The Validation of MBS Multi-Megawatt Gearbox Models on a 13.2 MW Test Rig

Robust and reliable gearbox designs for wind turbines require in-depth insight into gearbox dynamics. Dynamic load simulation and resonance analysis based on advanced multi-body simulation (MBS) models of gearboxes are useful methods to predict loads and dynamic phenomena already in early phases of gearbox design. Moreover, the quasi unlimited number of simulated 'sensors' in virtual MBS models provide a much wider insight than a limited number of measured sensors in real gearboxes.

In addition, MBS based gearbox analyses can even be done before the real gearbox is available. However, insights gained from MBS models are only useful for design purposes if the simulation results prove to be representative of real world situations. In order to increase the confidence level in these results, ZF Wind Power, formerly known as Hansen Transmissions, invested considerable effort in research activities focusing on the experimental validation of MBS gearbox models on its 13.2 MW dynamic test rig, see Fig. 1. As a result, ZF Wind Power is able to perform resonance analyses with confidence and collabotate with customers with respect to dynamic analyses using MBS models of the complete drive train.

THE RESEARCH PROJECT

This research project was unique in that the development of the gearbox MBS models for the investigation of gearbox and drive train dynamics in wind turbine applications was supported by an extensive measurement campaign at ZF Wind Power. The project took place from 2007-2011. An extensive measurement campaign was required for the validation of MBS models in order to guarantee the validity of the dynamic analyses with these MBS models. It was decided to validate these gearbox MBS models on a test rig rather than in a wind turbine environment because the conditions on a test rig can be controlled much better. In a wind turbine, one is at the mercy of arbitrary wind conditions, and the "test platform" is less accessible, making a well-structured validation plan much harder to achieve. The measurements used for the validation campaign took place on the developed 13.2 MW dynamic test rig [1] where a 2 MW and a 3 MW gearbox were placed in a back-to-back set-up. This enabled the validation of two different types of MBS gearbox models varying both in size and composition. The 3 MW gearbox has two planetary stages and one parallel stage and the 2 MW gearbox has one planetary stage and two parallel stages. The validation campaign was comprised of over 200 measurement sensors including accelerometers, strain gauges, rotational speed encoders, distance measuring equipment and torque sensors. The measurement campaign included both quasi-static measurements and measurements of dynamic events which were designed to be representative of wind turbine conditions. The set-up of the test rig is a highly accurate representation of a real drive train in a wind turbine. This test set-up was modeled in SIMPACK and included the two gearboxes and major parts of the 13.2 MW test rig such as the generator rotors, the optional speed reducers, and the high speed and low speed couplings, see Fig. 2. The complete MBS model of the test set-up had 600 degrees of freedom, and included more than 150 bodies, 13 of which were flexible.

ZF WIND POWER'S GEARBOX MODEL

ZF Wind Power has developed a general modeling strategy for dynamic analyses with MBS gearbox models. This gearbox MBS model is a flexible 6 degrees of freedom model and is composed of several parts, see Fig. 3. Structural components with a complex geometry such as the planet carrier and gearbox housing are represented by means of flexible models using modally reduced Finite Element (FE) models. The gears are represented by rigid bodies whereas spring-damper systems representing the gear mesh non-linear behavior are included in the model by means of Force Element FE225. This non-linear gear mesh stiffness causes internal excitation in the gearbox whereas clearances between the rigid gear bodies are also included in order to simulate events in which gears are going through their clearance. Shafts are modeled using the SIMBEAM approach. In this way, shafts are also represented by a flexible model in the form of a reduced one-dimensional finite element model using Timoshenko beams. Bearings are modeled at the system level as spring-damper systems with full 6x6 stiffness/damping matrices. Therefore, the important cross-coupling terms are included in the model. The stiffness matrices are usually linearized at nominal working conditions where the stiffness values are usually based on Hertzian theory. All of the previously described gearbox components were verified and/or validated in the course of the research project. The focus in this article is on two parts: the validation of complex structural components, such as the planet carrier and the housing, and the validation of the complete MBS model with the two gearboxes in back-to-back configuration on the test rig in the frequency domain using dedicated measurements.

"...dynamic resonance analyses of gearboxes with MBS should always include a flexible housing model.."

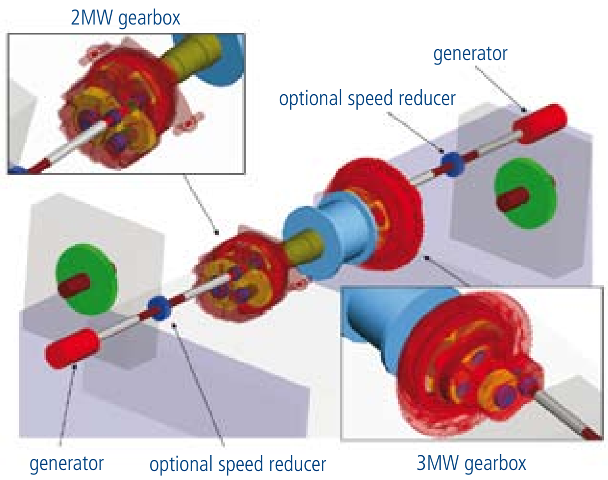

Fig. 2: MBS model of the test set-up

VALIDATION OF FLEXIBLE STRUCTURAL COMPONENT MODELS

The validation of FE models of complex structural components with measurements is important as these components contribute in a major way to the overall dynamic behavior of the complete gearbox. The validation of complex structural component models included the following items: the planet carrier of the first planetary stage of both gearboxes and the "empty" housing of both gearboxes. The "empty" housing model includes the parallel stage housing, covers, ring gear(s) and reaction arm but no shafts, bearings, and internal gears. Each flexible FE model was compared with its respective experimental modal model. This experimental modal model was obtained from a classic

Fig. 1: 13.2 MW dynamic test rig with REpower's 6 MW gearbox in a back-to-back configuration

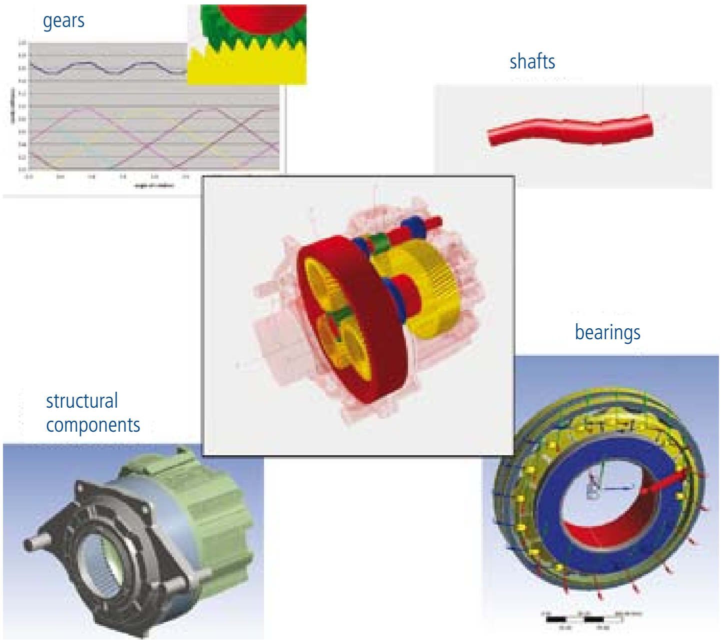

Fig. 3: ZF Wind Power gearbox MBS model

Experimental Modal Analysis (EMA) on each of the previously described structural components using a shaker and accelerometers, see Fig. 4. The evaluation of the correlation between the flexible FE model and the experimental modal model was based on the Modal Assurance Criterion (MAC) [2]. It was found that the flexible FE model (i.e., both mode shape and eigenfrequency) correlated well with the experimental modal model. An example of this correlation exercise can be seen in Fig. 5 and Fig. 6. In Fig. 5, the FE model of the 2 MW housing is shown in blue and the experimental modal model is represented by the red wireframe for a particular mode shape of the empty housing.

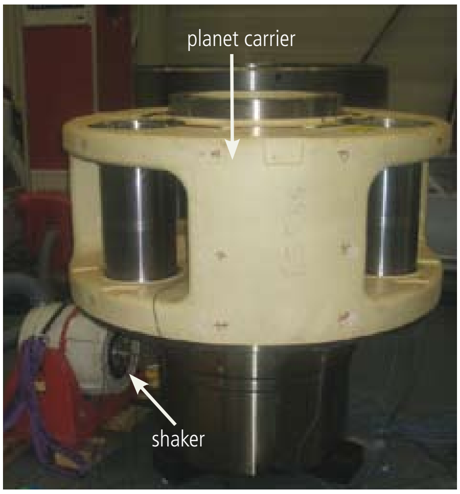

Fig. 4: Experimental Modal Analysis of the 2 MW planet carrier

The MAC matrix of the 2 MW housing is shown in Fig. 6. The first 3 eigenmodes have an MAC value larger than 90% on the diagonal whereas modes 4 up to 8 have MAC values on the diagonal which are larger than 70%. The good correlation between the housing model and experiments supports the validity of the model. An important result from the EMA's and the FEA's was that the first eigenfrequency of the "empty" housing of the 2 MW and 3 MW gearbox has been identified between 100 and 300 Hz.

VALIDATION OF THE TEST RIG MODEL IN THE FREQUENCY DOMAIN

An important part of the validation of gearbox MBS models for wind turbine applications is situated in the frequency domain. Operating in variable speed wind turbines, these gearboxes will inevitably encounter resonance phenomena during operation. The purpose of the MBS gearbox model is to be able to predict the "most important" resonances (i.e., resonances with relatively high amplitudes which occur frequently in the operational range of the gearbox). For the validation of the MBS model in the frequency domain, the modal model obtained from simulations needs to be compared with the modal model obtained from experiments. In contrast with structural components which are supported statically, classic experimental modal analysis (EMA) cannot be used for the validation of MBS models of complete drive trains or gearboxes since these need to be in "operation" in order to obtain reliable results for the modal analysis. Moreover, several types of measurement sensors were used in the gearboxes and on the test rig which needed to be combined in order to define the overall mode shapes and eigenfrequencies. These issues provided an extra challenge for the validation of complete gearbox models in the frequency domain. Therefore, it was decided to come up with a composite technique for the modal analysis of gearboxes. This method is both semi-quantitative and semi-qualitative yielding quantitative values for the eigenfrequencies and qualitative descriptions for the mode shapes. A brief summary of the method is presented below. Campbell diagrams for all sensors are obtained from speed run-up measurements, see Fig. 7. In these diagrams, eigenfrequencies and their "confidence level" are identified based on engineering judgement for all measurement sensors. All sensors, eigenfrequencies and confidence levels are then combined in a "flat" 3D plot which is called a "Mode Map", see Fig. 8. A Mode Map indicates the eigenfrequencies in the drive train or gearbox. Moreover, the Mode Map shows the sensors in which the eigenfrequency is found. This allows for the possibility of obtaining a qualitative description of a mode shape. This method has been applied to both the measurements and the simulations. Since, with this composite method, mode shapes were not available in numeric form but only as a qualitative insight, correlating the eigenmodes cannot be done based on the MAC. This provided a new challenge. The correlation of eigenmodes and eigenfrequencies identified from measurements and the simulation model is obtained from mutual comparison of Mode Maps and orders. Further means for the correlation of eigenmodes and eigenfrequencies are (Auto)-Frequency Response Funtions and eigenmode animations from the MBS model which can also be compared with measured Mode Maps and Operational Deflection Shapes (ODS) determined with the measurement set-up. From this exercise, the major part of "most important" resonances for the test set-up have been correlated. The

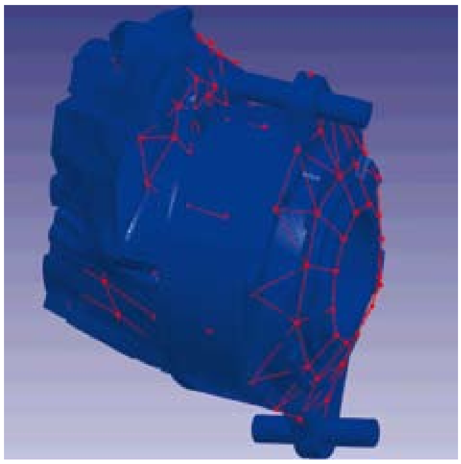

Fig. 5: Visual correlation of the modal FE model (blue) and the experimental modal model (red) of the 2 MW housing for an eigenmode

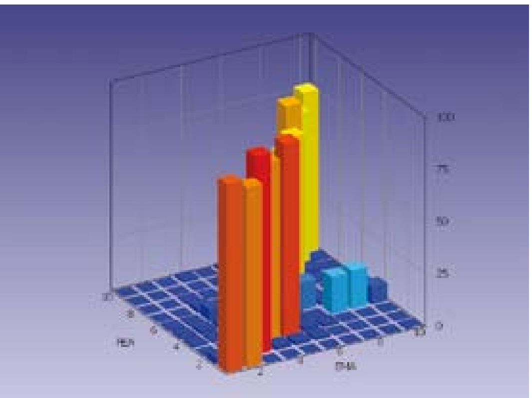

Fig. 6: Visualisation of the MAC matrix showing the mathematical correlation of numeric model (FEA) and experimental model (EMA) of the 2 MW housing shown in Fig. 4

first type of important resonances was the well-known torsional resonance which occurred between 0-100 Hz. These torsional modes are usually specific for the test set-up and have a more global character.
The second type of important resonance was a local gearbox mode. In this mode, the first mode of the "empty" housing which was found in the validation of structural components, was coupled with the parallel stage(s) as they are "connected" to the generator side of the housing. This coupling resulted in a significant decrease in the first "empty" housing eigenfrequency resulting in a shift of the eigenmode to a frequency range of 100-200 Hz. This research project was an eye opener as to the importance of the housing with respect to its contribution to the dynamics of multi-megawatt gearboxes. Therefore, dynamic resonance analyses of gearboxes with MBS should always include a flexible housing model. The result of the validation of the test set-up model in the frequency domain is that ZF Wind Power can confidently use MBS gearbox models for the prediction of the "most important" resonances [3].

CONCLUSIONS

ZF Wind Power has chosen SIMPACK as its main tool for dynamic analyses of gearboxes in wind applications. The validation of complex structural component models such as the housing and the validation of the MBS model of the test set-up with a 2 MW and a 3 MW gearbox in a back-to-back configuration on the 13.2 MW test rig has led to new insights into the importance of gearbox housings for the dynamics of multi-megawatt gearboxes.
During the project, major improvements were made to the techniques for measuring vibrations, and extensive experience has been gained in vibration analysis using the composite modal analysis technique.
Due to the extensive validation of gearbox MBS models on its 13.2 MW test rig, ZF Wind Power is now able to confidently use MBS gearbox models for resonance analyses of wind turbine gearboxes in (1) a retro-active phase for troubleshooting issues and (2) pro-actively in design decisions.

"...ZF Wind Power can confidently use MBS gearbox models..."

For the investigation of specific issues regarding gearbox dynamics, a dedicated approach is sometimes required, e.g., the investigation of the influence of non-linear bearing behavior. ZF Wind Power is gaining experience in these areas which sometimes require dedicated SIMPACK solutions.
It is ZF Wind Power's vision that MBS will become part of the virtual prototyping strategy. In this strategy, black-box models of the gearbox are incorporated into the virtual wind turbine. These combined MBS models will eventually lead to much shorter development times and gearbox dynamics which are better tuned to the wind turbine drive train right from the start.

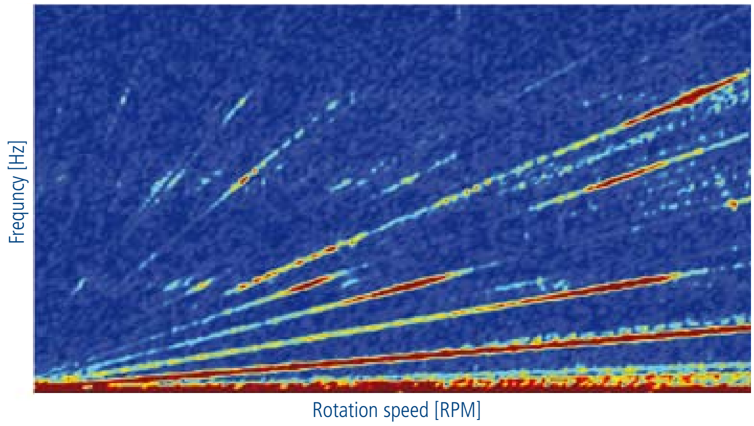

Fig. 7: Campbell diagram from speed run-up measurement for a torque sensor

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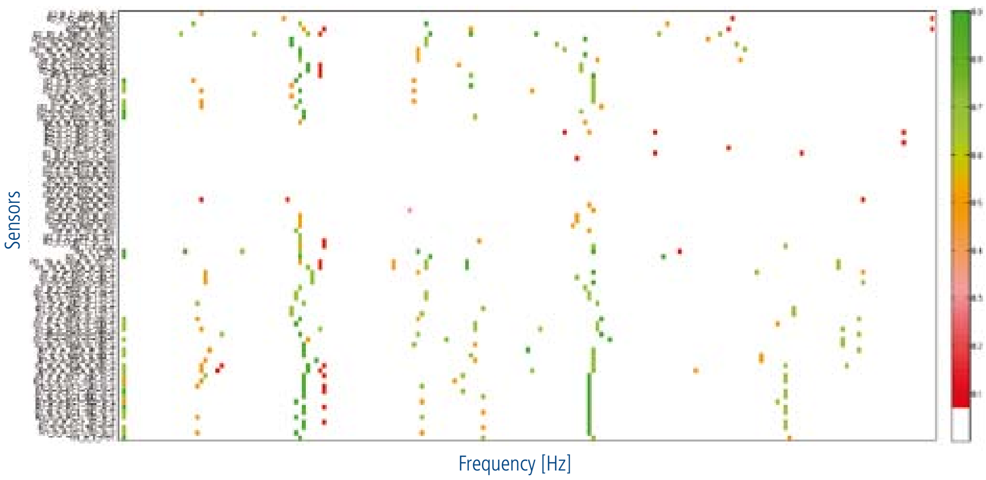

Fig. 8: Mode map indicating the identified eigenfrequencies for each measurement sensor. The colorbar indicates the confidence level for the eigenfrequencies (green = high confidence, red = low confidence)