MODELING THE STRUCTURAL DYNAMICS OF CHOSEN COMPONENTS OF THE HORIZONTAL AXIS WIND TURBINE

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Abstract

As the horizontal axis wind turbines are getting larger, their dynamic behaviour is becoming more important. Dynamic analysis gives knowledge how to improve efficiency and safety also in small wind turbines. This article describes numerical models of chosen components of upwind, three-bladed wind turbine. Geometry of each component is generated separately and then assembled together by transformation matrices. Material of the blades is composite, the hub is assumed to be made of steel and material of the planet carrier is casted iron. These mentioned components are modelled by shell elements. The numerical model of the hub takes into account aerodynamic and gravity loads of blades, inertia forces due to rotation of the rotor and aerodynamic damping. The aerodynamic loads, calculated according to the modified Blade Element Momentum theory, are attached to aerodynamic centres. Wind conditions were assumed for I-class wind turbine according to Germanischer Lloyd. Stress Reserve Factors were calculated for DLC 6.1 load case according to Germanischer Lloyd, too. As a first step, numerical strength analysis with using AnSYS software was performed with maximum values of principal stresses as an output. Then, based on FEM analysis results, Stress Reserve Factors were calculated. SRF values show that analyzed hub and planet carrier have sufficient strength for extreme loads. Methodology of safety margin evaluation presented in this paper allows assessing if the object fulfils relevant standards demanding.

Keywords: HAWT, Hub, Planet Carrier, Blade Element Momentum theory, Stress Reserve Factors

1. Introduction

The most important component of the wind turbine is a rotor. Wind turbine power, rotational speed and its size depend on the rotor. Based on wind turbine configuration, other components, like gearbox, bearings, generator and tower height are selected.

For numeric calculations, three-blade horizontal axis upwind turbine with pitch regulation was taken. Blade length was 45 meters and tip speed ratio λ=6. Blade was divided for 23 parts, each one having other airfoil, thick at the root and more narrow at the tip. Numerical model contained FFA-W3-xxx and RISØ airfoils series with different aerodynamic properties and different thickness/chord ratios. The thickness/chord ratio depends on used aerodynamic profiles. Near the root of the blade there are used circular profiles with t/c ratio equal to 100%, smoothly decreasing to the 14% on the tip of the blade.

2. Loads calculation

Local coordinate systems were based on PN – IEC 61400 - 1, according to Guidelines (2002). The fixed coordinate system used for load and stress assessment of the planet carrier and the hub is shown on Fig. *1*. Coordinates system centre location is the same as hub centre. The x-axis is co-linear to main shaft axis. The x-axis has a downwind direction. The z-axis is tangent to main shaft axis in a vertical plane containing the shaft axis. The y-axis is tangent to main shaft axis and belongs to horizontal plane. The coordinate system is rotating together with the main shaft of the wind turbine.
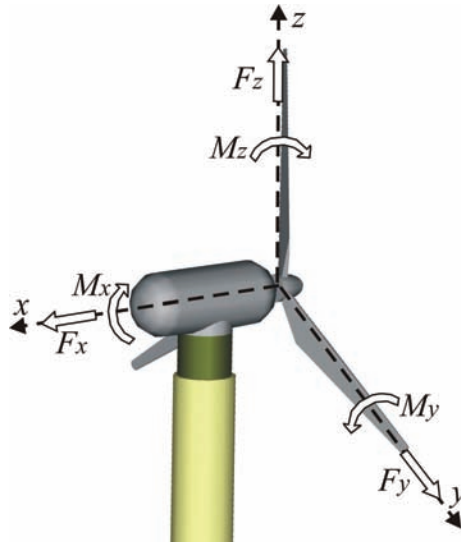
Fig. 1. Coordinate system used for loads calculation of chosen components of the wind turbine

In numerical analysis of hub and planet carrier was applied the following loads:
- *My*, flap-wise moment,
- *Mx*, edge-wise moment,
- *Mz*, moment,
- *Fx*, thrust force in main shaft direction,
- *Fy*, tangential force in rotor plane,
- *Fz*, force along the blade axis.

The most important for stress analysis of the planet carrier is torsional load *Mx*, so that load case with highest *Mx* value was taken for strength calculation.

Loads calculated for given coordinate systems and load case DLC 6.1, according to Alternative (2005), is listed in Tab. 1. The wind turbine is parked (standstill or idling), with extreme wind speed model and recurrence period 50 years.

Tab. 1. Calculated loads of planet carrier and the hub of the wind turbine

	Hub			Rotor
	Blade no 1	Blade no 2	Blade no 3	
Fx [kN]	300	-350	-150	40
Fy [kN]	-70	20	40	314
Fz [kN]	-65	80	-15	-470
Mx [kNm]	2400	-200	-370	330
My [kNm]	-7500	5700	1700	3600
Mz [kNm]	-80	20	-25	580

Planet carrier is subjected for torsion, transferred from the main shaft, to satellite axes coupled with sun and ring gears of the gearbox. In the model used for described calculation, loads were transferred through the main shaft.

3. Numerical analysis of chosen components of the wind turbine

The front end of the main shaft was attached to the ground with using displacement on a front plane. Rear end of the main shaft was attached to the planet carrier with using bonded contact. Torsion was applied to the planet axes holes (Fig. *2*).
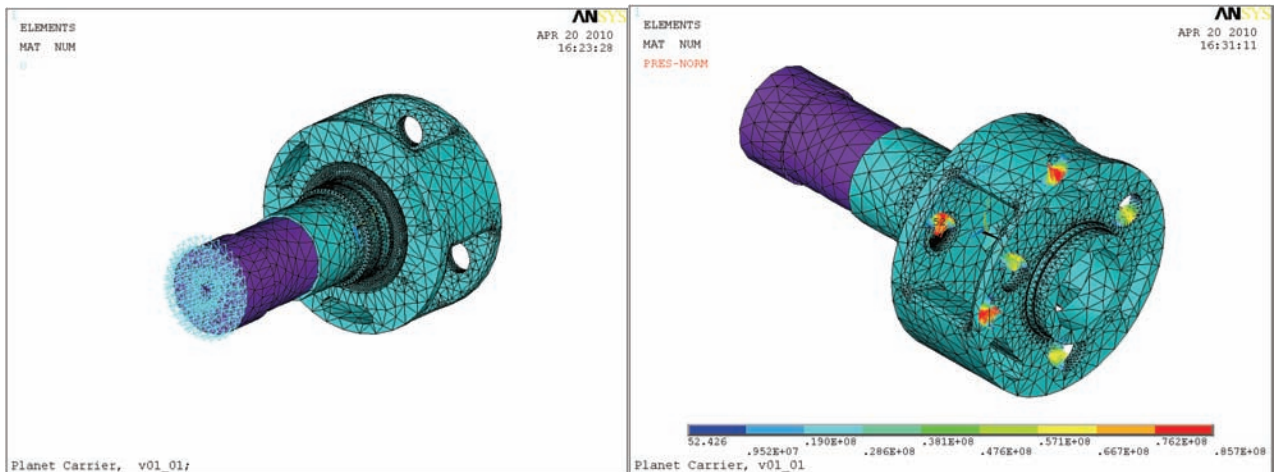
Fig. 2. Displacement and loads applied to the numerical model of the planet carrier

FEM calculation results are stress levels for each node. Stress patterns are shown on Fig. 3.
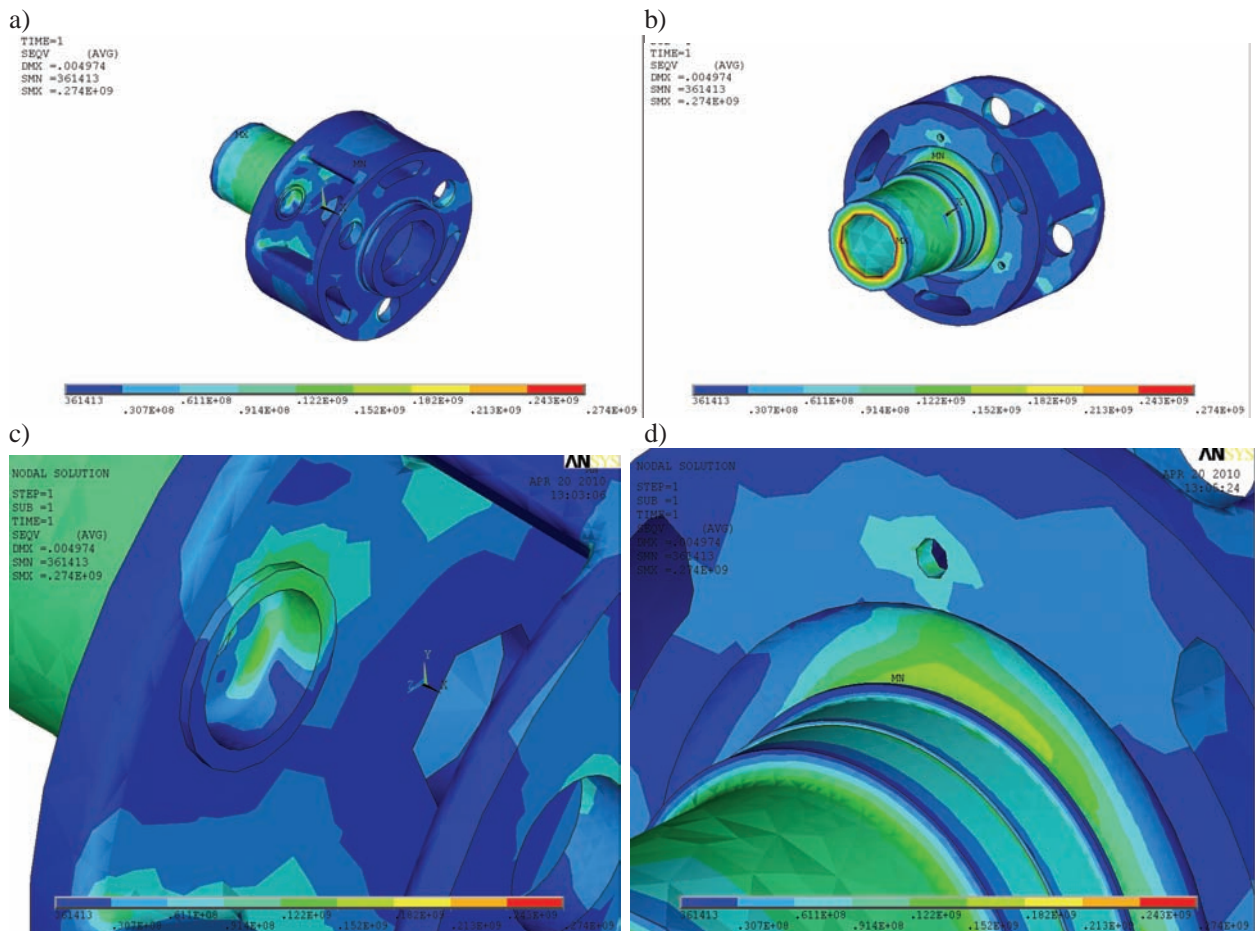

Fig. 3. Stress patterns for the planet carrier: a) front view, b) rear view, c) stress relief cut area, d) planet axes holes

The numerical model of the hub with one of load cases (*My* bending moment with loads applied with using pressure in blades bearings) is presented on Fig. 4.

The strength analysis results performed in AnSYS are stress values for all nodes. Load pattern for hub model is shown on Fig. 5.

The highest principal stress value in the hub, |σ|max=140 MPa, is on internal surface of blade 1 flange. The highest stress in the planet carrier can be seen on the main shaft contact area. The stress value is 274 MPa. Highest stress for stress relief cut is 150 MPa. The highest stress for planet axes holes is 120 MPa. Strength analysis was performed basing on FEM stress results. Strength for extreme loads was calculated.
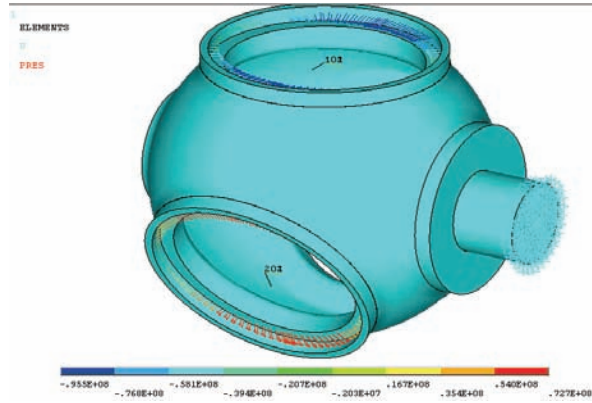

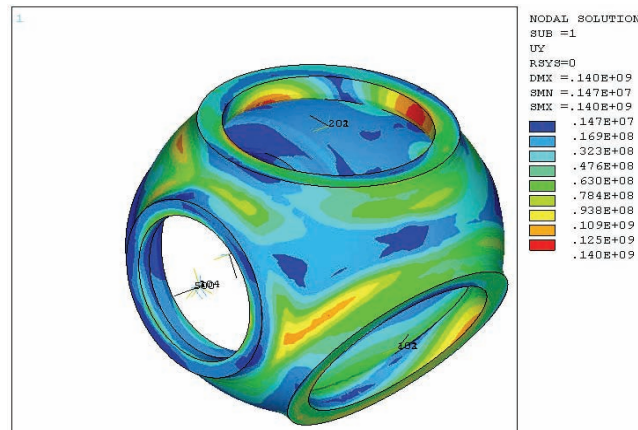*Fig. 4. Loads applied to the numerical model of the hub*

Fig. 5. Maximum principal stress distribution |σ|max for the hub

4. Calculation of Stress Reserve Factors

After FEM analysis resulting in maximum absolute values of principal stresses |σ|max, strength analysis was performed according to the Guideline (2003).

The SRF was calculated according to:

$$SRF = \frac{R_e}{\gamma_f \cdot \gamma_m \cdot \gamma_n \cdot |\sigma|_{max}} \, , \qquad (1)$$

where:

γ_f - Load factor [-],

γ_m - Material factor [-],

γ_n - Consequence of failure factor [-],

$|\sigma|_{max}$ - Absolute maximum principal stress [MPa],

R_e - yield strength for planet carrier material EN-GJS-700-2U: Re = 380 [MPa],

R_e - yield strength for hub material EN-GJS-400-18U-LT: Re = 220 [MPa].

The lowest values of SRF in nodes with highest principal stresses values |σ|max are listed in Table 2. On Fig. 6 and Fig. 7, SRF patterns for planet carrier and hub are shown.

Tab. 2. SRF values for chosen locations

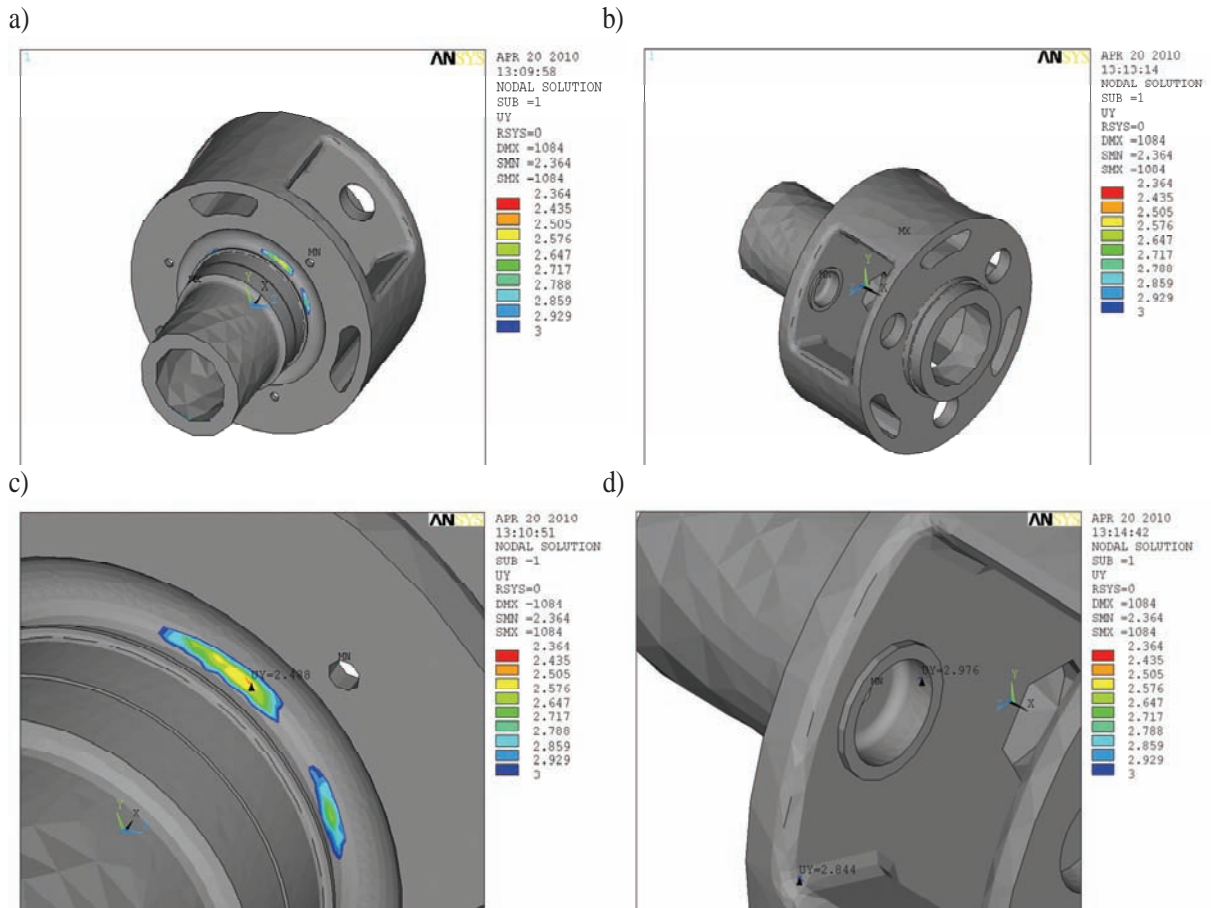|	location	SRF [-]
	Stress relief cut	2.49
Planet carrier	Planet axes holes	2.98
	Butterfly corner	2.84
hub	Fillets on hub flanges	1.061
	Outer surface of hub	3.01

Fig. 6. Stress reserve factors for chosen locations: a) front view, b) rear view, c) stress relief cut area, d) planet axes holes
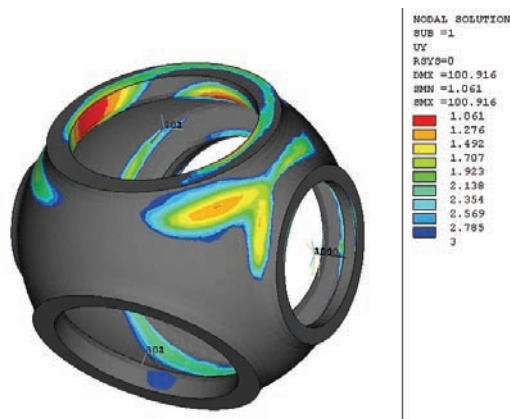

Fig. 7. Stress reserve factors distribution for the hub

5. Results analysis and conclusions

The analysis shows that for chosen components, extreme loads strength is sufficient.

Strength analysis results show that for all locations in the planet carrier, calculated SRF factors are above 2.0. That will be useful in case of increasing the wind turbine power.

In preliminary analysis the SRF value of hub was below 0.4, and modifications in geometry were needed. On edges of hub flange were applied fillets, increased hub shell thickness and redesigned hub flanges. After each modification results were reviewed and some conclusions were received. Fillets on edges of hub flanges increase SRF value up to 0.45. Increasing hub shell thickness didn't have sufficient influence on SRF value on the hub flange. The total mass of the hub highly increased, what could result in higher cost of manufacturing. That is why the total hub shell thickness was set to previous value t=0.05 m. By increasing hub flange thickness from 0.2 m up to 0.4 m the lowest SRF value increased from 0.4 to 0.92. Further modifications were made by changing the height of hub flanges from 0.25m to 0.28 m what finally resulted in the lowest SRF =1.061. SRF expresses the reserve between the occurring maximum stress and the yield strength and estimates if a single load will lead to the instant failure of wind turbine components.

References

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