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Design of Foam Core Sandwich Composite Blade for A Small Scale Wind Turbine System Considering Fatigue Life

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SUMMARY

This work is to propose a specific structural design and analysis procedure for development of a low noise 500W class wind turbine system that will be applicable to relatively low wind speed region like Korea. The proposed structural feature has a skin-spar-foam sandwich composite structure with E-glass/Epoxy face sheets and Urethane foam core for lightness, structural stability, low manufacturing cost and easy manufacturing process. Structural design and analysis including load cases, sizing, stress, deformation, buckling and vibration was performed using the rule of mixture and the Finite Element Method. Moreover fatigue life was estimated using the proposed method. In order to evaluate the designed blade structure the structural tests were carried out, and their test results were compared with the estimated results.

Keywords: Wind turbine, Structural design, Composite sandwich structure

INTRODUCTION

Recently, development of alternative energy instead of the fossil fuels is very hectic. Among them the wind energy is especially a strong candidate due to unlimited natural energy resource and clean, hence lots of studies on the wind turbine system using the wind energy are actively carried out in many countries. Current development trend of the wind turbine system is mostly several MW class large scales. Large scale wind turbine systems more than several MW need some special requirements, for instance, provision of large wind turbine site, expensive manufacturing facility and equipments. However, even though the small scale wind turbine system produces relatively a small electric energy, it has been continuously developed due to many advantages such as easy manufacturing, low cost, personnel handling and operation, etc.

Because most recent commercialized small scale wind turbine systems have been designed at the rated wind speed more than 12 m/s, they show a great reduction of aerodynamic performance in low wind speed region like Korea.[1,2] Therefore the proposed wind turbine system in this study has the proper rated wind speed, i.e., 8 m/s.

This work shows an aerodynamic and structural design result for the 500W-class wind turbine system with the low noise character for local area use. Material for this wind blade is the glass/epoxy that has good structural performance such as long fatigue life and low cost.[3] Prototypes of the designed blades are manufactured by autoclave curing process. The structural test is performed to evaluate the structural design and analysis results with the real structural behaviours.

BLADE DESIGN

In the blade design, aerodynamic design is firstly performed according to design requirements, and its results are evaluated through performance analysis and test. The blade structure is designed according to structural design loads that are found from the load case analysis. For the structure analysis, the Finite Element Method(FEM) is used as a mean for investigating stresses, displacements, structural stability, etc. Also the blade resonance is checked by eigenvalue analysis which can obtain natural frequencies and modes. Using the Spera's empirical equations and the S-N linear damage method[6], the required durability more than twenty years is examined. Finally, after manufacturing the prototype wind turbine blade, the structural experimental test is conducted. Fig. 1 shows the blade design flow that is applied to this work.

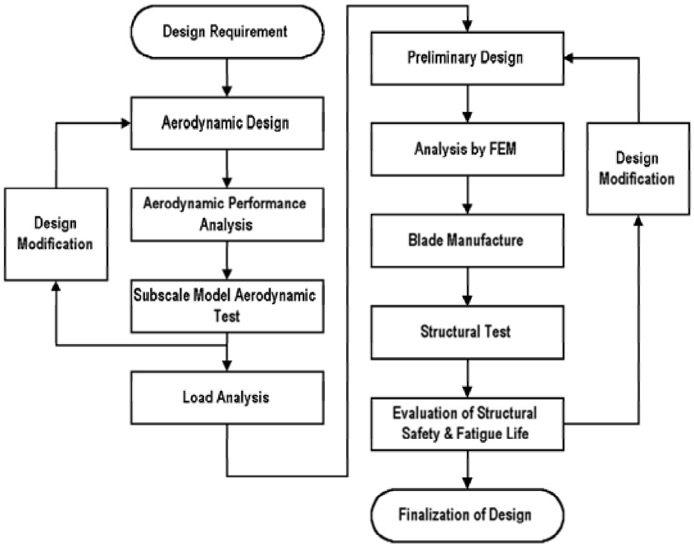

Figure 1. Flow of aerodynamic and structural design.

Blade aerodynamic design

Table 1 shows the wind turbine system design specification including rated power of 500W, rated wind speed of 8m/s, cut-out wind speed of 20m/s, number of blades of 3 and blade material of glass/epoxy. For electric power generation, the gearless direct drive type AFPM(Axial Flux Permanent Magnet) generator is used for design simplicity. Aerodynamic design results including DU 93-W-210 airfoil, blade diameter of 2.47m, twist angle of 24.353 degrees and blade tip and root chord are expressed as Table 2, and blade aerodynamic configuration is shown in Fig. 2.

Table 1. Design specification of 500W wind turbine system

Type	Horizontal axis wind turbine system
Target rated power	500 W
Operation condition	Rated wind speed: 8 m/s, Cut-out wind speed: 20 m/s
Number of blades	Three
Blade material	Glass/epoxy composite

Table 2. Aerodynamic design results of small wind turbine blade

Rated power	500 W
Rotor diameter	2.47 m
Blade root chord	149.208 mm
Blade tip chord	42.727 mm
Blade total twist	24.353 deg
Airfoil	DU 93-W-210

Figure 2. Designed aerodynamic configuration of blade

Blade performance analysis

Blade performance analysis is performed for investigation whether the aerodynamic design of blade is acceptable for the design target performance. For this analysis, the following equations are used, and the calculation flow is coded by a computer program. Where the power coefficient is calculated using following equation; [4]

$$C_p = \frac{2M \times \lambda_0}{\rho S V_1^2 R} \quad (1)$$

And the mechanical power and the electronic power are calculated as follows;

$$P = \frac{1}{2}\rho C_p S V^3 \quad (2)$$

$$P_e = \eta_g P \quad (3)$$

Where M; moment, R; radius, ρ; air density, V_1; wind speed before passing blade, λ_0; tip speed ratio, η_g; generator efficiency.

As a result of the performance analysis, the power coefficient for tip speed ratio of 7 is maximum, and it is decreased for other tip speed ratios. The power coefficient versus the tip speed ratio and the calculated power according to various power coefficients is shown as Fig. 3. Through this analysis, it is confirmed that the aerodynamic design acceptable for the target rated performance because the blade power at the rated wind speed of 8m/s is a bit higher than the target requirement power.

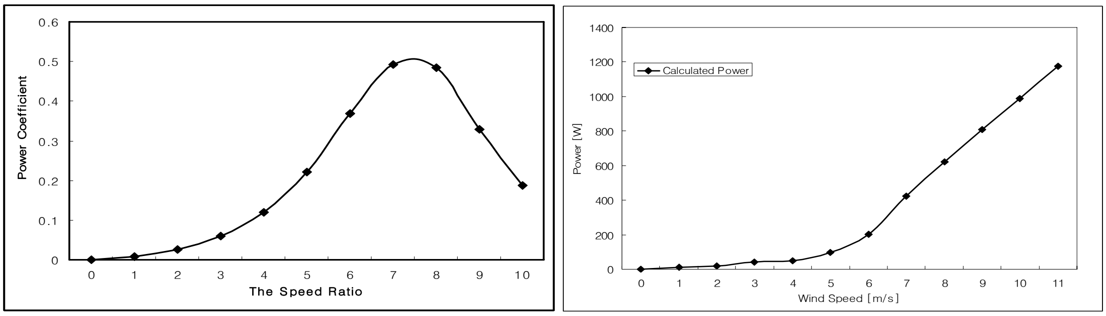

Figure 3. Power coefficient vs. tip speed ratio and calculated generator power at various wind speeds

Blade structural design

The aerodynamic loads and the centrifugal forces are mainly acting on the blade. The centrifugal forces can be simply calculated from rotational speed, and the aerodynamic loads are calculated by aerodynamic coefficients in several load cases which are mentioned in Table 3. The shear force and bending loads can be defined by the normal force distribution acting on each section of the blade, and their variations depend on the wind speed and the incidence angle. Therefore, the bending loads must be calculated by consideration of operating conditions. According to the load analysis, the load case 2 is the most severe condition. Hence the structural design was performed on the basis of the load case 2. Figure 3 shows the flap-wise moment distribution of the design load case 2. The blade adopts the skin/spar/foam sandwich type structure.[4]

By the preliminary composite design method such as the netting rule and the rule of mixture that was proposed at the previous study.[2] The initial structural design is carried out, and then the initial design feature is repeatedly modified by structural analysis using the finite element method. The bending force is kept by the spar flange layered by plies angle of 0°/90° and the torsion is kept by the upper and lower skins layered with the angle ply ±45°. Table 4 shows the final structural design results.

Table 3. Several load cases for structural design

Load case	Case 1	Case 2	Case 3
Reference wind speed	8m/s	20m/s	55.0m/s
Gust condition (±20m/s, ±40°)	Without gust	With gust	Storm
Rotational speed	433rpm	1069rpm	stop

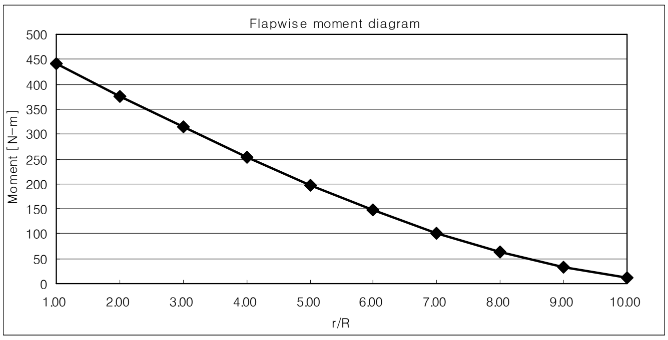

Figure 3. Flap-wise bending moment diagram at load case 2

Table 4. Final structural design results

Station (r/R)	Thickness (mm)	
	Upper surface	Lower surface
Root ~ 0.1	Skin 1t(4ply)/Spar 6.75t(27ply)	Skin 1t(4ply)/Spar 6.75t(27ply)
0.1 ~ 0.2	Skin 1t(4ply)/Spar 2.25t(9ply)	Skin 1t(4ply)/Spar 2.25t(9ply)
0.2 ~ 0.3	Skin 1t(4ply)/Spar 2.70t(11ply)	Skin 1t(4ply)/Spar 2.70t(11ply)
0.3 ~ 0.4	Skin 1t(4ply)/Spar 3.15t(13ply)	Skin 1t(4ply)/Spar 3.15t(13ply)
0.4 ~ 0.5	Skin 1t(4ply)/Spar 3.15t(13ply)	Skin 1t(4ply)/Spar 3.15t(13ply)
0.5 ~ 0.6	Skin 1t(4ply)/Spar 3.15t(13ply)	Skin 1t(4ply)/Spar 3.15t(13ply)
0.6 ~ 0.7	Skin 1t(4ply)/Spar 3.15t(13ply)	Skin 1t(4ply)/Spar 3.15t(13ply)
0.7 ~ 0.8	Skin 1t(4ply)/Spar 1.35t(6ply)	Skin 1t(4ply)/Spar 1.35t(6ply)
0.8 ~ 0.9	Skin 1t(4ply)/Spar 0.45t(2ply)	Skin 1t(4ply)/Spar 0.45t(2ply)
0.9 ~ 1.0	Skin 1t(4ply)/Spar 0.225t(1ply)	Skin 1t(4ply)/Spar 0.225t(1ply)

Structural analysis

In order to perform the structural analysis, a finite element code, MSC Patran/Nastran, is used. In this analysis, the linear static stress analysis, the eigenvalue analysis and the buckling analysis are carried out. The intra-lamina 'Tasi-Wu' failure criterion is used to find the structural safety.[5] The boundary condition is assumed that the blade root is fixed. Distributed aerodynamic loads were applied on the blade along length direction and centrifugal body force also applied. According to the analysis results, it is confirmed that the blade is safe from the strength. The linear static structural analysis results, the natural frequency analysis result and buckling analysis result are presented in Table 5. Stress distribution and deformed blade shape at the load case 2 are shown in Fig. 4. In this analysis, it is confirmed that the final structural design feature is safe from strength, the deformed blade tip clearance requirement, buckling and resonance possibility. (See Fig. 5, Table 5)

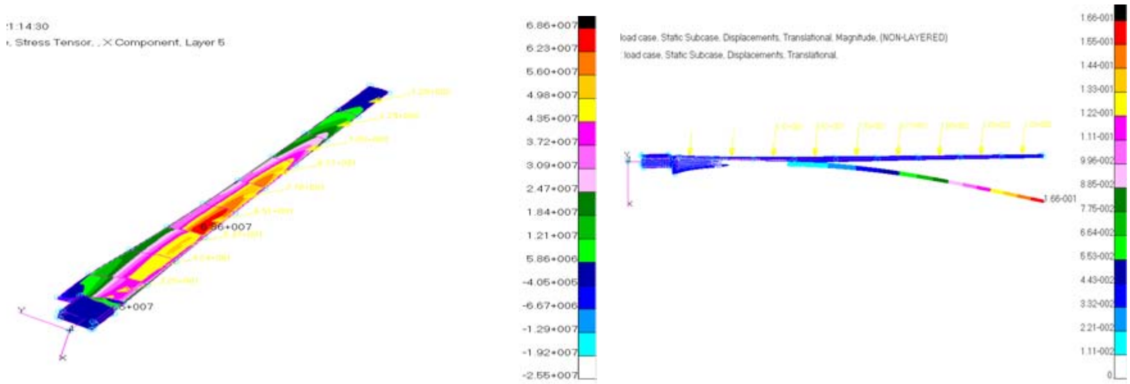

Figure 4. Stress distribution and deformed blade shape at load case 2

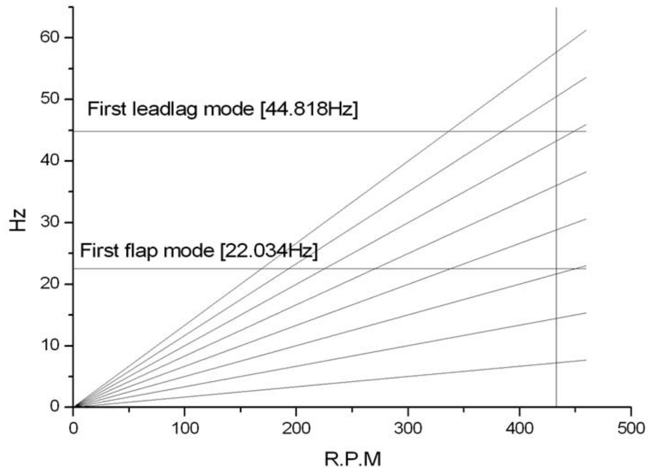

Figure 5. Campbell diagram for load case 2

Table 5. Structure analysis results

		Case 2
Max. stress [Mpa]	Ten.	68.7
	Com.	25.5
Max. Disp. [mm]		166
Natural frequency(Hz), (First flap mode)		22.034
Buckling load factor, (First buckling mode)		1.2073

EVALUATION OF FATIGUE LIFE

Spera proposed the empirical equations, which were based on a set of test data that was broad enough in scope to include the sizes and types of rotors and towers, the types of terrain, and the types of wind conditions expected for future HAWT power stations.[6] Calculation of dynamic loads with these equations is essentially a process of interpolation rather than extrapolation. The flap-wise and chord-wise cyclic moments will be calculated in this study. Table 6 shows calculation results for blade flap- and chord-wise cyclic bending moment(δ My, δ Mz), where n is the number of standard deviation(n=0 for the 50th percentile load, n=1 for the 84th percentile load, n=2 for the 98th percentile load)

Table 6. Results of fatigue load calculation

n	δ My [Nm]	δ Mz [Nm]
0	17.606	121.215
1	27.378	201.857
2	43.405	348.227

In order to perform comparison between the allowable fatigue stress for 20 year fatigue life and the estimated fatigue stress by fatigue load from the Spera's empirical formulae, the calculated maximum flap-wise and chord-wise bending moments in Table 7 are applied to the finite element model of the wind turbine rotor blade.

According to the FEM analysis results, the maximum tensile and compressive stresses are 39.4MPa and 23.5MPa, respectively. Fig. 6 shows stress distribution at the spar layer under maximum fatigue load.

Therefore, the designed wind turbine blade satisfies the design criteria for the fatigue life of 20 years.

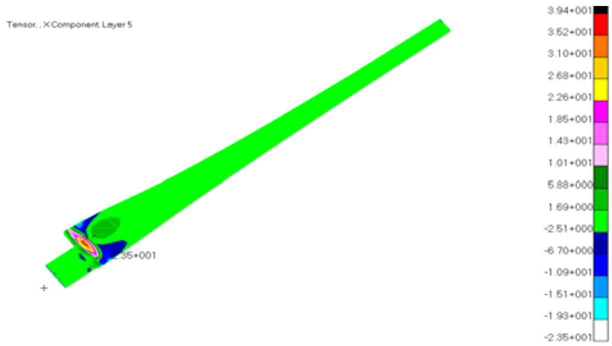

Figure 6. Stress analysis result for cyclic load

BLADE MANUFACTURING AND STRUCTURAL TEST

Blade manufacturing

In order to manufacture the prototype blade, the autoclave curing method is adopted. In the manufacturing process, the styrofoam mold is firstly manufactured using steel plate templates and hot wires, and then glass fabrics for the second mold are layered-up on the styrofoam mold with special coating. After that, the glass fabrics are layered-up on the second mold once again for last mold, and then glass fabrics for the upper and lower surface skins of the blade are layered-up on the last mold according to the structural design result.(See fig. 7) The cured upper and lower surface skins are bonded by epoxy, then the urethane foam is injected into the space between upper and lower skins. After completely curing the blade, the proper coating is applied. The manufactured blade is shown in Fig. 8.

Figure 7. Lay-up process on the mold

Figure 8. First blade prototype

Structural test

In order to evaluate the design results, the structural test must be performed, and then the structural test results must be compared with structural analysis results. The structural test of prototype was conducted by the hydraulic structural test equipment which can adjust the applying load and speed by 3 hydraulic cylinders and a controller. In this test, three point loads are applied to the prototype for test simplicity. Figure 9 shows that the blade is tested at the load case 2 simulated by the three point loading method. Table 7 shows comparison between the FEM analysis results and the structural test results of the prototype. This comparison shows that analysis results are well agreed with the experimental results.

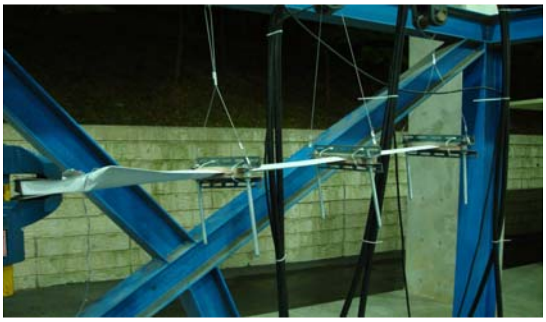

Figure 9. Structural test of prototype blade

Table 7. Comparison between the static analysis results and the test results

Item	Analysis result	Test result
Stress	29.1MPa	27.4MPa
Tip deflection	166mm	152mm

BLADE PERFORMANCE TEST

In order to evaluate the target design performance, the blade performance test is performed with the test purpose tower. In the performance test, because the natural wind conditions around laboratory could not be made, a truck was used instead of the natural wind condition. The simulated speed of the wind turbine system installed on the truck is from 3 to 11 m/s.

For the designed wind turbine performance test, a gearless generator ‘SYG-A208-600-570’ that has advantages such as simple due to gearless, easy blade mounting and low is used. Other test equipments are a rectifier, resistances for electrical loading, a multi-meter and a photo sensor and an instrument for measuring the blade rotational speed. Figure 10 shows the wind turbine performance test equipments and the wind turbine system during experimental performance test

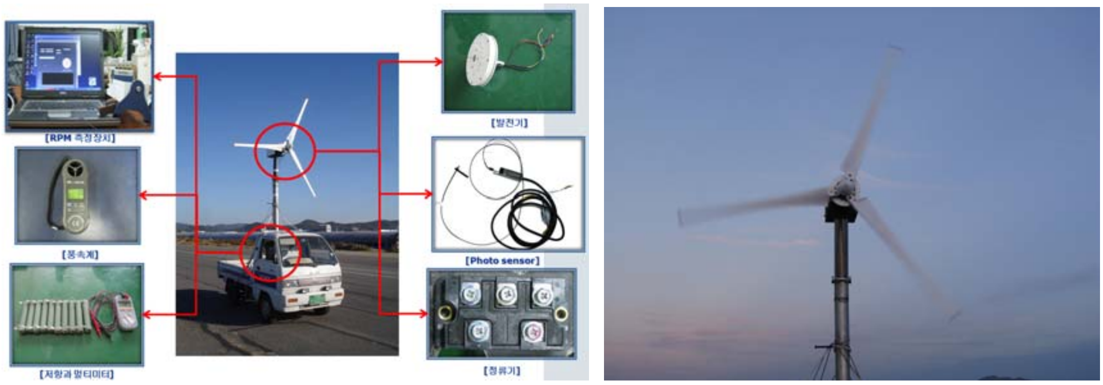

Figure 10. Blade performance test equipments and wind turbine system during experimental performance test

Figure 11 shows comparison between the estimated power and the experimental result of power. In the graph, it can be found that the wind turbine start around the wind speed of 4 m/s, and its power increases rapidly at 6 m/s. This comparison result reveals that the test result is reasonably agreed with the estimation result in all operating range. Table 9 shows comparison between the test result and the analysis result at 8 m/s.

Table 8. Comparison between test result and analysis result at 8 m/s.

	Analysis result	Test result
Power [at 8m/s]	669 W	663 W

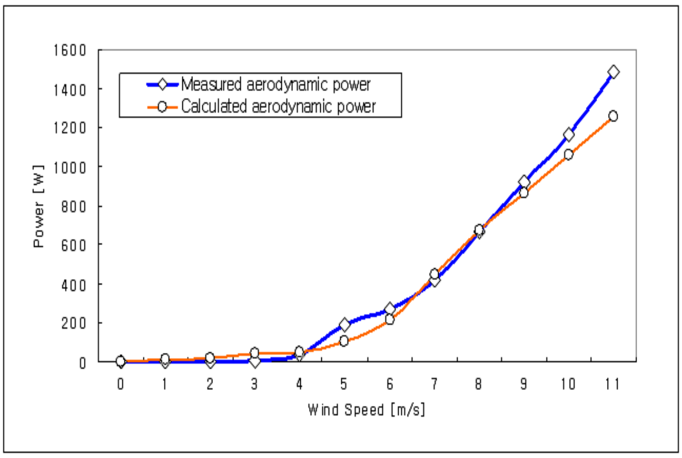

Figure 11. Comparison between estimation and the experimental test result of power

CONCLUSIONS

In this work, all design activities including aerodynamic and structural design, analysis and test of a 500W-class wind turbine system were performed. Through the structural analyses, it was confirmed that the final structural design feature is safe from strength, the deformed blade tip clearance requirement, buckling, resonance possibility and maximum strain requirement for fatigue life. In addition, it was found that the blade has enough safety for the bird strike in extreme storm condition.

The prototype of the designed blade was manufactured, and the structural test was performed to evaluate the structural design and analysis results Through the evaluation, it was found that structural analysis results are well agreed with the experimental results

Finally, in order to evaluate the wind turbine system including the designed blades and the gearless generator, the performance test is performed using a truck for simulating the natural wind speed and other measuring equipments. According to the performance evaluation result, the estimated performance was well agreed with its experimental test result in all operating ranges.

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