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Experimental and Model-based Analysis of the Force Transmission in a Rotor Bearing Support System

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Abstract. Due to complex interactions, the mechanical components of a rotor bearing support system cannot be designed and evaluated independently. Only the combination of all elastic components determines how external loads are transmitted into the tower. Simulation models attempting to answer this question only become reliable by using validated modelling techniques. The parallel investigation of a rotor bearing support system on a system test bench and in a simulation model enables a detailed analysis of the mechanical interactions between the main bearing, the gearbox and the structural components. Significant nonlinear effects, that can be observed on the test bench and should therefore be considered in the model, are the clearance and stiffness-characteristics of the main bearing, the planet carrier bearings and the clamping bushings in the torque arms. The assembly process of the drivetrain and gravity loads also have an influence on component movement and force transmission under external loads. The analysis of the mechanical phenomena with the help of simulation promotes system understanding and thus the possibility of further development and improvement.

1. Introduction and motivation

When comparing the drivetrain configurations of today's wind turbines on the market it is obvious that no rotor bearing support concept has become clearly established. The solutions for geared turbines provided by the manufacturers range from the classical three- and four-point suspension to single bearing concepts and preloaded bearing units with tapered roller bearings. In some concepts even a coupling is included at the low speed side. Each concept has its own advantages and disadvantages regarding costs, serviceability, tolerance to misalignment, preload requirements and non-torque loads acting on the gearbox. Therefore it is not surprising that the failure rates of main bearings found in statistics vary widely [1,2]. Among others, they depend on the wind turbine model, the site conditions and the production quality of the bearing.

Recommended technical upgrades for existing rotor support concepts, e.g. asymmetric bearings or tighter clearance for a higher axial stiffness, are available, but the interaction effects on other drivetrain components, especially the gearbox, are often unknown or not validated [3].

Additionally, the continuously increasing rotor diameters of new turbines lead to a goal conflict in the development phase. On the one hand, enormous quasi-static load components in the bending moments require a stiff rotor support to protect other sensitive drivetrain components from non-torque loads. On the other hand, the weight of the structural components, e.g. the main bearing housing or the main frame, cannot be increased arbitrarily due to limitations in manufacturing and logistics. As a result, new concepts have to be developed in which deformations and displacements may need to be



deliberately tolerated. Tuning of the stiffnesses of the elastic components for a defined transmission of the six-dimensional rotor loads into the tower requires complex simulation models considering relevant non-linear effects.

In addition, in the field of large size bearings with outside diameters greater than 1 m, the deformation of the structural environment as well as the relative movement of the bearing rings have a significant influence on the internal load distribution and the rating life [4]. Today's guidelines therefore rightly require the consideration of these system-dependent variables [5,6,7]. A suitable simulation of the rotor bearing support system at an early development stage with the aim of investigating the deformations is therefore essential for a reliable bearing dimensioning [8].

The simulation models used to initially design or improve a rotor support concept and to monitor, understand and evaluate the mechanical behavior of a rotor support in the field need to be validated appropriately. Only a validated modeling methodology allows a realistic investigation of the force transmission between the hub and the tower.

This contribution presents a realized methodology and results for the analysis of the rotor bearing system of a 3 MW wind turbine on a system test bench compared to a simulation model. In this way, requirements for a realistic modelling of a rotor bearing support system and concepts for monitoring its operating conditions respectively the rotor input loads can be derived.

2. Methodology

In order to determine the model depth, which is needed to analyse the mechanical interaction of the components in a rotor bearing support system, and to evaluate the quality of simulation results, a suitable validation procedure is necessary. To clarify the basic structure of the subsystem investigated here and to derive first requirements for the simulation and its validation, the theoretical force transmission between the hub and the tower flange is illustrated, assuming a linear behaviour. For a classical three-point mounted design, as examined here, figure 1 provides a corresponding overview.

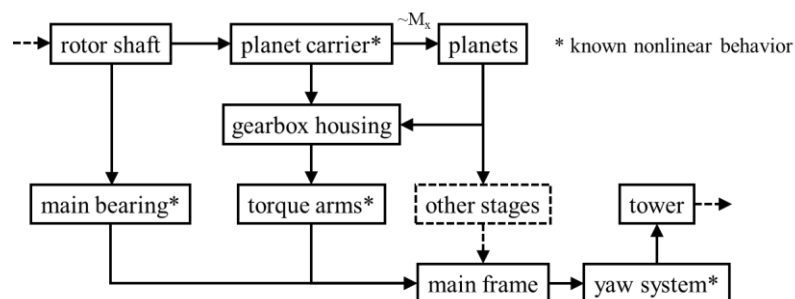


Figure 1. Theoretical force transmission between the hub and the tower flange.

It becomes obvious that the forces and moments applied by the rotor can take different paths through the support system. Along the way the loads pass through various mechanical components. These have nonlinear properties because of their internal structure and can divide the loads on subsequent paths or can influence neighbour components by their nonlinear relationship between load and deformation. The consequence can be unexpectedly high component loads which have not been considered in the design phase. A good example is the widely used spherical roller bearing (SRB) as the main bearing which includes over 60 moving parts, resulting in a nonlinear stiffness behavior incl. clearance. The axial sliding movements of the rolling elements may lead to an unwanted thrust introduction into the gearbox.

To understand these and other phenomena during operation and to be able to consider relevant interactions in simulations, the validation procedure puts the focus on the nonlinear components. The simulative investigations are carried out with the help of an implicit finite-element software, see section 2.2, because a detailed representation of the nonlinear and structural components as well as the load application is required. The experimental investigation is done on a full-size system test bench with

a complete nacelle, see section 2.1. The validation procedure (figure 2) is based on displacements which can be measured much easier than internal loads.

On the test bench as well as in the simulation, quasi-static loads are applied on the rotor flange. The displacements are compared to improve and finally validate the model. Provided that the simulation satisfactorily represents the mechanical displacements under the chosen load case, the model can be used to analyse the internal loads of different paths. In case of deviations, it must be clarified why the effects observed on the test bench are not included in the simulation results and which cause they have respectively.

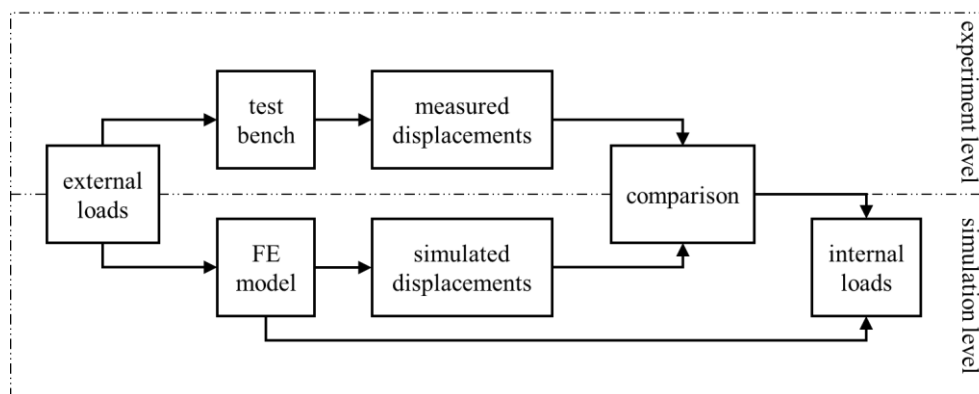


Figure 2. Approach for analysing the force transmission in the rotor bearing support system.

2.1. Experimental investigation

In order to be able to analyze the complex interactions between the mechanical, electrical, aerodynamic and control components from a scientific point of view and independent from a special OEM, a generic 3 MW research nacelle was built in the context of the project “Loads on Drivetrain Components of Wind Turbine Generators” [9]. The investigations described here are based on this nacelle. The drivetrain has a classic, modular, three-point mounted design with a SRB and a three-stage gearbox ($i \approx 63$) for a fast-running generator.

The research nacelle is operated on the 4 MW system test bench of the Center for Wind Power Drives [9,10]. The dynamic wind loads at the mechanical rotor interface are applied by a 4 MW Direct Drive Permanent Magnet Motor, followed by a servohydraulic, backlash-free Load Application System [11]. Therefore, the full six dimensional load vector can be transmitted dynamically at the hub flange of the nacelle. Compared to a time consuming field test, the system test bench offers the unique possibility to examine the components in system configuration under realistic and deterministic operating conditions.

To characterize the force transmission in the classical three-point suspension, various sensors were installed along the load path between the hub flange and the upper tower flange. Figure 3 gives an overview. The relative displacement between the inner and outer ring of the SRB is determined by three radial and three axial non-contact, inductive distance sensors. The deformation in all three dimensions of the clamping bushings in the torque arms is detected by LVDT (linear variable differential transformer) transducers at all four supports, figure 4. The characteristic stiffness curves of the bushings were determined on a component test bench prior to assembly. In this way, the currently acting reaction forces can be approximated. On the rotor shaft six strain gauge full bridges were applied for investigating the cutting forces and moments for further investigations. The signals are transmitted via a telemetry unit. In addition, eight strain gauge full bridges were applied on the stationary system on the main frame. Thin-walled structures, such as ribs near the SRB, were selected as measuring locations. Within the gearbox, the relative displacement of the planet carrier and its bearings is detected.

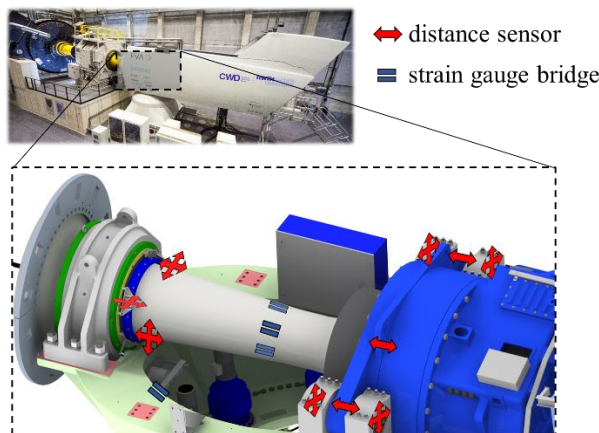


Figure 3. Installed sensors in the rotor bearing support system of the research nacelle (selection).



Figure 4. Displacement sensors at a clamping bushing of the torque support.

The realized measurement campaign initially consists of six quasi-static characteristic maps of the main rotor loads (thrust force, tilt moment, yaw moment and torque). Each of these 2D maps contains the systematic variation of two main rotor loads starting from the pure rotor weight at standstill. The rotation speed of the drivetrain was kept constant during the experiments. The number of sampling points for each variable is 10 to 15, uniformly distributed over the normal operating range of the turbine. Too small numbers increase the risk of missing nonlinear effects and complicate the interpretation. For example, the map of thrust force and the tilt moment includes 225 load points. Each point is held for 1 min. The evaluation of the measured raw signals is completely automated. The resulting map of each sensor signal, e.g. a displacement, contains the mean values of every load point. The maps of different displacement sensors serve as input variables of a kinematic model which calculates the rigid body motion of selected components, e.g. the gearbox housing.

It is obvious that the measured maps only represent cutting planes in a hyperspace of all possible rotor load combinations, but they give an overview for supporting system understanding and model validation.

2.2. *Simulative investigation*

Parallel to the investigations on the test bench, a static finite-element model was set up, which serves to explain the measurement results on the one hand, and to derive modeling recommendations on the other hand, figure 5. The model structure is mainly based on the content from figure 1, with special attention to known nonlinear component behavior. All rolling element stiffnesses of the yaw- and main-bearing as well as the stiffness of the clamping bushings are modeled by nonlinear spring elements, figure 6. The gearbox is currently approximated by a rigid beam construction, but the axial clearance of the carrier bearings is considered via a stop connector. In this way axial movements of the rotor shaft only lead to a force transmission when the clearance is exceeded.

The masses of the gearbox and the generator are included via point masses. To ensure a realistic load application into the front, cast iron part of the main frame, the rear, welded part of the frame incl. crane runway are considered with shell and beam elements. In this way also the stiffening effect is considered. The yaw brakes, connecting the main frame and the tower adapter, are modeled with rigid connector elements, which inhibit the rotation of the yaw bearing and emulate a realistic force transmission. As a boundary condition, the tower adapter is constrained at its lower surface. The static loads are applied at the hub flange.

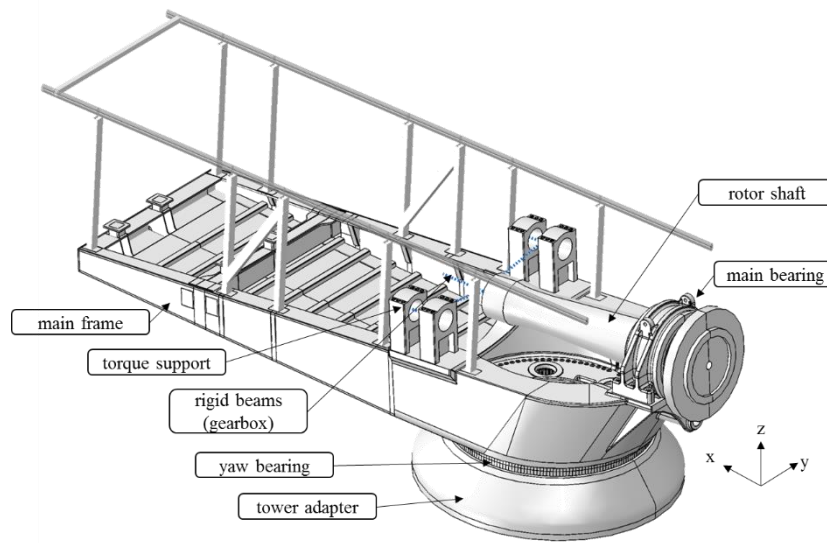


Figure 5. Finite-element model (unreduced).

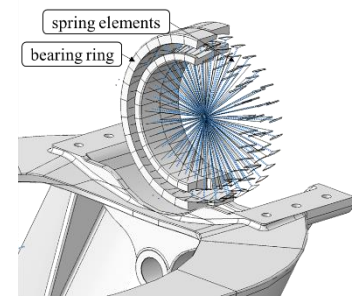


Figure 6. Modelling of the main bearing.

The described model contains over 800.000 elements and over 6 million degrees of freedom (DOFs). The calculation of the measured characteristic maps of rotor loads would require a lot of computing capacity. In addition, the solution cannot be determined directly because of the included nonlinear effects. An iterative solver is needed in any case. To reduce the calculation effort without a significant loss of information, all structural components, e.g. the main frame, are replaced by substructures. The linear stiffness and mass matrices of the meshed components are reduced to a selected number of nodes/DOFs while remaining the mechanical properties at the retained nodes. The nodes which serve as an interface to other components or to nonlinear elements are systematically maintained. In this way, the mechanical behavior of the model remains the same while the computation effort is reduced. The only general drawback is that without unpacking the substructure, no results of the reduced nodes and elements can be readout. But simulation outputs which require a lot of nodes, such as strain distributions on structural components, are of no interest in the context of the investigations presented here. The focus is clearly on the interfaces. By using the static model reduction in this case, the calculation time can be reduced by a factor of 140 with the same hardware. The final model only has about 700 elements and 9000 DOFs with identical results.

In the context of the simulative investigation, the same characteristic load maps as on the test bench are examined. These include about 900 single static load cases which are started and evaluated by scripts. Selected results from the measurement and simulation campaign will now be presented.

3. Results

In the following, a first comparison between the finite-element model and the real system will be carried out to investigate which observed effects can already be explained by the simulation. This is done using two examples.

3.1. Interaction of the main bearing and the planet carrier

On the basis of the map F_x - M_y (thrust force – tilt moment) the non-linear behavior of the main bearing can be shown. Figure 7 contains the measured axial displacement of the inner ring relative to the bearing housing in dependence of the additional rotor loads.

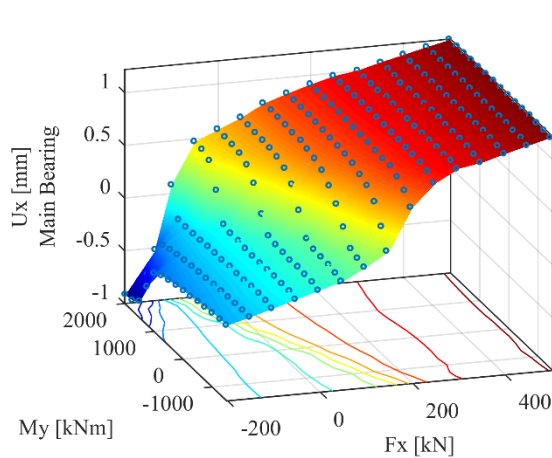


Figure 7. Measured axial relative displacement of the main bearing (SRB).

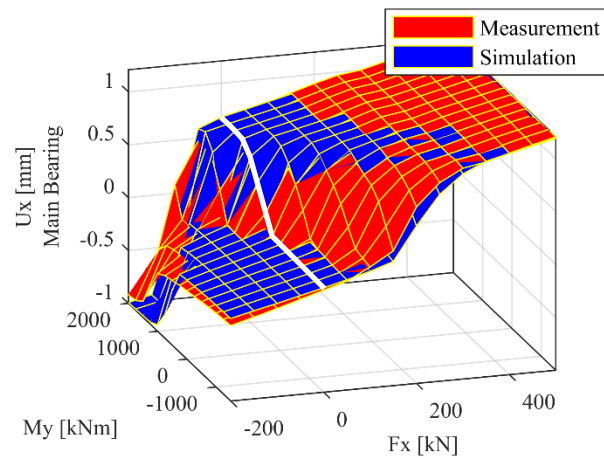


Figure 8. Comparison: measured and simulated axial displacement of the main bearing (SRB).

All values are starting from the reference state (standstill, only rotor weight). A positive value corresponds to a displacement of the inner ring in gearbox direction. The resulting surface contains approximately three plateaus of different sizes with interjacent transition regions. The plateaus can also be interpreted as states of the main bearing while the transition regions indicate a clearance pass. We will analyze these transitions in more detail below using the finite-element model. The requirement for this step is a sufficient match between the model behavior and the measurement. Figure 8 compares the two maps. It can be noted that the model represents the observed nonlinear effects qualitatively and quantitatively in a wide range.

To link the plateaus from figures 7 and 8 to main bearing load states, figure 9 shows the simulated load distribution of the two bearing rows in dependence of the tilt moment. The external thrust force is kept constant at 0 kN, so only the white section marked in figure 8 is considered.

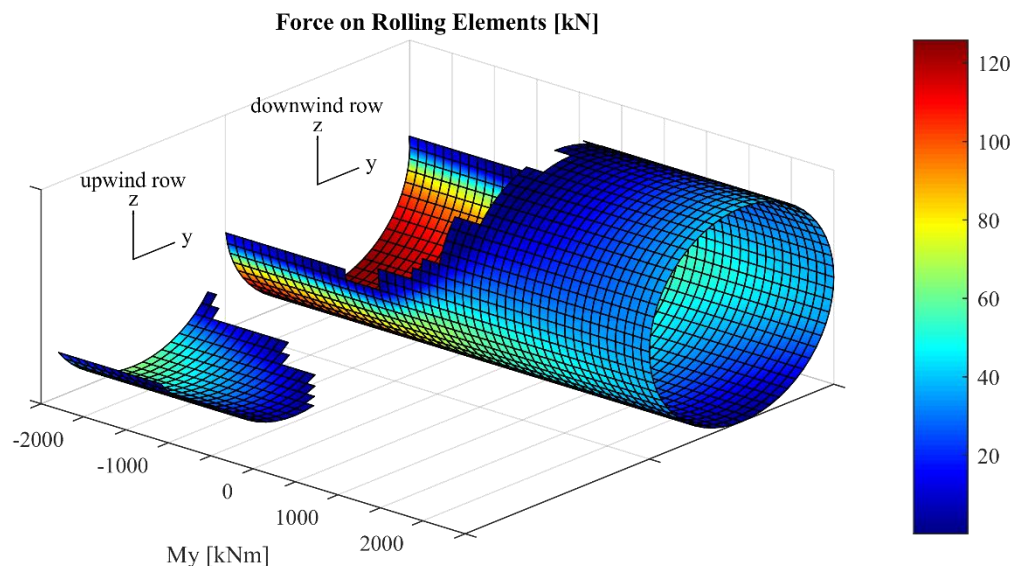


Figure 9. Simulated load distribution at the main bearing for a varying tilt moment.

While for negative tilt moments (in addition to the rotor weight) both bearing rows are loaded at the lower zone, positive moments lead to a complete unloading of the upwind row and a loading of the

downwind row on the entire bearing circumference. The change between these two states is associated with a clearance pass and can therefore be identified with high gradients in figure 7 and 8. The third state (only the upwind row is loaded) occurs only in combination with a negative thrust force is therefore not included in figure 9. In addition, it can be seen that a positive tilt moment M_y leads to a reduction of the total bearing load, as there is a compensation of the rotor weight. The observed axial sliding movements inside the bearing are a known disadvantage of SRBs relating to possible wear and unwanted axial constraining forces applied to the gearbox. The interaction with the gearbox in axial direction will now be analyzed in detail.

The question is whether axial forces are applied on the gearbox and, if so, in what amount. This is of particular interest because axial forces reduce the life time of the carrier bearings which, in this case, are designed as cylindrical roller bearings (CRBs). The axial clearance of the CRBs was adjusted by the gearbox manufacturer and is also considered in the finite-element model via a stop connector. By using the measured relative displacements between the carrier and the housing, the model can be validated. Figure 10 compares measured with simulated values. It becomes apparent that the axial bearing clearance is completely passed in both, the model and on the test bench. This results in an axial load application on the gearbox. The amount depends mainly on the stiffness between the planet carrier and the main frame. The axial stiffness of the clamping bushings of the torque supports has a significant influence in this chain. The stiffnesses of the CRBs and the gearbox housing have only a small influence and are therefore currently not considered in the model. The good agreement of measured and simulated axial displacements at the clamping bushings as well as the small gradient of the measured axial displacement of the carrier after passing the clearance in figure 10 confirm this statement. Since the model thus satisfactorily represents the axial movement of the carrier and the gearbox housing, the resulting axial constraining force can be read out.

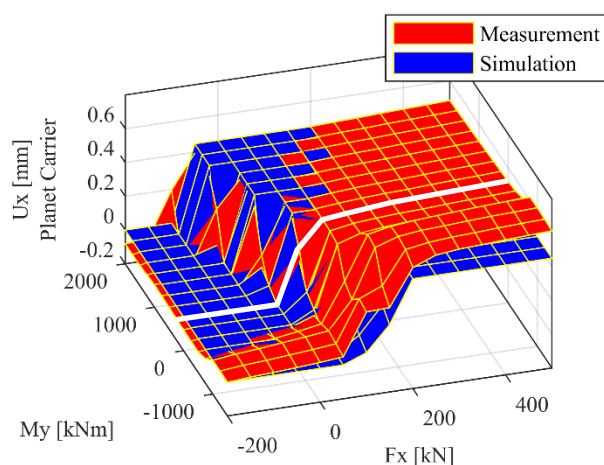


Figure 10. Comparison: measured and simulated relative, axial displacement of the planet carrier.

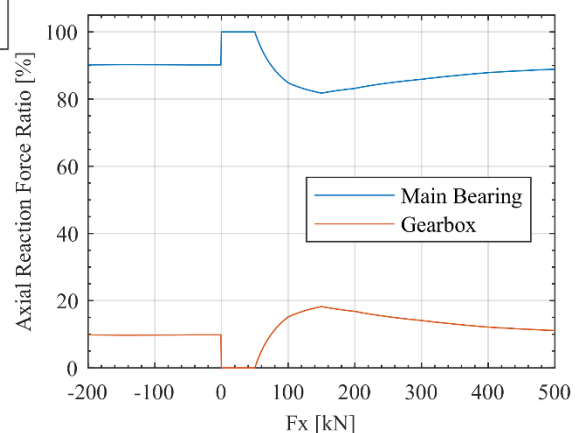


Figure 11. Simulated axial reaction force ratio for a varying thrust force.

Figure 11 shows the percentage distribution of the axial reaction force between the main bearing and the gearbox for the external thrust applied at the rotor flange. The tilt moment is kept constant at 0 kNm (white section in figure 10). Only between 0 and 50 kN, the main bearing can absorb the entire thrust force. In this range the clearance of the CRBs of the planet carrier is sufficient to compensate the axial sliding movements of the main bearing. A variation of the force ratio can be attributed to a sudden change in the stiffness ratio, e.g. a clearance pass or a change of the load distribution inside the main bearing. For example, between 0 and 150 kN there is a continuous unloading of the upwind row, while on the other hand more and more rolling elements are involved in the load transfer at the downwind row. Above 150 kN, the downwind row is loaded on the entire bearing circumference and the reaction force ratio only depends on the nonlinear stiffness curves of the rolling elements and the clamping

bushings. At a thrust force of 500 kN the gearbox must absorb a constraining force of about 57 kN. With an additional positive tilt moment, the value increases even further.

3.2. Movement of the gearbox housing due to external moments

Since the gearbox is part of the rotor support structure, the associated reaction forces and moments must be transmitted via the housing into the torque support. Due to the elasticity of the clamping bushings, a movement of the entire housing is the consequence. Below, the map M_y - M_z (tilt moment – yaw moment) will be investigated at first for evaluating the model quality regarding this output variable. Figure 12 compares the measured and simulated movement of the output shaft in the plane which is perpendicular to the rotation axis (y-z-plane). In the post processing of the simulation, a rigid gearbox housing was adopted for the conversion between the clamping bushings and the output shaft.

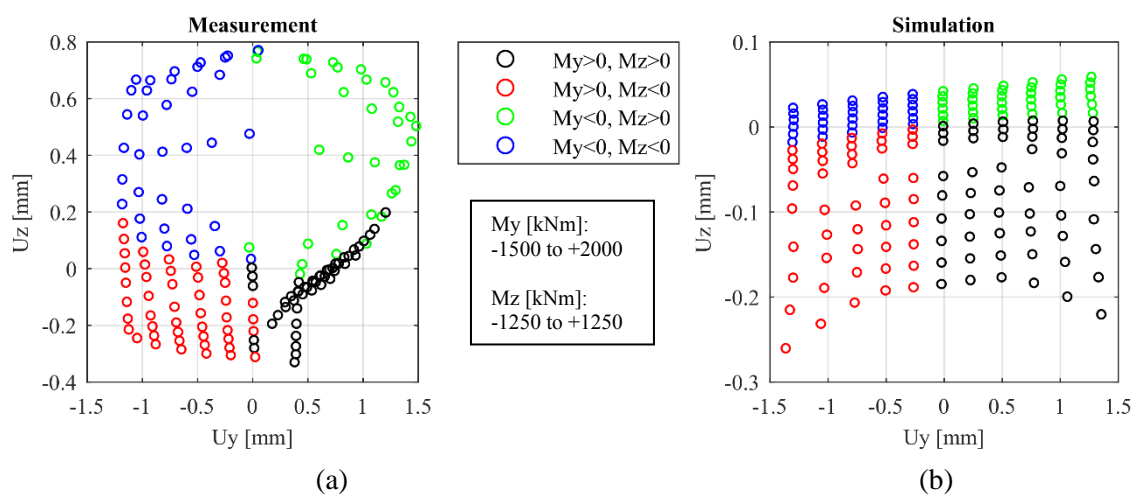


Figure 12. Comparison: measured and simulated movement of the output shaft.

The measurement and simulation results show significant deviations. While the maximum values in the y-direction are in the same range, there seem to be nonlinear effects in the z-direction which were not taken into account in the simulation. In the range of positive tilt moments the bolts in the clamping bushings of the torque support pass a kind of clearance, although they have been pretensioned during the assembly of the drivetrain. According to the current state of the author's knowledge, this can be explained by the weight and center of gravity of the gearbox. Since this point is located about 0.5 meters behind the torque arms, the four supports are loaded by a constant tilt moment during standstill and assembly. This leads to an eccentric zero position, figure 13. Negative tilt moments at the rotor flange result in an unloading of the rear bushings and a clear upward movement of the output shaft.

In addition, for positive yaw moments the gearbox seems to hit a stop (high stiffness) which depends on the vertical position of the gearbox. This effect is also not included in the model so far and therefore requires further analysis work. One possible reason could be a misalignment of the gearbox relative to the four torque supports.

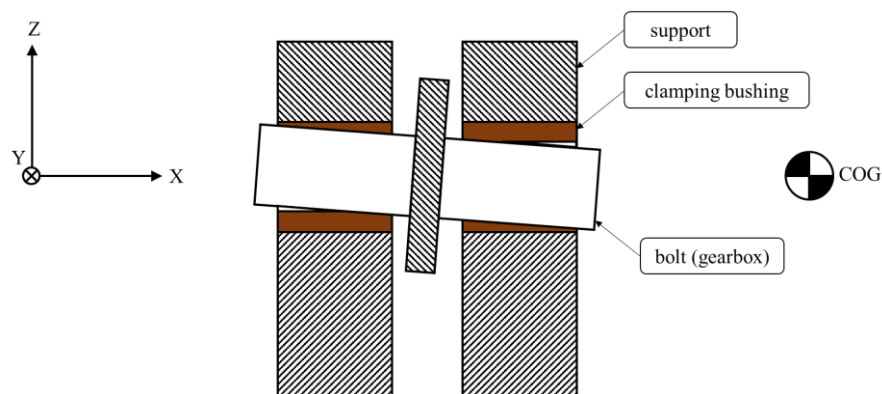


Figure 13. Schematic diagram: Eccentric zero position of the bolts (exaggerated).

4. Discussion

The investigations carried out in this contribution show that by applying a simplified simulation model, some effects that occur on the test bench can be explained and analysed in detail, e.g. the relative displacement of the main bearing and its effects on the planet carrier. Other effects require a deeper analysis of existing measurement signals in order to understand the mechanisms that are not included in the simulation results and, if necessary, to integrate them into the model. The radial behaviour of the gearbox suspension is currently approximated by two nonlinear spring elements per bushing pair. Although the characteristic stiffness curves have been determined on a component test bench, the measured displacement behavior of the gearbox shows deviations. It has to be investigated how the sudden stiffness reduction (clearance) for negative external tilt moments can be taken into account in the model. The cause of unexpected limitation of the horizontal movement of the gearbox can be further isolated by analyzing the deformation of each individual bushing pair and by evaluating the strain gauges on the main shaft and the main frame. Possibly there is an internal tension or misalignment within the system.

5. Conclusion

The specific application of distance sensors in the rotor bearing support system on a system test bench allows a quantitative investigation of the nonlinear force transmission. This enables the validation of simulation models, which are then extremely valuable because internal load paths can be analyzed. In this way, it can be verified that the drivetrain components are properly coordinated to each other at their interfaces so that no unexpected load paths occur. In this contribution, it could be shown that non-negligible forces are applied on the CRB of the planet carrier on the generator side under normal operating conditions.

The main requirement for a realistic modeling of the rotor bearing support system with the aim of analyzing the force transmission is the consideration of significant non-linear stiffnesses. As it has been shown, this includes especially roller bearings. It is not necessary to implement the surface contact of each rolling element, instead the focus should be on the application of suitable analogous models, e.g. connectors, which allow the calculation of the load distribution and reduce the computing effort.

Nevertheless, there are always effects that cannot be represented by simulation models in advance unless the user implements them. These include misalignments for example.

However, there is still room for improvement relating the behavior of the presented model under normal load conditions. In addition to a more detailed modelling of the planet carrier and its bearings, the first gearbox stage could also be included in a simplified way to analyze the impact on the planet load distribution. Also, the mechanical behavior of the clamping bushings must be considered more precisely.

Once the system is fully understood, the sensor signals can be evaluated for their suitability for monitoring purposes and inverse input load determination.

6. Acknowledgement

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