10.0	112.4
7.5	2367	10.4	113.4
8.0	2686	10.5	113.3
8.5	2920	10.5	112.9
9.0	3054	10.5	112.8
9.5	3093	10.5	112.8
10	3095	10.5	112.8
10.5	3100	10.5	112.9
11	3099	10.5	112.8

The serrated structure has no discernible effect on the overall trend of the wind turbine noise curve. Across the entire tested wind speed range from 5.5 ms^{-1} to 11 ms^{-1}, the overall aerodynamic noise initially increases, reaching its maximum near the rated rotor speed, followed by a slight decrease and subsequent stabilization, as shown in Figure 4. When the wind speed at the hub height is 6.5 ms^{-1}, the noise reduction effect of the serrated structure reached the maximum value, which is 3.9 dB. When the wind speed reached 8.5 ms^{-1}, the rotor speed is rated, and the noise reduction is 2.6 dB.

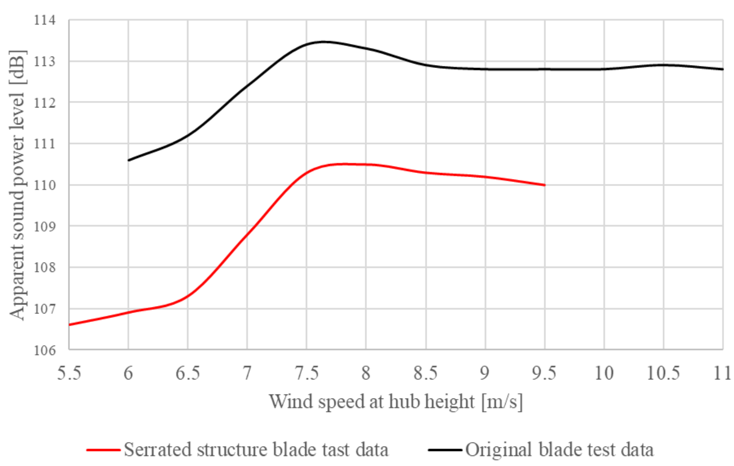

Figure 4: Blade noise test results comparison

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Using the data from the 8 ms^{-1} wind speed bin, a detailed comparison was made of the noise spectrum using the one third octave bands. Over the whole frequency range, the noise values of blades with serrated structures were consistently lower than those of the original blades. Moreover, a significant reduction was observed in the 20 Hz to 125 Hz range. The maximum reduction occurred at 400 Hz, reaching 4.7 dB.

4.2 Simulation and performance comparison

To assess the impact of serrated blades on operational performance, the rotational speed and power output of the wind turbine with different structure were simulated and both compared, as shown in Figure 5 and Figure 6, respectively.

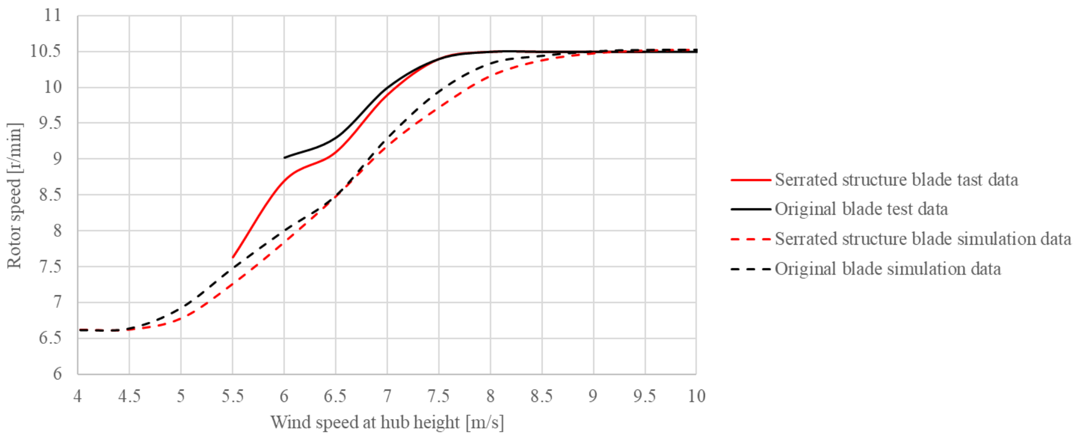

Figure 5: Rotor speed comparison

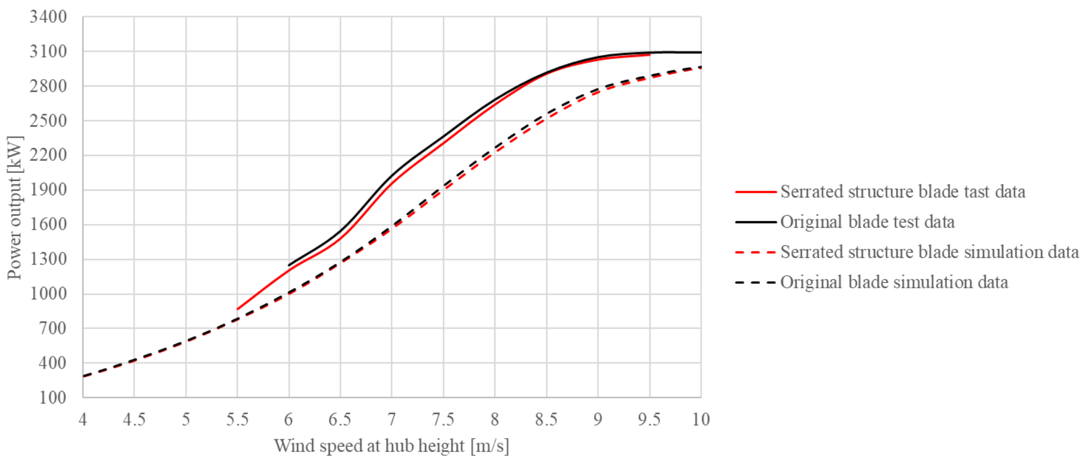

Figure 6: Power production comparison

On the simulation side, the power output was calculated across the full operational wind speed range, from the cut-in wind speed of 2.5 ms^{-1} to the cut-out wind speed of 20 ms^{-1}, based on the wind conditions at the site. The annual equivalent power generation hours of the turbine were calculated based on a Weibull distribution fitted to the site's average wind speed. Both the experimental and simulation results indicate that the rotational speed of the wind turbine with serrated blades was slightly lower than that of the original configuration. While the power curves show minimal differences between measured and simulated ones, the measured power output was generally higher than the simulated values.

Since the anemoscopes were mounted at the end-top of the nacelle, the measured wind speed represents the flow passing through the rotor, which is typically lower than the free-stream wind speed upstream of the rotor. Consequently, this discrepancy can be attributed to variations in real-time wind conditions during the test campaign, leading the turbine to achieve its rated speed earlier than predicted by the simulation. From the perspective of the wind turbine power curve characteristics, both the experimental and simulation results indicate that the performance differences between the serrated and original blades are minimal. Based on the Weibull wind speed distribution, the annual equivalent simulated power generation hours of the turbine with serrated blades are less than 1% lower than those of the original configuration, corresponding to a difference of approximately 25 hours, as shown in Table 4.

Table 4 Annual equivalent power generation hours of two model

Annual equivalent power generation hours [h]	
Serrated structure blade	2530.5
Original blade	2555.8

5 Load simulation and analysis

5.1 Analysis of ultimate load

The Blade model was used to carry out load calculations, and the wind turbine ultimate loads for key components were compared, between the serrated structure blade and original blade The comparison results are shown in Figure 7. The difference in load levels between the two blades is very small. The hub Myz is the only load component that exceeds that of the non-serrated blade by 4%.

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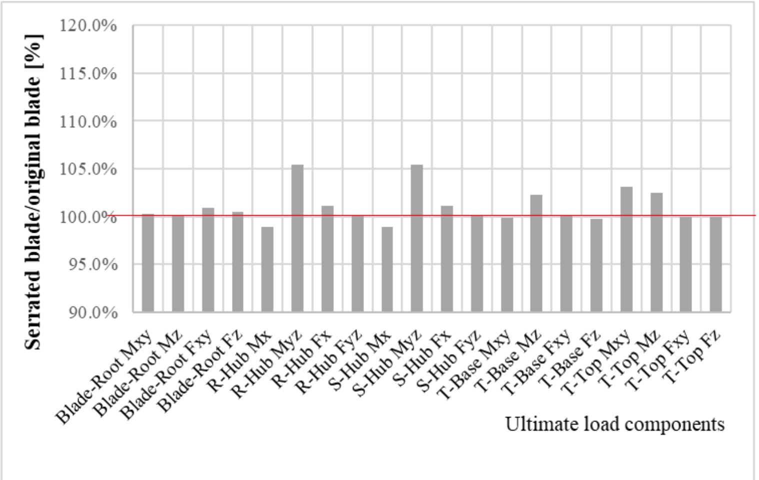

Figure 7: Ultimate load comparison

The timing difference of the load components in the rotating hub coordinate system is analyzed. The load case corresponded to the pitch actuator fault condition defined in IEC 61400-1 The timing condition corresponding to this peak load component is the occurrence of Blade-2 pitch actuator fault at the rated wind speed. The time series is shown in Figure 8. According to the pitch angle corresponding to the peak load moment, the pitch action of serrated structure Blade-2 is delayed by 0.5 seconds compared with that of original blade. This leads to the pitch angle of the serrated structure blade being consistently 2 deg smaller than that of the original one after simulation time 22 seconds, shown in Figure 9. This leads to a more sever imbalance of the rotor, so the rotating hub Myz will be larger than the serrated blades. Therefore, when conducting simulation calculations or site tests with serrated structure blades, especially for the fault condition of pitch angle, the action delay characteristics of blade pitch angle with serrated structure needs to be considered. Moreover, attention should be paid to optimizing the trigger conditions of the pitch change action.

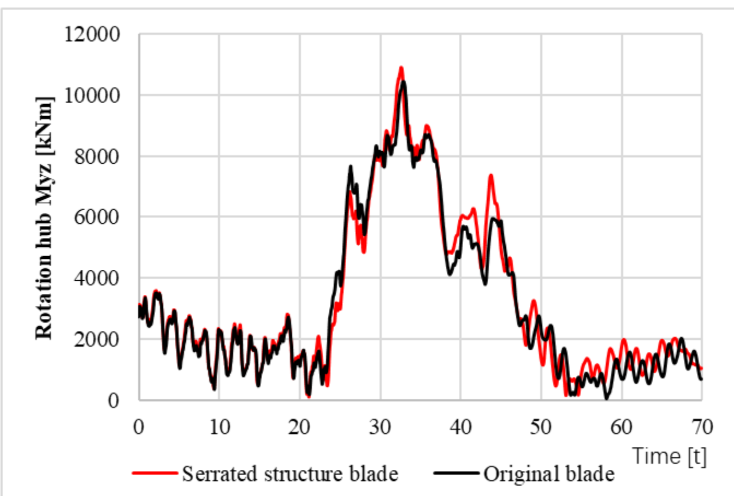

Figure 8: Ultimate load time series comparison

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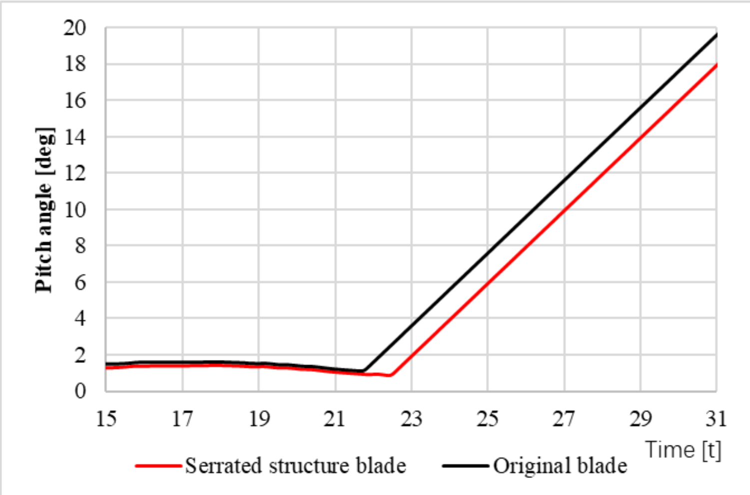

Figure 9: Blade-2 pitch angle series comparison

5.2 Analysis of fatigue load

By comparing the damage equivalent fatigue load of each individual load case, it is found that the load difference is most prominent in the normal power production at 8 ms^{-1} wind speed. For the analysis of a single working condition, the influence of the serrated structure will affect the influence of the frequency of the blades and the entire unit. Based on the spectral analysis of the rotating hub My shown in Figure 10, the peak value of the serrated blade is relatively large near the first order frequency of the tower.

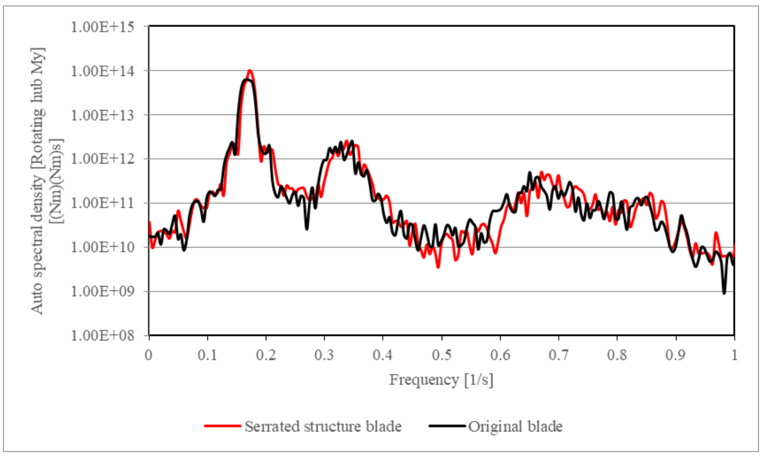

Figure 10: Blade-2 pitch angle series comparison

Overall, the fatigue load of serrated structure blades increases by approximately 2% to 4%, compared to that of original blades, as shown in Table 5.

Table 5 Fatigue load comparison

Coord	*SN*	Mx	My	Mz
B_root	4	100.0%	102.6%	101.1%
R_hub	4	104.3%	103.3%	103.4%
S_hub	4	104.3%	102.2%	102.2%
T_top	4	104.5%	102.3%	102.3%
T_base	4	101.5%	100.2%	102.3%

For the above equivalent fatigue loads, the overall Mx and My coordinate systems of the rotating hub with serrated blades are 3% to 4% larger than those without serrated blades. This load component is also a key parameter during the overall design verification of the machine.

6 Conclusion

This paper assesses the noise, load and power generation of both with and without serrated structure blades based on a wind turbine with a blade extension of 76 meter and a rated power of 3000 kW. Conclusions are as follows:

1) With the installation of the serrated structure on the blade, the aerodynamic noise of the entire machine was significantly reduced. The research wind turbine can achieve a noise reduction of 3.9 dB in the site average annual wind speed range of 6.5 ms^{-1}.
2) The serrated structure has a minimal impact on the power generation characteristics of the wind turbine, with an overall power generation impact of less than 20 hours and an equivalent power generation hour deviation of less than 1%.
3) After the serrated structure were installed on the blades of the wind turbine, fluctuations occurred in the load frequency of the unit and the extreme and fatigue load of the entire machine increased by about 3% to 4%. From the perspective of operational safety, it is necessary to appropriately consider the safety margin of each component with serrated structure.

Author contributions

YZ conducted the software simulations, designed the article structure, and drafted the manuscript.
TH supervised the structural optimization of the paper and provided experimental guidance.
PR contributed to section adjustments and refinement of the manuscript language.
PS contributed theoretical and data support for the experimental part.
JG supported the simulation work and contributed to data analysis.
All authors reviewed and approved the final version of the manuscript.

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