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## Assessment of Gearbox Operational Loads and Reliability under High Mean Wind Speeds

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### Abstract

This paper investigates the dynamic loads occurring in the drivetrain of wind turbines with a focus on offshore applications. Herein a model of the gearbox of the 5 MW wind turbine is presented. The model is developed in a multi-body framework using commercial software MSC ADAMS. Validation of the model was based on the experimental data provided by NREL for 750 kW prototype gearbox. Failures of gearboxes caused by high dynamic loads have a significant influence on the cost of operation of wind farms. For these reasons in the study, the probability of failure of the gearbox working in an offshore wind turbine that operates in storm conditions with mean wind speeds less than 30 m/s is presented. In the study, normal shut-downs of a wind turbine in storm conditions were investigated. The analysis were conducted for two storm control strategies and different wind conditions from an extreme operating gust, normal turbulence model and extreme turbulence model. In the paper, loads in the planetary gear are quantified as well as the torsional moments in the main shaft. On the basis of simulation results the annual probability of failure of the gearbox in a wind turbine with soft storm controller is calculated, and compared with the one had the gearbox working in a wind turbine operating with hard storm controller. In the study, it was found that normal shut-downs do not have a significant influence on the ultimate loads in the gearbox, since they are related mostly to the gusts occurring during turbulence. Application of the storm controller with reduction of the wind turbine power allowed the decrease of the probability of failure for ultimate stresses.

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*Keywords:* gearbox model; extreme gearbox loads; offshore wind energy; reliability assessment; multibody; planetary gear

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## 1. Introduction

Offshore wind turbines are exposed to the extreme events such as emergency stops, wind gusts and grid losses, that often occur at mean wind speeds near 25 m/s. These events can have significant influence on the reliability of wind turbine drivetrains, especially of gearboxes. Despite the well-defined international standards, gearboxes require repairs before the expected lifetime is reached. In the lifetime of a wind turbine gearbox repairs and replacements have to be done even few times, what have significant influence on the overall energy costs [1]. In spite of the great efforts of the manufacturers to improve gearboxes design, their failures are still a main reason for downtimes of wind turbines. Nowadays, gearbox related failures are responsible for over 20% of downtimes of wind turbines [2], and they need replacement after 6–8 years which is much less than expected 20-years of failure free operation [2, 3]. Several basic problems that influence wind turbine gearboxes reliability can be listed [4]: the majority of wind turbine gearbox failures initiate in the bearings, mostly failures are not directly related to the quality issues and most of the problems are not specific to a single gear manufacturer or turbine model. To cover the risk due to the possibility of gearbox failures, wind turbine manufacturers increase significantly the sales price what increased cost of the electrical energy produced by wind farms. To reduce these prices one solution is to increase reliability of gearboxes components [5, 6]. Gearboxes mounted in offshore wind turbines are exposed to extreme loads due to emergency stops, wind gusts and grid losses that occur at high mean wind speeds. In order to reduce these loads proper storm control strategy can be applied.

The traditional controller strategy applied in wind turbines will shut-down the turbine when the mean wind speed reaches the cut-out value, which is usually 25 m/s. When the mean wind speed drops below the shut-down value, the wind turbine starts again. However, many offshore wind turbines are equipped with a storm control system so that the wind turbine does not shut-down at 25 m/s mean wind speed, instead has reduced power and operation till mean wind speeds of 30 m/s, an example of which can be the Enercon Storm Control System [7, 8].

At high mean wind speeds, significant dynamic loads occur in the gearbox components. These loads may be reduced by application of a proper storm control strategy. In this study, particular attention is paid to the loads in the planetary gear. Simulations of the wind turbine drivetrain are performed using appropriate generator control and blade pitch control algorithms. Normal shut-downs are simulated at mean wind speeds from 24 m/s to 28 m/s at different blade pitch angles and with normal turbulence, extreme turbulence and gust inputs. In the study, loads in the planetary gear are quantified and compared for other operation scenarios. On the basis of maximum loads the probability of failure of the gearbox is calculated and compared for other investigated storm control strategies.

## 2. Multibody Model of 5 MW Drivetrain

### 2.1. Reference Gearbox

In the study, the preliminary design of a 5 MW gearbox was developed by application of upscaling rules to the 750 kW prototype, of which the technical documentation, tests and field data were provided by NREL [1, 9]. The upscaling rules applicable to wind turbines were defined by Sieros et al. [10]. The authors stated that if geometrical similarity is enforced, the weight and power in a wind turbine scale according to  $m \sim s^3$  and  $P \sim s^2$  where  $s$  is the scaling factor. Scaling in such a way, moments corresponding to the aerodynamic forces follow the  $s^3$  rule, so the gearbox input torque will increase cubically. Sieros et al. presented general upscaling rules applicable to wind turbines but for upscaling of a gearbox, detailed information related to the geometry and inertia of gears, torsional stiffness of shafts, bearing stiffness and contact parameters are required. Basic upscaling rules applicable to gearboxes parameters are presented hereunder.

The radial stiffness of a roller bearing without clearance working under pure radial loads and line contact on each raceway can be defined as follows [11]

$$K_{radial} = \frac{Z^{0.9} l^{0.8} \cos^{1.9} \alpha}{2.72 \cdot 10^{-4} F_r^{0.1}} \quad (1)$$

where  $F_r$  is the bearing radial force,  $l$  is the length of rolling element,  $Z$  is the number of rolling elements and  $\alpha$  is the contact angle. Assuming that number of rolling elements and contact angle in bearings are constant, radial stiffness of roller bearing is proportional to the  $l$  with power of 0.8 and is inversely proportional to the  $F_r$  with power to 0.1. Summarizing, bearing radial stiffness will follow  $K_{radial} \sim s^{0.6}$  rule. The contact stiffness of gears pairs according to the Hertzian elastic contact theory is defined in [12, 13]. The contact stiffness with assumption that geometrical relations and material properties of gears become constant is proportional to the dimension of gears with power of 0.5, so the contact stiffness of gears will follow  $K_{contact} \sim s^{0.6}$  rule.

In the study, scaling factor  $s = 2.582$  was determined on the basis of the power relation between prototype gearbox (750 kW) and the desired design of 5 MW gearbox. In table 1 and 2 specification of 5 MW reference gearbox is presented.

Table 1. Geometrical parameters of a 5 MW reference gearbox

Parameter	1st stage	2nd stage	3rd stage
Type	Planetary	Parallel	Parallel
Ratio	5.714	3.565	4.00
Number of planets	3	-	-
Normal module [mm]	25.82	21.30	12.91
Normal pressure angle [deg.]	20	20	20
Helix angle [deg.]	7.4947	14	14
Face width [mm]	650	-	-
Centre distance [mm]	795.3	1161.9	735.8
Number of teeth, Sun gear/Pinion	21	23	22
Number of teeth, Planet gear/Gear	39	82	88
Number of teeth, Ring gear	99	-	-
Pitch diameter, Sun gear/Pinion [mm]	556.7	509	294.4
Pitch diameter, Planet gear/Gear [mm]	1033.8	1814.8	1177.4
Pitch diameter, Ring gear [mm]	2624.3	-	-

Table 2. System parameters of a 5 MW reference gearbox

Parameter	Component	Value
$I_{axis\ of\ rotation}$ [kgm <sup>2</sup> ]	Sun gear	64
	Planet gear	334
	Ring gear	14685
	Carrier	7452
	Bearing carrier upwind	3.18e9
$K_{radial}$ [N/m]	Bearing carrier downwind	2.47e9
	Bearing planet upwind	3.00e9
	Bearing planet downwind	3.00e9
	Sun spline	3.53e10
$K_{contact}$ [N/m]	Sun gear/ Planet gear Pinion/Gear	2.15e9
	Planet gear/ Ring gear	3.09e9

Designed gearbox consists of three stages: one planetary and two parallel with overall gear ratio of 81.5. In the literature, there are available other designs of 5 MW gearboxes, like one proposed by A. R. Nejad et al. [6], which consist of two planetary and one parallel helical stages. It has to be mentioned that presented design is preliminary, and it has to be normalized and verified according to the design guidelines set in the standards IEC 61400-1 and IEC/FDIS 61400-4 [14, 15].

## 2.2. Drivetrain Model

The multibody model of 5 MW wind turbine drivetrain was developed in commercial software MSC ADAMS. The model contains of main shaft, gearbox and generator controller. The torsional stiffness and damping of the main shaft was modeled by bushing connection between the shaft and carrier, and has a value of  $8.676 \cdot 10^8$  Nm/rad and  $6.215 \cdot 10^6$  Nm/(rad/s) respectively. The main bearing was modeled as revolute joint. The generator was modeled by inertia connected by a torsional spring with high speed shaft of the gearbox, and its parameters were based on the NREL 5 MW reference wind turbine specification [16]. In Fig. 1, the drivetrain model of the 5 MW wind turbine in ADAMS is presented.

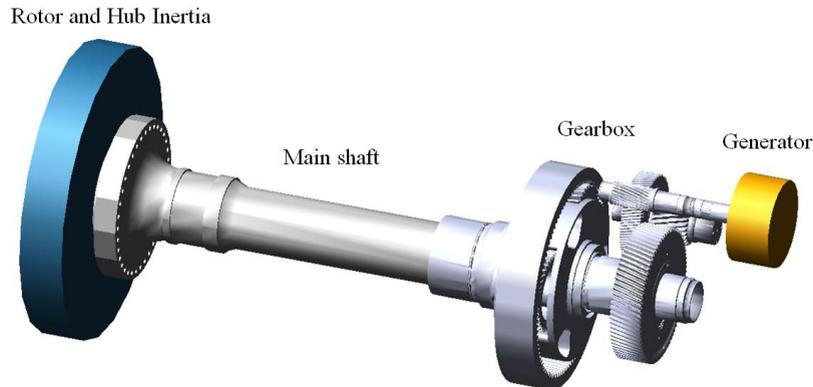


Fig. 1. Schematic diagram of the drivetrain of 5 MW wind turbine.

In the model generator controller was based on the PI algorithm with the value of proportional gain  $K_p = 1.93 \cdot 10^7$  and integral gain  $K_i = 4.33 \cdot 10^6$ .

The multibody model of the gearbox was developed with the assumption that shafts and gears are represented by rigid bodies connected by bearings which are represented by stiffness and damping, torsional stiffness of shafts was modelled by bushing connections. The interactions between gear teeth were modeled by contact algorithm [17-19]. The validation of the model was based on 750 kW prototype and experimental data provided by the National Renewable Energy Laboratory of U.S. Department of Energy.

## 3. Analysis of Gearbox Operational Loads

### 3.1. Storm Controller

In the paper, two control strategies applied to high wind speeds are analyzed. First one, Hard Storm Transition (HST) control strategy will shut-down a wind turbine when the 1-min average wind speed reaches 25 m/s and will start-up it again when the average wind speed is lower than 25 m/s. The second one, Soft Storm Transition (SST) control will decrease a wind turbine power when the 1-min average wind speed exceeds 20 m/s and will stop it completely when wind speed reaches 30 m/s [8].

Presented control scenarios have influence on the reliability of a wind turbine drivetrain. In the next sections, loads in the gearbox will be analyzed for presented storm control strategies and the annual probability of failure will be calculated.

### 3.2. Simulations Results

To analyze the loads in the wind turbine gearbox, firstly simulations were conducted in the aeroelastic code HAWC2, using NREL 5 MW reference wind turbine [16]. The aerodynamic torque and dynamic speed of the high speed shaft calculated in HAWC2 are inputs to the ADAMS model of the drivetrain, whereby the dynamic response of the drivetrain to the input time series is calculated. The multibody model allows to calculate forces in bearings, meshing as well as reaction torques in each shaft in the gearbox.

In the study, cases of normal shut-downs were simulated (DLC 4.2 from IEC 61400-1). Firstly, simulations were conducted for the wind turbine with HST control system working at mean wind speed of 25 m/s. The shut-downs were simulated using wind conditions from an Extreme Operating Gust (EOG), Normal Turbulence Model (NTM) and Extreme Turbulence Model (ETM). For NTM and ETM three seeds were analyzed. For the extreme operation gust shut-downs were realized in three points: e1 – minimum wind speed velocity, e2 – maximum wind acceleration and e3 – maximum wind velocity. For the NTM and ETM shut-down was conducted in 200 s from start-up of the wind turbine. Besides of simulations for hard storm transition the analyses for soft storm transition control strategy were conducted. For this case, the same conditions were used and the normal shut-downs were simulated at mean wind speeds 24, 26 and 28 m/s.

In this paragraph, example simulation results are presented. In Fig. 2 shut-down for extreme operating gust in the point of maximum wind acceleration for HST control is presented.

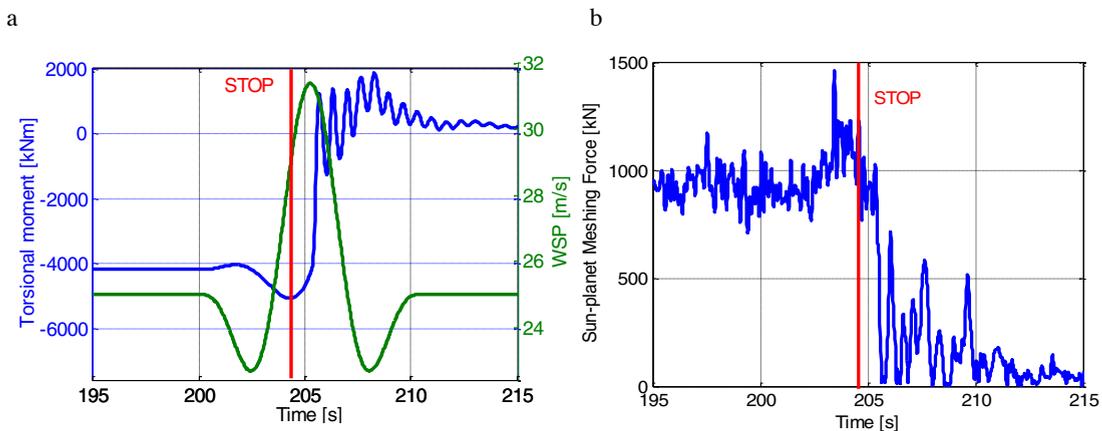


Fig. 2. HST control, EOG, 25 m/s, stop at e2) (a) main shaft torsion and wind speed (WSP); (b) meshing force.

For normal operation of the 5 MW wind turbine, the rated torque for HST control is about 4180 kNm and meshing force in the sun-planet meshing of planetary gear about 1000 kN. Analyzing the torsional moment in the shaft and meshing force from the simulation, it is noticeable that both are increasing due to the wind gust, the torsion reaches a maximum value of 5000 kNm and meshing force maximum of 1500 kN. The maximum values of the analyzed torsional moment and force were reached before the shut-down was conducted. Other simulated cases were the operation of the wind turbine under extreme turbulence and SST control, Fig.3.

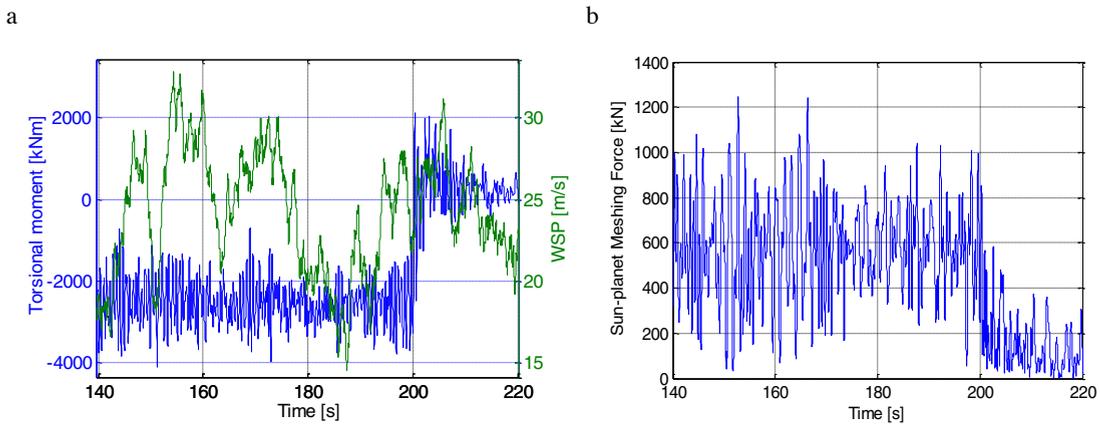


Fig. 3. SST control, ETM, 24 m/s (a) main shaft torsion and wind speed (WSP); (b) meshing force.

On the plots above simulation results for soft storm transition control and wind with extreme turbulence are presented. For this control strategy a wind turbine has reduced power so the maximum meshing force is much lower than for HST control. One can observe (Fig. 3a) that the maximum instantaneous wind speed reaches above 30 m/s, and the maximum meshing force is related to the gusts occurring during turbulence.

### 3.3. Gearbox Loads

In this section, the ultimate loads occurring in the gearbox due to normal shut-down are presented. The analysis was conducted to determine the extreme loads that occur in the sun-planet meshing. In Fig. 4 the maximum meshing forces for investigated controls and each operating scenario are presented.

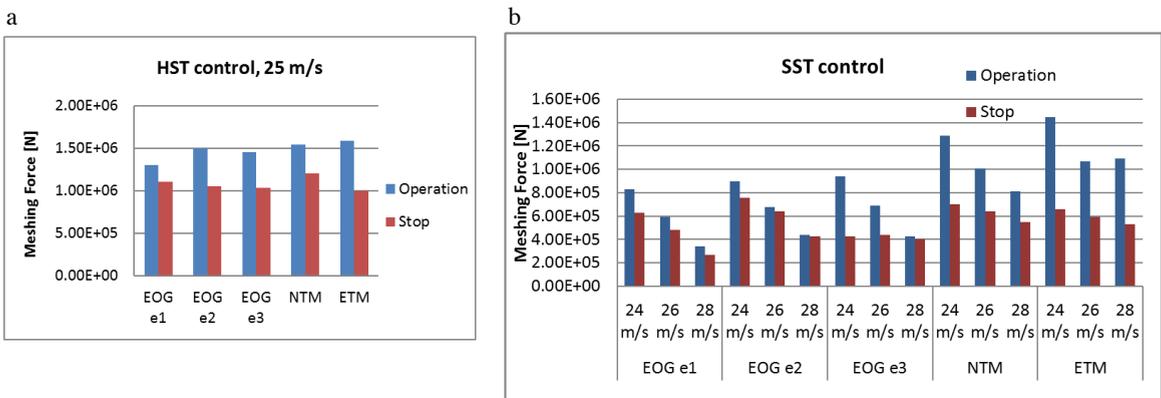


Fig. 4. Maximum loads in sun-planet meshing, (a) HST control and mean wind speed 25 m/s, (b) SST control and other mean wind speeds.

The extreme forces acting on the gear tooth in sun-planet meshing of the planetary gear were analyzed before and after shut-down was initiated. From figures above it is clearly seen that higher loads appear before the shut-down and depend on the wind gusts, as it was shown in the previous section. The maximum meshing force between sun

and planet gear for HST control occurs for ETM and is equal to  $1.6 \cdot 10^3$  kN. For SST control the maximum load in the sun-planet meshing occurs also for ETM and it is the highest for mean wind speed of 24 m/s, and reaches value  $1.45 \cdot 10^3$  kN. For each analyzed scenario, loads in the gearbox are higher before the shut-down was conducted. For the soft storm transition control strategy a wind turbine is operating in high wind speeds at reduced power, but nevertheless the extreme gear tooth loading is significant for ETM, such as at a mean wind speed of 28 m/s. Presented values of meshing forces do not include partial safety factors for loads that are used in ultimate limit state analysis.

#### 4. Reliability Analysis

In this section, reliability analysis of the gearbox operating in an offshore wind turbine with two different cases of storm control strategies is presented. The probability of failure was calculated on the basis of simulated extreme loads in the gearbox working in a wind turbine under high mean wind speeds. The reliability analysis was conducted for the sun gear tooth root bending stresses. Gear tooth root stress is defined by following expression

$$\sigma_{tooth} = \left[ \frac{F_{max}}{bm} Y_F Y_\beta Y_S \right] K_v K_{F\beta} \quad (2)$$

where  $F_{max}$  is the maximum meshing force,  $b$  is the gear face width,  $m$  is the normal module, and according to ISO 6336-6  $Y_F = 1.65$  is the factor considering influence of shape of tooth at the tooth-root stress,  $Y_\beta = 0.9$  is the helix angle factor and  $Y_S = 1.72$  is the stress correction factor,  $K_v = 1.05$  is the dynamic factor and  $K_{F\beta} = 1.15$  is the face load factor for tooth root stress [6, 20]. To determine probability of failure First-Order Reliability Method (FORM) was used. The failure function in the study was defined as follows

$$g(\mathbf{X}) = \frac{R}{X_{Resistance\ tooth}} - \sigma_{tooth} X_{Structural\ dynamics} X_{Aerodynamic} X_{Simulation} X_{Sress} \quad (3)$$

Where  $R$  is the material strength of gears, which are made from alloyed heat treated steel 42CrMo4 with yielding strength of  $700 \text{ N/mm}^2$ . The stochastic variables  $X$  representing uncertainties in the model were calculated as it is presented in [21], beside  $X_{Resistance\ tooth}$  which represents tooth root bending resistance strength and has lognormal distribution with mean value 1 and standard deviation 0.1.

The probability of failure for the root tooth bending stresses of the sun gear was calculated for two analyzed control strategies. For HST control maximum meshing force of  $1.6 \cdot 10^3$  kN was recorded, and for SST control the value of  $1.45 \cdot 10^3$  kN, both values were registered for ETM. For registered values of loading for hard storm transition control the corresponding probability of failure for the sun gear tooth ultimate stress is determined as  $P_f = 8.06 \cdot 10^{-4}$ , whereas for the soft storm transition control strategy  $P_f = 2.75 \cdot 10^{-5}$ . The probability of failure was estimated on the basis of distribution of the failure function  $g(\mathbf{X})$ . The convergence analysis due to the number of realizations of the stochastic variable vector  $\mathbf{X}$  was conducted.

#### 5. Conclusions

In the paper, analyses of extreme loads in the gearbox of a 5 MW offshore wind turbine working under high mean wind speeds have been presented. The two storm control strategies (HST and SST) and different wind conditions (EOG, NTM and ETM) have been studied. The simulations were conducted using a model of the wind turbine drivetrain developed in a multi-body framework and additionally the design of a 5 MW gearbox was presented. Proposed design of the case study gearboxes is preliminary and has to be verified according to the design guidelines contained in the standards, but with comparison to the other designs proposed in the literature it is characterized by simpler structure what can reduce costs of manufacturing and deployment.

The design load case 4.2 from IEC 61400-1 was simulated; the maximum values of torsion in the main shaft and loading in the sun-planet meshing in each operation scenario were reached before the shut-down was conducted. From this analysis, it was found that normal shut-downs do not have significant influence on the ultimate loads in the gearbox. To investigate influence of the hard storm and soft storm transition control strategies on the reliability of gearbox, the probability of failure for the root tooth bending stresses of the sun gear was calculated. For wind turbine operating with HST controller the maximum force in the sun-planet meshing occurs for ETM and it was registered at a mean wind speed of 24 m/s. In the presented simulations, the maximum values of meshing forces were related to the gusts in the turbulent wind profile, the higher mean wind speed having the stronger gust amplitudes. The application of the storm controller with reduction of the wind turbine power allowed a reduction in the probability of failure calculated for the sun gear tooth ultimate stress from the value  $8.06 \cdot 10^{-4}$  to value of  $2.75 \cdot 10^{-5}$ .

In the future study, design of 5 MW gearbox will be improved according to IEC/FDIS 61400-4 and the reliability of all gearbox components will be calculated according to IEC 61400-1.

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