ANALYSIS OF A WIND TURBINE'S COMPOSITE BLADES USING A FINITE ELEMENT MODEL

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INTRODUCTION

Along with the development of computer aided design (CAD) tools, design, analysis and manufacturing of wind turbine blades were made very cost effective and feasible.

The main aim of this research is to present some issues concerning structural design optimization of the imported composite blade presently used with the 10 kW wind turbine developed at Technical University of Moldova (TUM) [1].

The blade for TUM wind turbine was bought on the market for a relatively good price. It is strong and rigid but has a rather large weight (30 kg).

The following criteria have taken into account in the process of optimal blade design: minimize blade weight, does not exceed allowable stresses, minimize blade vibration and obtaining its modal frequency out of resonance. Blade mass and cost is mutually dependent and is related on the blade shell thickness. If the composite layer thickness for different blade section is at optimal level then we obtain the improvement of these parameters.

The load analysis of the blade consist of a 3D CAD model analyzed using the FE method.

For static behavior of the blade the very strong winds conditions that occur in Republic of Moldova were considered [2].

Regarding the dynamic behavior of the blade and the entire assembly of the wind turbine are imposed the following conditions: 1) the natural frequencies of the blade at 8 m/s wind speed must be above the ~2.5 Hz frequency of the turbine rotor (130 rpm) and 2) the natural frequency of the blade should be separated from the harmonic vibration of the tower (~1.16 Hz estimated first mod) [11].

Finite Element Model

Blade Geometry. The blade has a length of 3,9m and was designed in Solid Works 2010, Fig. 2 according to [3, 4, 5] and then aerodynamic design optimization was performed using ANSYS CFX module by another member of our research team.

The blade was meshed entirely with 7539 layered shell elements and 7697 nodes in ANSYS Workbench. ANSYS Composite PrepPost (ACP) was used as a preprocessor for composite layups modeling as well as for post processing to check the stresses and the failure criteria that occur in the composite layers Fig. 1. Structural design of the blade and layer schedule is shown in Table 1.

Material Parameters. Material parameters for principal directions in the fiberglass lay-up listed in Table 2 were derived from experimental data for similar fabrics [6, 7, 8] and from the ANSYS

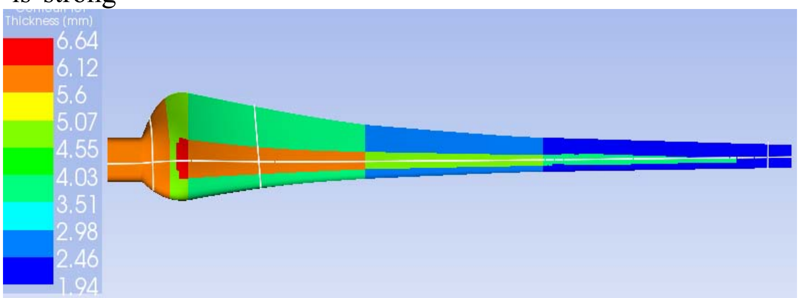


Figure 1. The composite layer thickness for different blade

Workbench database. The UD 600 (unidirectional fabric and 600 g/m^2 fiber weight) and WR 300 (woven roving and 300 g/m^2 fiber weight) lamina use E-glass fabrics that are embedded in low cost polyester resin. The parameters listed in Table 2 are for the material with 52% fiber volume. Total weight of the blade was obtained of 24 kg for material density 1850 kg/m^3.

Static Behavior

Blade Loads. In Republic of Moldova there are from 5 to 50 days per year with strong winds (15 m/s and more), depending by the relief. Wind intensification up to 25 m/s and more takes place relatively seldom, on average 1-2 times per year [2]. Wind turbine is orientated outside of the wind flow at speeds above 15 m/s by a special system.

For safety reasons wind turbine was analyzed at wind speed of 20 m/s. The aerodynamic loads were determined using ANSYS CFX module and were transferred to the static analysis module for structural analysis. For the wind speed specified above, the axial thrust becomes ≈ 4.5 kN and tangential forces were obtained ten times lower.

Table 1. Layer schedule for the blade.

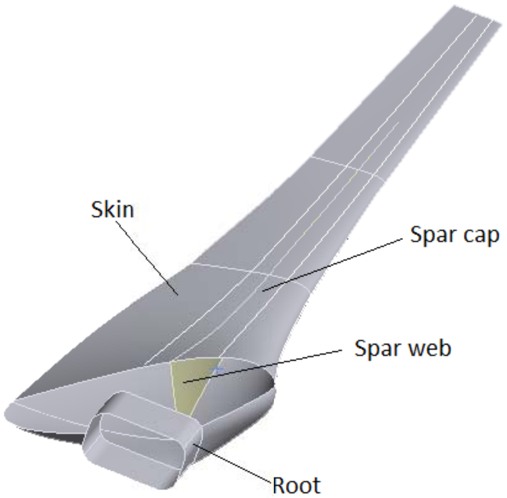


Figure 2. Components in layer schedule.

Component	Radius [mm]	Layer schedule	Thickness [mm]
Root	200 - 400	[±45/0$_2$/±45$_7$]s	4,5
	400 - 750	[±45/0$_2$/±45$_6$]s	4,25
Spar cap	750-2500	[±45/0$_6$/]s	3,76
	2500 - 3500	[±45/0$_5$/]s	3,2
	3500 - 4000	[±45/0$_2$/]s	2,2
Spar web	750 - 4000	[±45/02/±45$_3$]s	3,9
Skin	750-2500	[±45$_{14}$]	3,5
	2500 - 3500	[±45$_{10}$]	2,5
	3500 - 4000	[±45$_8$]	2

Table 2. Designed ply material properties necessary as input data in ANSYS Workbench.

Material parameters		E_X [GPa]	E_Y [GPa]	Ez [GPa]	υ_{xy}	υ_{yz}	υ_{xz}	G_{xy} [GPa]	G_{yz} [GPa]	G_{xz} [GPa]	*UTS-L** [MPa]	*UCS-L** [MPa]	Thickness [mm]
Lay-up Material	*UD600[0]$_2$*	40	15.9	15.9	0.29	0.29	0.29	4.7	3.5	4.7	629	-530	1
	WR300[±45]$_4$	15	15	8	0.3	0.3	0.3	4.7	2.7	2.7	144	-215	1

*UTS-L, UCS-L - Ultimate longitudinal tensile and compressive strength.

Also, the aerodynamic forces were calculated as described in [4, 5] using MathCAD software and ≈ 4.7 kN axial load was obtained. For wind turbines with the rotor under ten meters in diameter the gravitational and centrifugal loads are negligible [3, 4].

Flapwise and Edgewise Rigidity. With bending, the blade can fail by either of two processes:

1) material failure due to excessive stresses and/or strains or,

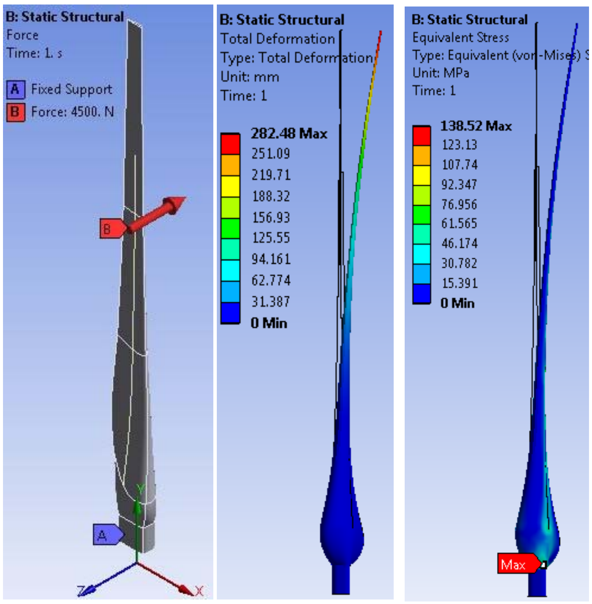
2) geometric instability, otherwise known as buckling. Also, if the blade is not sufficiently rigid in the flapwise direction it may be striking the tower and destroy itself.

To evaluate flapwise rigidity of the blade, this was constrained at the root end surface by fixing all six degrees of freedom and an axial force of 4.5 kN was applied on the blade surface as indicated in Fig. 3 *a*. The resultant deflection profile is illustrated in Fig. 3 *b*, from which it can be seen that the peak tip deflection is 282 mm (the distance from the blade tip to the tower is 480 mm) and the maximum compressive stress is 138 MPa, Fig. 3 *c*.

Bending moment in the edgewise direction is a result of blade mass and gravity which are negligible in our case. For a fully loaded 10 kW generator the maximum torque on the shaft is ≈ 600 Nm. However, to verify the edgewise bending stiffness of the blade, a tangential force of 2 kN was applied to the leading edge of the blade as shown in Fig. 4 *a*. The corresponding deformed geometry and equivalent stress is displayed in Fig. 4, *b* and *c*. The tip displacement is 59 mm and the maximum compressive stress is ≈ 97 MPa

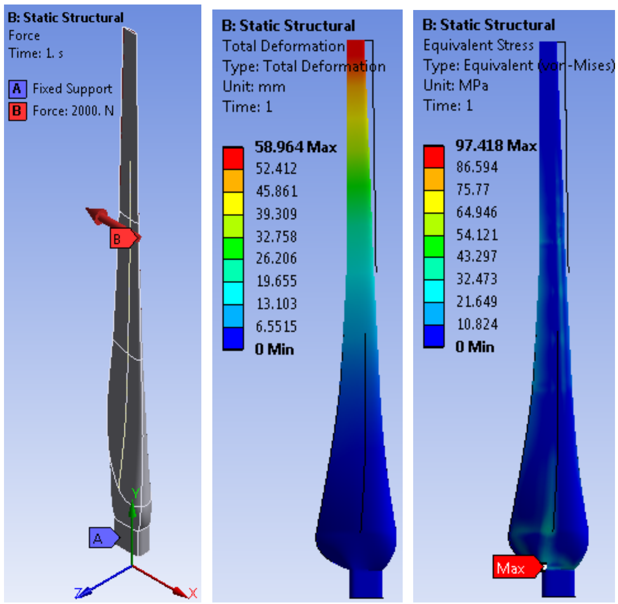
Dynamic Behavior

Harmonic Modes. To estimate the mode shapes and natural frequencies of the blade, a modal analysis was conducted in ANSYS Workbench. For a stopped rotor, the fundamental flapwise and edgewise vibrational modes occurred at frequencies of 7.5 and 15.23 Hz, respectively.



a *b* *c*

Figure 3. Flapwise bending: *a - axial loading, b – total deformation, c - equivalent stress.*



a *b* *c*

Figure 4. Edgewise bending: *a - tangential loading, b – total deformation, c - equivalent stress.*

Because rated speed of the rotor is 130 rpm at 8 m/s wind speed the blade was subjected to these prestressed inertial load. For the spinning blade model the fundamental flapwise and edgewise vibrational modes occurred at frequencies of 7.9 and 15.37 Hz, respectively.

The frequencies presented in Table 3 compare well with frequencies reported for the presently used blade.

Table 3. Natural frequencies for mode shapes.

Mode shape	Frequency [Hz]			
	New design blade		*Currently used blade*	
	Spinning rotor	*Stopped rotor*	*Spinning rotor*	*Stopped rotor*
1st mode, flap-wise	7.9	7.5	8.1	7.6
2nd mode, edge-wise	15.37	15.23	15.7	15.6

The results were validated, in an approximate manner, using a cantilevered beam model [9, 10]. The first natural frequency (f_1) for a prismatic beam was modeled by

$$f_1 = \frac{1.875^2}{2\pi \cdot l^2}\sqrt{\frac{EI}{m}} \ , \qquad (1)$$

where l is the beam length (m), m - mass per unit length (kg/m) and EI - flexural rigidity (N·m^2).

For the mean values of the flexural rigidity and of the mass per unit length - $f_1 \approx 5.7$ Hz.

FE results are reasonable given the approximate nature of the beam model for the non-prismatic blade geometry.

Periodic Excitations. Sources of periodic excitations for a wind turbine blade are the following:

1) the constant rotational speed (130 rpm) of the turbine rotor and 2) tower vibrations.

For a turbine with a tree-bladed rotor, the aerodynamic frequency of excitation occurs at three times the rotational frequency of the rotor (3 Ω) [4, 5]. To verify possible interactions between these frequencies and the natural frequencies of the different structural components a Campbell diagram was elaborated, Fig. 5. The lines radiating from the origin represent possible excitation frequencies as the rotor spins up to its operating speed. Horizontal curves illustrate the fundamental natural frequencies for the blades and tower. Resonance is likely to occur at points where excitation frequency curves and natural frequency curves cross one another.

The bending flexibility of the tower represents the spring stiffness; the damping is given in the form of a damping coefficient.

For a tubular steel tower with a top mass the first natural frequency (Hz) can be estimated with the following expression [11]:

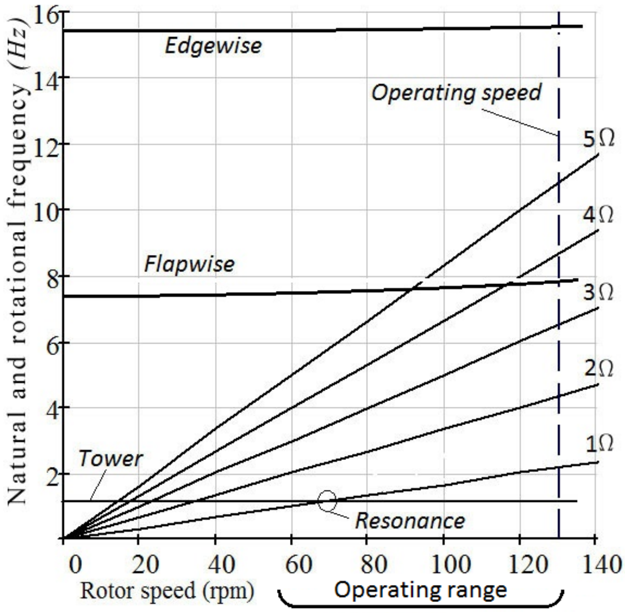


Figure 5. Campbell diagram for 10 kW wind turbine with the new designed blade [4].

$$f_1 \cong \frac{D}{L^2}\sqrt{\frac{E}{104(\frac{M}{\rho_C \cdot \pi \cdot D \cdot t \cdot L} + 0.227)\rho_C}}, \qquad (2)$$

where D is tower average diameter (m), L – tower height (m), t – tower wall thickness (m), M – top mass (rotor and generator), ρ_c – density of steel (kg/m^3), E - elastic modulus of steel (Pa).

CONCLUSIONS

In this analysis was found a compromise between optimized parameters of the blade for the TUM 10 kW wind turbine.

Blade structure optimization results are the following:

- mass reduction ≈ 20%;
- maximum tip deflection is 282 mm (the distance from the blade tip to the tower is 480 mm);
- maximum equivalent stress for 20 m/s wind is approx. 138 MPa (215 MPa – ultimate stress);
- Campbell diagram shows that the resonance between tower vibration (~1.16 Hz estimated first mod) and rotor may occur when it has approx. 70 rpm; the FE model indicates that the natural frequencies of the blade are all above the 2.5 Hz rated frequency of the turbine rotor.

To reduce stress concentrations that occur in the matrix of the composite material at the blade root, further measures will be taken. Also, to perform blade fatigue analysis, for the used composite material will be determined alternating stresses at specific cycles and tensile ultimate strength.

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