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Validation of the gearbox load calculation of a wind turbine MBS model

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Abstract. Due to the increasing cost pressure, a reliable wind turbine (WT) is a major priority. Reliability is strongly determined by the design process. Therefore, it is required to have accurate and reliable load assumptions, ideally at early stages of the design process. This contribution discusses suitable simulation methods and then presents the validation of the load calculation of a full scale multi-body simulation (MBS) wind turbine model. The modelled wind turbine is a 2.75 MW generic research wind turbine. The aim of the MBS model is to produce load time series of components’ internal loads and states. These results are used as input for more detailed component models e.g. finite element (FE) tooth contact or bearing models. Subsequently these are then used to calculate detailed bearing and tooth loads. The MBS is compared with comprehensive measurements from a full scale 4 MW system test bench. The main focus will be on the validation of the MBS models internal components loads, displacements and deformations on which the bearing and tooth loads depend on. It is evaluated which model parameters influence these calculation objectives as well as the modelling methods and fidelity required to represent them meaningfully. It is evaluated that model and test indicated comparable results.

1. Motivation

A WT is a very complex system consisting of a lot of and often very flexible components such as rotor blades. These components interact with each other and thereby represent a system very susceptible to vibration. Due to the high dynamic loads in all six degrees of freedom (DOF) that a rotor couples into the drive system and the influence of the electric system and controller, load calculation for WT and their components is challenging. This is assumed to lead to higher than expected failure rates of wind turbines which directly impact the cost of energy [1]. Thus, to improve the design process and the reliability of WTs and their components, simulation models are needed that can robustly calculate these loads while taking above mentioned effects and system wide reciprocal influences into account. Therefore, entire system models are developed and validated at the Center for Wind Power Drives (CWD) [2]. For this kind of validation, system test benches for wind turbine nacelles present an efficient tool. It is possible to test system behaviour and measure loads under controlled conditions repeatedly and reproducible. At the CWD, a 4 MW system test bench is operated. The device under test (DUT) that is currently investigated is the “FVA-Gondel”, a state of the art generic 2.7 MW WT nacelle. A Hardware-in-the-loop (HIL) system is used to calculate aerodynamic loads and emulate inertia, dynamic behaviour of the rotor as well as grid loads to account for realistic system behaviour [3] [4] [5].

With respect to the levelized cost of energy the gearbox is one of the most critical components, due to long downtimes and expensive repairs in case of failure [6]. Literature suggests that some of the most common or costly failures of gears and bearings result from certain load states not taken into
consideration during the design process [1][7]. This contribution focuses on the dynamic load state time series calculation of the gearbox and its subcomponents, i.e. gears and bearings, in an entire system simulation under realistic operation conditions. To maintain a meaningful scope, only the validation of the first gear stage, a planetary gear stage, is considered in this paper.

2. Outline

Some of the most crucial design relevant loads of the critical gearbox components are the tooth flank pressure distributions and tooth root strains as well as bearing pressure distributions. These depend on local microgeometry and contact conditions as well as global deformations caused by the above discussed high loads in addition to complex system wide interdependencies. One single model is not feasible to implement the necessary level of detail to calculate local pressure distributions while simultaneously considering the system behavior of relevant subcomponents. Therefore, an entire system model of the turbine on the test bench with a reduced fidelity is set up by the authors with the aim to calculate dynamic load time series [8]. These are then used as input for more detailed local component models such as FE-tooth contact or FE-bearing models, in order to calculate the actual pressure distributions of the machine elements. The focus of the paper is on the validation of the entire system model, more specifically its MBS model of the drive train gearbox, and its ability to calculate suited and reliable load time series for the local component models. The load time series states that are relevant for the local component models are evaluated, as well validated by comparison with suitable measurement parameters. The relevant model parameters are identified and the best modelling methods and level of detail are discussed.

The following section gives an overview on the state of the art of wind turbine drive train load calculation and design. The presented work is put into perspective of current international research with respect to these topics. Then the setup of the models is discussed in section 4. In section 5 the strategy for the validation is explained. The consequently devised measurement setup and campaigns are presented. The validation results and an in-depth discussion of the deviated model strategies and modelling method improvements are presented in section 6. The paper then closes with a conclusion and an outlook to future research.

3. State of the art

The current state of the art of turbine design mostly consists of modification and redesign of existing turbines and subsequent field testing. Furthermore, design deficiencies are approached by re-engineering deficient components, replacing them in the turbine and testing them in the field. This process is costly and while it leads to a better design, turbine reliability is not improved in a fast matter. During the preliminary design as well as during the detailed component design, simulation models are used for load calculation. However, typical state of the art model fidelity as advised by the IEC [9] or GL [10] for models for the design or certification process are rather low. The authors believe that with more reliable and more detailed load calculation tools, the design process and the reliability of turbines can be improved significantly. This is a problem the industry begins to acknowledge, recognizable by the content of the last guidelines and recommendations of the DNV GL [11] or current research of Nalliboyana1 et al. (Winergy) [12].

Consequently, the international research community has taken on the task of researching load states in gearboxes and developing improved drive train simulation methods. There is a lot of research being conducted with respect to more detailed MBS models of WT drive trains [13] or the setup of system test benches for gearbox reliability research and model validation [14][15]. Nonetheless the Gearbox Reliability Collaborative (GRC) of NREL is the only other contribution that combines systematic analysis of load states and load calculation in a wind turbine gearbox under realistic reproducible torque and non-torque loads combined [1]. GRC was able to show the importance of non-torque loads on gear tooth load distribution, importance of model fidelity (e.g. representation of the bearing clearance and assembly variations) and representation of the flexibility of the planet carrier and gearbox housing [1]. Those findings are based on a 750 kW DUT. Current state of the art on-shore turbines range closer to
the 2.75 MW “FVA-Gondel”. Furthermore, NREL could not apply all five DOF of non-torque loads independently to determine isolated influences of the non-torque loads and had no HIL system.

4. Modelling
In this section, the setup of the models used in this contribution is presented. First, the FE models are discussed that are used to calculate the local loads like pressure distribution. Especially the inputs that are required for an accurate load calculation are evaluated. This is done because the proper setup and validation of the entire system simulation that calculates these inputs depends on these models. Then the entire system simulation with a focus on the MBS model, more specifically the gearbox model, is presented.

4.1. Setup of the local component model
FE models to calculate the tooth flank pressure distribution and tooth root stress distribution of the sun and ring gear mesh as well as models to calculate the pressure distribution on the rolling bodies of the planet bearings are set up. To maintain a reasonable scope in this paper, only the model for the sun to planet mesh and its calculation and validation results are discussed in-depth.

A model of the sun to planet mesh is set up in the finite element software “FE-Stirnradkette” (STIRAK) from WZL, RWTH Aachen University [16]. Tooth macro and micro geometry such as flank modifications are defined in the model. Further, the model needs the overall loads of the mesh, thus the transmitted torque, as well as the deviation and declination error of axis.

4.2. Setup of the entire system simulation

4.2.1. Co-Simulation. An entire system simulation of the “FVA-Gondel” on the test bench is set up as a co-simulation as shown in Figure 1. The mechanical MBS model is connected to electromagnetic transient (EMT) models of the generator, converter and grid. This “turbine model” is then either operated in manual operation or in HIL operation mode. In Manual mode (Figure 1 left), the user gives reference values for torque, non-torque loads and rotational speed as an input into the simulation. In HIL mode (Figure 1 right), the turbine is connected to the HIL rotor model and turbine controller and the user operates the system by providing a wind field. The “turbine model” then operates autonomously as in the field. Both operation modes can be used on the test bench as well. In this contribution they are used to validate the MBS model step by step.

4.2.2. MBS model. A MBS model of the whole turbine including the test bench in six DOF is established. In Figure 2 a detailed representation of the model topology is shown. FE-models of all shafts, planet carrier, support and connection structures and housings are set up to calculate the stiffness characteristics. Based on those, the models of the components showing relevant deformation are then modally decomposed and integrated into the MBS model as flexible bodies. All relevant connecting elements such as rubber bushings or couplings are modelled as force elements with stiffness characteristics and damping based on manufacturers’ data, FE-models or measurements. The stiffness
characteristics of the bearings are represented by characteristic curves in all directions including non-linearity and clearance. The non-linear stiffness characteristics themselves are based on manufacturers’ data. The operational clearances of the bearings have been determined with the test bench by applying different non-torque loads while idling. In a first step, all the gears are modelled using a force element that calculates the tooth meshing forces analytically based on DIN 3990.

![Figure 2](image-url) Model topology of the MBS model with special focus on the gear box and the planetary gear stage

5. Validation strategy and measurements

As discussed in section 4.1., the target figures to be calculated with the MBS model are the overall loads of the mesh as well as the deviation and declination error of axis. These are measured as accurate as possible and compared to the MBS model, in order to validate its ability to calculate suitable load time series as input for the local component models.

The good accessibility and the reproducibility of the test bench allow for an easy measurement of these states. Loads such as tooth and bearing loads are measured by strain gauges in the inner ring of the planet bearings, the sun and the ring gear. Due to the relative movement of the gears in the planetary gear stage, the desired deviation and declination error of axis between planet and sun as well as planet and ring gear cannot be measured directly. However, they can be derived from measurements of the relative displacements between the planetary gear stage components as shown in Figure 3 and Figure 4 [17]. One can calculate the displacement between the planet and the ring gear by measuring the displacement or rather the error of axis between planet and sun as well as planet and ring gear. These are not measured in detail and therefore their influence cannot be validated quantitatively. Nevertheless, a brief discussion is conducted with a qualitative estimation of the influence of the deformation on the error of axis between the gears.

Additionally, the displacement of the main shaft relative to its housing plus the displacement of the gearbox housing relative to the machine carrier are measured to better understand the overall deformation situation and validate the MBS model.

Comprehensive measurement campaigns have been conducted to measure the discussed states in static as well as dynamic and realistic load conditions. At first, measurements to determine the isolated influences of loads in single DOF onto the internal load states of the gearbox have been conducted. This have been done for static and dynamic load variations. Therefore, static loads have been applied to the DUT while stepwise varying the loads in one single DOF. Furthermore, dynamic steps and sinus shapes have
been applied in one DOF whilst the others have been kept constant. Finally, the DUT has been operated under realistic loads in the HIL operation mode, to gain realistic load time series of the internal gearbox load states and thus realistic local components loads of the gears and bearings. At the start of every measurement, the turbine has been operated in a reference point without any loads.

Figure 3 positions and orientations of the distance sensors

Figure 4 used coordinate systems and positions of the distance sensors

6. Results

6.1. Validation of the influence of static non-torque loads of the gearbox load states

Of the five existing non-torque DOF, the two lateral forces are not considered in this paper. While they certainly exist, especially at extreme yaw offsets or similar scenarios, they are less dynamic and have been found to have a negligible impact on gearbox loads [17]. In addition, the thrust is not considered, as most of it is absorbed by the main bearing.

Static loads have been applied onto the DUT in the entire system simulation under the same condition as on the live test bench. The calculation results are then compared to the measurements. For each load case, loads of four DOF are kept constant i.e. zero, except for a lateral force of 488 kN representing the rotor weight, while one load is varied gradually. For each degree of freedom, the stepwise variation is repeated for three different torques as seen in Figure 5, while the rotational speed is kept at 11 rpm. As in the measurements, a reference point without any loads is simulated at the start of every simulation, which is used to level simulation and measurement states.

Figure 5 applied torque

6.1.1. Validation of the influence of the tilt bending moment $M_y$. Mainly due to the wind turbulence as well as periodic effects such as tower shadow and wind shear, there are high and dynamic bending moments, especially in tilt-direction. It was shown in previous work, that $M_y$ has a high influence on the loads of the planetary gear stage [17].
Figure 6 applied tilt bending $M_y$ and resulting displacement of the planet carrier.

Due to an eccentricity of the drive train on the test bench, an oscillation once per revolution can be observed in the bending moment and consequently in all measurements. Also the model shows a high oscillation with the tooth meshing frequency at higher torques. This is due to an overestimation of the tooth excitation. Both effects occur in all shown load cases, but will not be discussed in this paper.

The developments of values can be explained in Figure 8. Without significant loads the planet carrier is tilted forward in relation to the gearbox housing. I.e. the load zone of the rotor side bearing is at the bottom and the load zone of the generator side bearing is at the top. This is due to the weight of the gearbox which causes it to tilt backwards. High negative tilt bending causes the main shaft to bend and consequently the planet carrier to tilt upwards. This leads to both bearings having their load zone on the top. The development of values of the planet tilt shows a compensating movement as visualized in Figure 8. Especially the tangential tilt shows this behavior, which is the more relevant tilt direction for the gear mesh. Due to the on the planet bolt fixed coordinate system, the planet tangential tilt oscillates once per rotation. While the magnitude of the carrier tilt is around $0.03^\circ$ both in the model and in the measurements, the compensation of the planet tilt is at an amplitude of around $0.02^\circ$. This leaves an axial error of around $0.01^\circ$. Furthermore one can clearly observe a severe reduction of the compensating planet movement at high negative bending moment, since the tilt of the carrier is very small at this point, as suggested by Figure 8. This matches the previous findings of load analysis of the bearing load distribution due to non-torque loads [17].

However, a mismatch between simulation and measurement can be observed regarding the influence of the torque on the planet tilt in Figure 7. The authors suspect that this is due to the deformation of the planet carrier, which cannot be properly evaluated between measurement and simulation at this point.
6.1.2. **Validation of the influence of the yaw bending moment** $M_z$. While yaw bending moments do not occur as periodically as tilt bending, they can have high values and can be very dynamic due to the turbulence of the wind.

![Figure 9](image1.png) applied yaw bending moment and resulting displacement of the planet carrier

![Figure 10](image2.png) resulting tilt of the planet carrier

![Figure 11](image3.png) displacement of the planet due to yaw bending moment

Figure 9 shows the applied yaw bending moment well as the resulting displacements of the planet carrier. The oscillations due to the overestimated excitation of the tooth meshing at higher torques (see Figure 5) are even higher as in the previous section. Yet, as mentioned before, this effect will not be discussed in this contribution. Apart from that, one can observe the same behavior in the measurements as in the simulation. The planet carrier roughly stays in its non-torque tilt position. Due to the yaw bending it then yaws around the $z$-axis, leading to a displacement of the front part of the carrier and a corresponding downwards displacement due to the bearing and its clearance. The measured and simulated tilt of the carrier show a similar corresponding behavior. Yet, the measurement of the yaw of the carrier due to the yaw bending is way larger than that calculated by the model. The authors suspect that this is due to the high deformation of the gearbox torque arms due to high yaw bending, where the displacement sensors used to determine these states are located. Furthermore, an actual yaw of the carrier as the measurements suggest would lead to bearing forces of well over 2 MN. This cannot be matched with any plausible equilibrium for the main shaft.

In comparison with the tilt bending moments, where the tilt of the carrier varies in a range of $6 \cdot 10^{-6}$ rad, the tilt of the carrier is lower for yaw bending moments, with a range of $2 \cdot 10^{-6}$ rad. Even when considering that the yaw bending was only applied up to 1000 kNm, this explains why yaw bending moments do not affect the load distribution in the gear stage as heavily as tilt bending moments [17], and shows that the model can represent this behavior.
6.2. Validation of the influence of dynamic non-torque loads of the gearbox load states

6.2.1. Validation of the influence of a dynamic tilt bending \( M_y \). Due to the high dynamics in the wind loads of a WT, the capability of the model to calculate the evaluated states will be considered in this section. The tilt bending moment will be looked at, since it previously showed the highest influence on the relevant load states, and frequencies up to 2 Hz will be used. This corresponds to usual wind frequencies as well as the range of periodical excitations, i.e. 0.9 Hz for the 3-per-revolution frequency at nominal speed. Therefore, a constant torque of 1500 kNm, a force due to the rotor weight of 488 kN and a rotational speed of 15 rpm is applied. The tilt bending moment \( M_y \) is then varied as seen in Figure 12. Due to the time delay of the hydraulic actuators, which are represented in the model, a decrease of amplitude can be seen at higher frequencies both in simulation and measurement.

![Figure 12 applied tilt bending moment and resulting displacement of the planet carrier](image)

Figure 12 applied tilt bending moment and resulting displacement of the planet carrier

![Figure 13 resulting displacement of the planet](image)

Figure 13 resulting displacement of the planet

As seen in Figure 12, the characteristic of the development of values of the simulation matches those of the measurements. For both the planet carrier displacements as well as the planet displacement. Even the influence of a tilt bending moment peak for a specific planet rotational position, thus a reduced carrier tilt and therefore a reduced planet oscillation, can be seen in Figure 13.

6.3. Validation of resulting pressure distribution

As explained previously, the relevant planet carrier states are used to calculate the error of axis of the planet sun mesh and fed into the STIRAK model. The calculated tooth root stresses of the sun are compared to measurement values for two different load cases in Figure 14 and Figure 15.

![Figure 14 simulated (left) and measured (right) sun tooth root stress for \( M_y = -1500 \) kNm and 500 kNm torque averaged over several revolutions (only the tooth in the mesh is considered)](image)
Although the simulation results show some deviations, the overall distributions show a similar pattern. As mentioned above, only the carrier states are used for this simulation. Therefore, a match as seen is only achieved at high negative tilt bending moments. This is the case since under such loads, the planet does not tilt very much as discussed in section 6.1.1. This indicates, that for a better match for all possible load states, the planet states have to be considered in the calculation as well.

7. Conclusions
From the finding in this contribution, several main conclusions can be drawn.

First of all, the usefulness of a system test bench to determine and understand the load states and the impact of external loads on WT drive trains could be shown. By being able to vary each non-torque load separately and applying it onto a full drive train, the reaction of the WT, and in this case especially the planetary gear state, can be measured and evaluated.

Secondly, some knowledge of the load states inside the gearbox due to non-torque loads could be gained. Especially the influence of bending moments on the components displacements and the part the gearbox weight plays in this. With this knowledge and the model validated with these measurements, design enhancements can be derived, evaluated with the model and validated on the test bench. This will be part of future work of the authors.

Lastly some finding regarding the modelling methods to build suitable WT drive train MBS models could be shown. The findings of the GRC regarding the importance of bearing clearance and the flexibility of the planet carrier and gearbox housing [1] could be shown. Furthermore, especially the importance to model the planets with 6 DOF could be shown.

8. Summary and outlook
To improve the reliability of wind turbine gearboxes and subsequently avoid excessive cost due to failures based on design deficiencies, more sophisticated models which take system wide dynamic interdependencies into consideration are necessary. Therefore, a MBS model of a wind turbine including a gearbox model is set up as a part of an entire system simulation and presented in this paper. The aim was to validate this models ability to calculate load states for detailed load calculation for gears and bearings. This was done by comparison with measurement results from a 4 MW system test bench.

The usefulness of the system test bench to evaluate WT drive train load states could be shown and some insight on those states for a 3 point suspension drive train concept could be gained and was shown in the paper. Furthermore, the test bench measurements were used to validate the gearbox model. Consequently, the concept to calculate detailed pressure distributions using the authors’ simulation framework of an entire system simulation with a gearbox MBS model in combination with a FE-tooth contact model could be shown.

On the other hand some deviations were observed and should be looked into in the future. These are mainly a proper quantitative evaluation of the influence of the flexibility of components such as the gearbox housing and the planet carrier. In that regard the ring gear body flexibility will be looked at as well, which is modelled rigid in the current approach. Additionally, measurements previously conducted by the authors show an influence of the planet deformation on the bearing rolling body pressure
distributions [17]. To achieve a proper match between measured planet bearings rolling body loads and the ones calculated by the models, this will be included in the model in future work.

Also the authors intend to calculate accumulated load spectrums with the models and compare them to those of HIL measurements to further validate the model framework.

In a first shot the level of detail for the entire system simulation and especially the MBS model was chosen rather high. This leads to high simulation times of up to 10 hours of CPU time for 5 minutes of model time. To achieve reasonable simulation times, which is especially important to make these methods useable for the industry, future work will involve finding meaningful model reductions.

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