NUMERICAL SIMULATION ON FATIGUE TEST OF COMPOSTIE ROTOR BLADE FOR MULTI-MEGAWATT WIND TURBINE

Joong-Kyu Park[1], Ji-Sang Park[1], Hak-Gu Lee[1*], Sang-Hun Lee[1], Woo-Kyoung Lee[1]

[1] Wind turbine Technology Research Center, Korea Institute of Materials Science, Changwon, Republic of Korea,

* Corresponding author (hakgulee@kims.re.kr)

Keywords: *Wind turbine blade, Fatigue test, Test load, Dual axis loading*

1. Introduction

The lifetime of wind turbines requires over 20 years which is equivalent to fatigue cycles of tens of millions. For larger wind wind turbines such as a wind turbine blade of 5 mega watts, wind turbine designers should adopt a lighter turbine structure. This lightweight trend deteriorates the fatigue resistance of wind turbines. Thus, fatigue tests have to be performed with real size wind turbine components.

Previous fatigue tests of wind turbine blade are as follows. The RISO National Laboratory built a resonance excitation system to apply damage cycles to the blade in a single direction. Thus, the system equipped an electric motor that rotates eccentric mass [1]. The Delft University conducted the fatigue test of wind turbine blades by using an actuator. The Delft University performed a fatigue test by one actuator [2]. Other industries such as automobiles and aerospace also perform single-axis resonance tests. In the aerospace industries, the single-axis resonance tests have been used to test wings [3]. The NREL (National Renewable Energy Laboratory) developed a dual-axis fatigue test system in 1999. This system uses constant amplitude displacements to apply the cycles of fatigue damages [1]. Since wind turbine blades experience fatigue loads in both the flapwise and the edgewise direction, simultaneous dual-axis loading in the fatigue test is more reasonable than the single-axis loading. The NREL conducted a fatigue test in the dual-axis direction. However, since the NREL's equipment applied dual-axis loading at the same position, blade motions in the flapwise and the edgewise direction are interfered by each other.

In this study, a blade's responses under dual-axis loading applied to two separated positions are simulated in order to actualize the dual-axis fatigue test. First, accumulated fatigue damages of the blade under representative wind conditions for 20 years were calculated using 3D full blade model. Then, we calculated equivalent two point loading conditions that cause the same accumulated fatigue damage of the blade. Using the calculated two point loading conditions and a simplified beam model, we analyzed responses of the wind turbine blade under simultaneous dual-axis loading test. From the analysis, we showed the possibility of the simultaneous dual-axis fatigue test for the wind turbine blade.

2. Structure of wind turbine blade

The base model is the KM44 wind turbine blade (KM Co., LTD, Korea). The model's specifications are presented as shown below:

1. 44 m blade length
2. 10.14 ton total weight
3. 13.12 m center of gravity
4. 0.903 Hz and 1.422 Hz of flap and edge natural frequency

3. Finite element method

3.1 Finite element model

To generate the finite element model, we used ABAQUS, one of the commercial finite element programs. A finite element model of the blade is shown in Fig.2, which is a full 3D model having 8,443 shell elements and over 8,083 nodes [6]. The element used in this study is a laminated shell element of four nodes which enables us to calculate transverse shear deformation. Each node has 6 degree of freedom.

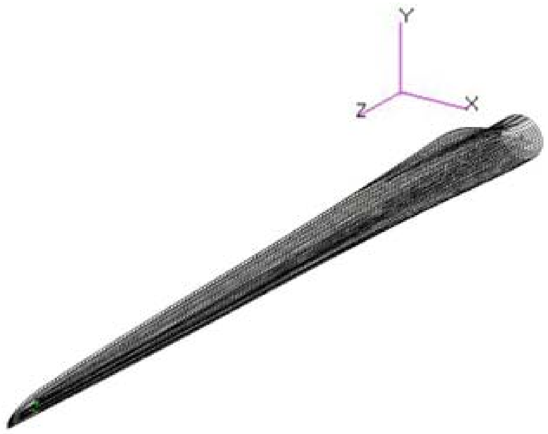

Fig.2. Finite element model of the blade.

3.2 Material properties

The blade is composed of 5 materials. The materials used for our wind turbine blade are glass / epoxy uni-directional 0 degree (UD0), glass / epoxy uni-directional mat (UD Mat), glass / epoxy 2-axis fabric, PVC foam and Balsa wood. The properties of the material are shown in Table.1.

Table.1. Blade material properties

	E_1 [GPa]	E_2 [GPa]	*Poisson ratio*	G_{XY} [GPa]
Glass/Epoxy UD0	43.1	13.2	0.24	3.62
Glass/Epoxy UD Mat	42.5	13.6	0.24	3.62
Glass/Epoxy 2-axis Fabric	28.4	28.4	0.11	3.62
PVC foam	0.06	0.06	-	0.02
Balsa wood	0.1	0.1	-	0.16

3.3 Loading and boundary conditions

Six loading components are applied at twenty stations along the sparcap as shown in Fig.3. At the blade root, all the six degrees of freedom are fixed.

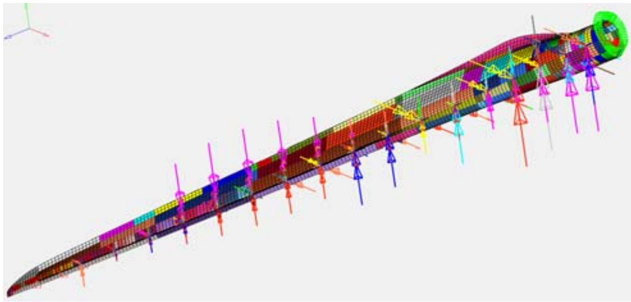

Fig.3. Loads and constraint conditions of the model.

4. Calculation of fatigue damages

It is important to know fatigue loads which experience wind turbine blades in order to simulate the blade fatigue test. The fatigue loads which used to conduct static and fatigue analyses for the design load cases are prescribed in the Germanischer Lloyd guideline. The design load cases are a combination of wind conditions around blade and wind turbine operating situations [4]. Generally, the design load cases (DLC) have about 1000 different loading conditions.

The static and fatigue analysis is conducted using around 200 to 300 design load cases. In order to compute the fatigue loads, it should be calculated considering load cases such as DLC 1.2, 1.10, 1.13, 2.3, 3.1, 4.1, and 6.4 in Germanischer Lloyd guideline.

If the fatigue loads are found, we should calculate the amplitude and mean value of the fatigue loads by rain flow counting method.

4.1 Markov matrix

After rain flow counting, we can derive a fatigue load spectrum called the markov matrix. The markov matrix is a chart which has a mean value and amplitude of the fatigue loads.

4.2 Miner's rule

Accumulated fatigue damages can be calculated by Miner's rule which requires the fatigue load spectrum. The equation is ratio of the number of applied load to the number of cycles at failure [4]. Miner's rule defines the accumulated fatigue damages D as

$$D = \sum_i \frac{n_i}{N_i} \tag{1}$$

D = accumulated damage
n_i = number of applied cycles of i actions
N_i = number of cycles to failure of i actions
i = load case index

Generally, permissible fatigue life of a composite laminate can be obtained by performing a tension-tension or tension-compression lab test. The wind turbine blade, however, is difficult to conduct the lab test due to its huge size. We need to use the equation prescribed in the Germanischer Lloyd guideline to calculate the permissible fatigue life [4]. The equation requires various factors such as a partial safety factor, mean values of the characteristic actions, and resistances for tension and compression. Accumulated fatigue damages in the blade are calculated by substituting the parameters into the rule. The distribution of the maximum fatigue damage at each wing section according to the spanwise direction is shown in Fig.5.

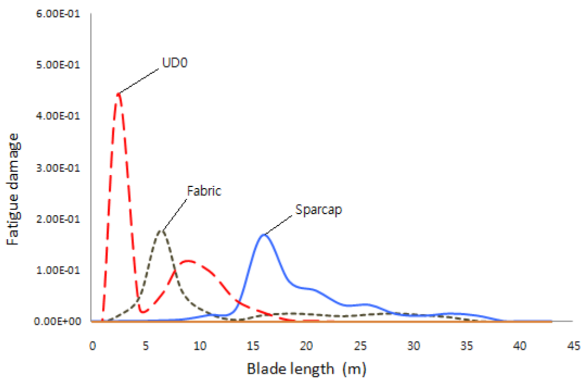

Fig.5. Distribution of the maximum fatigue damage at each wing section by distributed loads.

5. Equivalent two point loading conditions

5.1 Goodman diagram

Fig.6 shows relation between minimum and maximum load amplitude ratio (R-ratio) into the appropriate Goodman diagram [5].

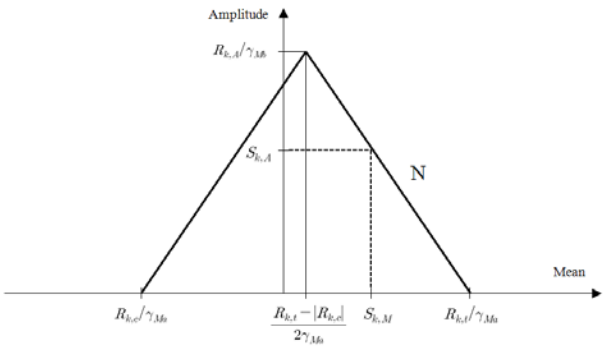

Fig.6. Goodman diagram.

In order to find the fatigue test load, we need to perform two transformation steps. First, all of the mean values in the markov matrix at control point should be converted into zero by the equation (2) [3]. This process is shown in Fig.7.

$$\frac{\sigma_a}{\sigma_{a,m=0}} + \frac{\sigma_m}{\sigma_u} = 1 \tag{2}$$

σ_a : Amplitude of load spectrum
σ_m : Mean value of load spectrum
$\sigma_{a,m=0}$: Modified amplitude
σ_u : Ultimate strength per materials

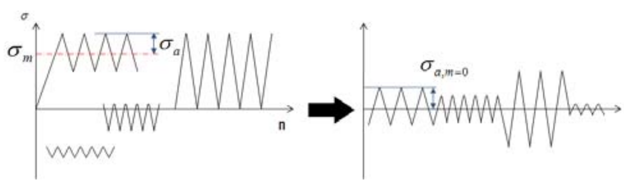

Fig.7. Setting the mean value of fatigue test to zero.

5.2 Effective equivalent load

Second, various amplitudes of the fatigue load spectrum should be converted to a constant amplitude as shown in Fig.8. Generally, the constant amplitude can be loaded by the test equipment. The modified load spectrum is converted to constant amplitude by using an effective equivalent load equation (3) [5].

$$\sigma_{ca} = \left[\frac{\sum (\sigma_i^{\,m} \cdot n_i)}{n_t} \right]^{1/m} \qquad (3)$$

σ_{ca} : Constant amplitude

σ_i : Modified load amplitude

n_i : Existing number of load cycles of a class i actions

n_t : Test cycle

m : Slope parameter

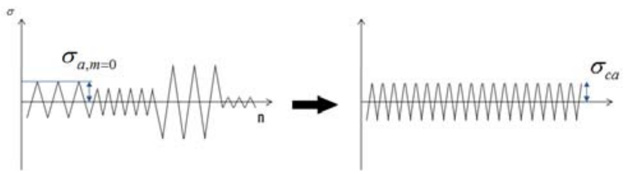

Fig.8. Converting various amplitudes to constant amplitude.

5.3 Two point loading conditions

The wind turbine blade experiences fatigue cycles of tens of millions during its lifetime. The fatigue load spectrum is very complex. Thus, it is impossible to apply the fatigue load spectrum to the blade during fatigue test because very sophisticated test equipment and long testing time are needed. Therefore, the blade fatigue test is generally conducted within one or two million cycles using either single or dual loading test. In this study, the fatigue test simulation was conducted using one million cycles and under the dual-axis loading.

A fatigue test load is relative to stress amplitude which matches the fatigue stress at the control point. The control point corresponds to the maximum damage position of each material.

If the position of loading, testing frequency and test cycle were determined, we can find the fatigue test loads by comparing to the accumulated fatigue damage as shown in Fig.9.

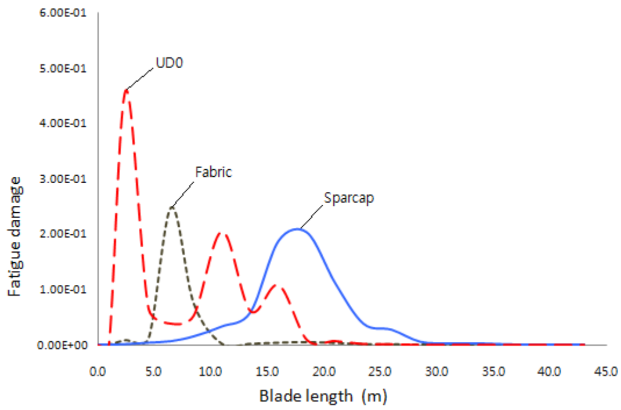

Fig.9. Distribution of the maximum fatigue damage at each wing section by two point test loads.

Fatigue damages in the UD0 and sparcap occurs due to the flapping motion while that in the fabric occurs due to the edgewise deflection. Hence, it is necessary for us to perform a dual-axis fatigue test.

In order to determine appropriate amplitude and a frequency of cyclic test load, deflection responses and resultant stress amplitude induced by the fatigue test load were investigated based on steady state dynamic analysis for the 3D full finite element model of the blade.

6. Beam model

Wind turbine blades are designed to satisfy various design conditions. It is not efficient to conduct blade fatigue test simulation with all 3D full models because of its long analysis time and difficulties in observing various blade fatigue behavior. Furthermore, obtaining the 3D full blade model is difficult as this is a company's confidential information. Therefore, this paper presents an alternative fatigue testing method which converts the complex 3D full blade model into a simplified beam model of 1D type. Missing shape data while converting 3D full model to 1D beam model were defined as element constants to the beam model. The beam model was then divided into multiple nodes, and section information was applied to each individual node.

A B32 beam element in ABAQUS allows basic axial, bending, and twisting deformation. As shear deformation is also significant in blades, it is considered to be under Timoshenko beam elements rather than Euler beam elements because the effect of shear deformation is significant in this case. A 3-node-element was then used on the B32 beam element to apply mass moment of inertia in the center of the node. Root of the model is clamped.

The cross sectional shape of the beam model is different from the shapes given by ABAQUS. Therefore, a section engineering property was applied to the beam elements using the beam general section in ABAQUS instead of cross section dimensions of the 3D full blade model. The properties and material information were inputted directly into the beam model.

In order to verify the reliability of the beam model, we compared the blade properties and static responses both the beam model and the 3D full model. The error of first natural frequency is 1.21% in comparison with the 3D full model. The errors of total mass and center of gravity are 1.32% and 0.7%. For static analysis, the same load was applied to the models' tip. Error of 4.4% and 7.3% were recorded in the flapwise and edgewise direction respectively. The results are shown in Figs.10 and 11. Therefore, the beam model matches the 3D full model within 10% errors, and we can use the beam model to resonance analysis of fatigue test.

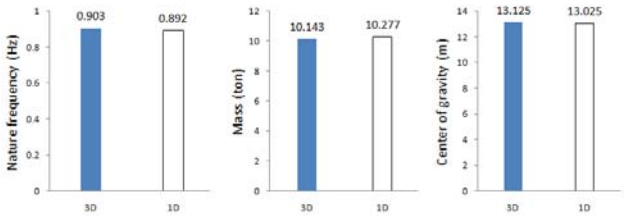

Fig.10. Results of blade properties of 3D full model and 1D beam model.

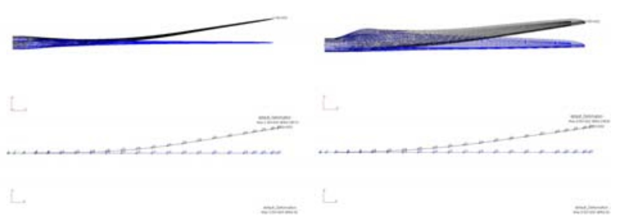

Fig.11. Results of static analysis response.

7. Blade responses under the dual-axis loading

The steady state dynamic analysis cannot perform the dual-axis loading in different frequency for each direction. Transient response analysis was conducted to simulate various test conditions of dual axis loading. For this purpose, the beam model was constructed with the beam element which has equivalent stiffness and mass distributions to the full 3D finite element model of the blade. In doing so, the transient response analysis could be conducted in reasonable time frame.

The blade deflections in both directions are shown in Figs.12 and 13. The model tip deflection is 4.8-5.1 m flapwise and 0.5-1.1 m edgewise direction. This results show that the motion of blade is reasonable. Therefore, the simulation of dual-axis fatigue test at different loading positions can be performed by model transient resonance analysis.

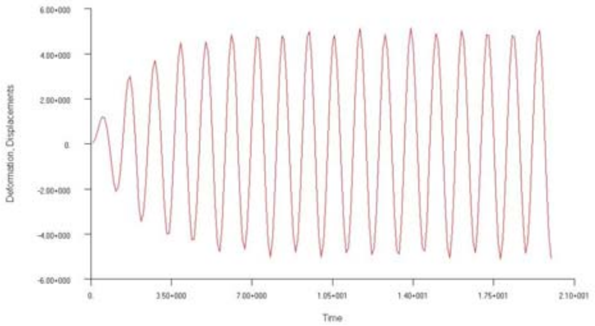

Fig.12. Tip deflection of flapwise direction.

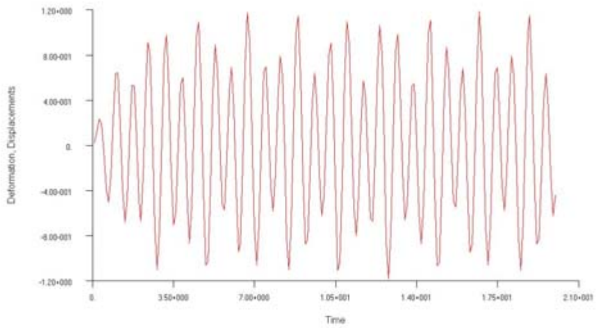

Fig.13. Tip deflection of edgewise direction.

8. Conclusion

Based on the above numerical simulations, various test conditions were studied including simultaneous dual loading, resonance type, and various loading positions. From the analysis, we can found that the modal transient resonance analysis is appropriate for the dual-axis fatigue test at different loading position. In conclusion, realistic and reasonable loading methods in fatigue test of the 44m long blade were successfully determined. The essential information for designing a fatigue test machine was presented.

9. References

[1] Darris White, *"New method for dual-axis fatigue testing of large wind turbine blades using resonance excitation and spectral loading", National Renewable Energy Laboratory*, 2004.

[2] van Delft, D., van Leeuwen, J., Noordhoek, C., Stolle, P., *"Fatigue testing of a full scale steel rotor blade of the WPS-30 wind turbine", Journal of wind engineering and industrial aerodynamics,* 27 (1988), Amsterdam.

[3] Bleakney, W., *"Fatigue testing of wing beam by the resonance method", NACA TN-660*, August 1938.

[4] Germanischer Lloyd, "*Guideline for the Certification of Wind Turbine*". *WindEnergie GmbH*, Edition 2003.

[5] Yung-Li Lee, Jwo Pan, Fichard, B. Hathaway, Mark E. Barkey, *"Fatigue and testing analysis (theory and practice)"*, Elsevier Butterworth Heinemann, 2005

[6] Ji-sang Park, Kyung-Su Cho, "Structual design and analysis of 3MW offshore wind turbine blade", World Wind Energy Conference and Exhibition, 2009.