Comparison of multibody simulations and measurements of wind turbine gearboxes at Hansen's 13 MW test facility

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Abstract

Continuous up-scaling of wind turbine size into the multi-megawatt class, together with developments for off-shore installation are calling for new wind turbine configurations and technologies. High product reliability is a key factor in these developments, cascaded down to each component manufacturer in the supply chain. Increasing the reliability of wind turbine drive trains for wind turbines with ever increasing size requires dedicated simulation models which can provide more insight in the internal gearbox dynamics in the early stages of the design process. However, simulation models can only add value to the design process if their results prove to be representative and reliable. Therefore validation based on measurements on real multi-megawatt wind turbine gearboxes is an absolute necessity. For this reason, Hansen Transmissions developed a back-to-back 13MW test rig set-up capable of dynamically testing gearboxes and validating their dynamical models. This validation is based on three foundations: validation of model parameters, validation in the frequency domain and validation in the time domain. The initial results of the validation of model parameters, validation in the frequency domain and validation in the time domain already demonstrate the added value of multibody gearbox models. Therefore the confidence level in the applicability of multibody simulation models in gearbox design is increased.

Introduction

Continuous up-scaling of wind turbine size into the multi-megawatt class, together with developments for off-shore installation are calling for new wind turbine configurations and technologies. High product reliability is a key factor in these developments, cascaded down to each component manufacturer in the supply chain. With a majority of currently installed wind turbines being gear driven, gearbox manufacturer's are challenged to deliver a drive system, at a high quality level, that will operate in a highly dynamic environment. Reliable drive train design requires good understanding of the gear unit and its dynamic behaviour, particularly in the operational conditions experienced in a wind turbine.

Increasing the reliability of wind turbine drive trains for wind turbines with ever increasing size requires dedicated simulation models which can provide more insight in the internal gearbox dynamics in the early stages of the design process. These drive train models should contain more than the typical one or two degrees of freedom in standard wind turbine design codes [1]. Therefore Hansen Transmissions chose to develop detailed multibody models of wind turbine gearboxes.
Multibody models represent the gearbox as a system of bodies, interconnected by spring-damper elements. These spring-damper elements represent force displacement relationships between two or more components. For the modelling of the gearbox dynamics full flexible 6 degree of freedom multibody models are used [2], [3]. These gearbox models can capture the most complex dynamic behaviour and contain, see Figure 1:

1) Structural components with complex geometry such as the housing and the planet carrier are represented as flexible components in the model using modally reduced finite element (FE) models.

2) Spring-damper systems representing the gear mesh non-linear behaviour. This non-linear gear mesh characteristic causes internal excitation. Clearances between the rigid gear bodies are also included in the model in order to simulate backlash.
3) Shafts are also modelled as flexible using reduced beam models (i.e. one-dimensional FE models).
4) Bearings which are represented as:
   a) Spring-damper systems with a full 6x6 stiffness/damping matrix, thus including all cross-coupling terms or,
   b) Spring-damper systems in the form of expressions for the representation of non-linear behaviour of bearings

These flexible components (structural components and shafts) allow for the description of local component flexibilities and the linked effect of component modes on the global gearbox behaviour. Moreover, dynamic stress evolution assessment is possible.

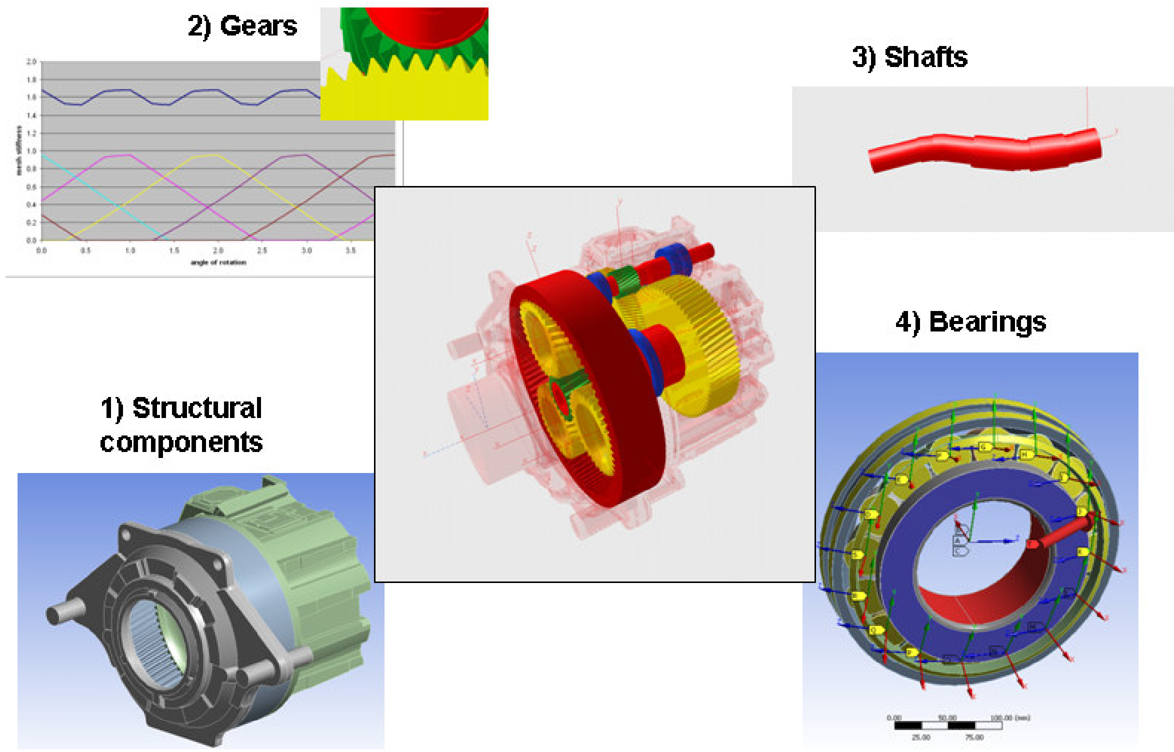

Figure 1: Full flexible 6 DOF gearbox models

These full flexible 6 degree of freedom multibody models give insight in the internal gearbox dynamics in:

- the time domain: torque, detailed reaction forces (e.g. in the bearings) and internal component deformations and
- the frequency domain: rotational, translational, tilt and flexible eigenmodes.

With these models it is possible to assess dynamic loads on all drive train components as well as (transient) phenomena, which could be harmful to the gearbox.

However, multibody models can only add value to the design process if simulation results prove to be representative and reliable. Therefore an extensive amount of time is dedicated to:

- thorough verification of these models by means of comparison with hand calculations, finite element analyses and existing gear calculation codes

- validation based on measurements on real multi-megawatt wind turbine gearboxes in the back-to-back 13MW test rig set-up at Hansen Transmissions.

This paper presents the validation methodology which is applied to the multibody models of wind turbine gearboxes as well as the initial results of the validation process.

The validation methodology

The aim of validating the multibody gearbox models is to make these models valuable for the design process by allowing a detailed analysis of gearbox dynamics. Correlation of measurements with numeric simulations and sequential model updating results in a validated gearbox model, being a transfer function from load input to load response, see Figure 2.

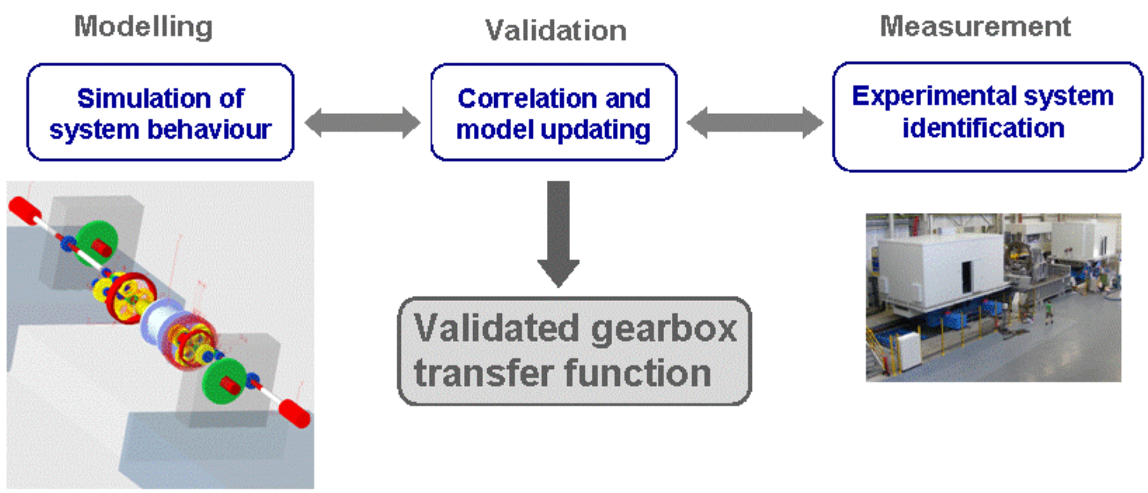

Figure 2: Validation procedure

Input for this validation should result from an extensive measurement program using instrumented gearboxes to yield sufficient insight in the actual internal gearbox dynamics. However, a measurement campaign on a wind turbine gearbox in the field poses several difficulties. These arise due to:

- the practical limitations on the amount of sensors which can be installed in a wind turbine gearbox in the field
- the limited amount of control on the measured load cases: the loads from the wind and the electricity grid are variable and cannot be completely controlled
- practical limitations in a commercial wind turbine

In order to circumvent these problems it was decided to test the gearboxes for the validation campaign on the newly developed highly dynamic 13.2 MW test facility which is also used for the validation of the new generation of wind turbine gearboxes. Therefore this test facility incorporates the following features:

- Reproducing wind turbine operational conditions in a controlled environment, with sufficient accuracy and repeatability
- Capturing the high level of dynamics in wind turbine applications by means of dedicated and parameterized load cases programmed into the test rig controller
- Test rig size and capacity matching the continuously increasing wind turbine power in the market with consequent increasing gearbox size. Therefore it is one of the world's largest test facilities in its kind.

Multibody gearbox models can be validated on this 13.2 MW test facility instead of in a wind turbine since the test facility set-up, developed at Hansen Transmissions, is based on the vision that it is possible to transform wind turbine behaviour into test rig conditions [4], see Figure 3. Starting point is the current knowledge of typical gearbox loadings in wind turbines, resulting both from simulation of design load cases and experience from measurements. The left side of the back-to-back gearbox test rig represents the 'wind & rotor' and is composed of an electrical machine (motor 1), an optional speed reducer (3:1 gearbox) and a wind turbine gearbox (gearbox 1). The right side represents the 'grid & generator' and is composed of an electrical machine (motor 2) and an optional speed reducer. The optional speed reducers enable full power testing of wind turbine gearboxes in future so-called hybrid concepts where a low-speed generator is combined with a gearbox with a smaller ratio, e.g. for a nominal generator speed of 500 RPM (in comparison with the usual 1500 RPM). Analogous to the operation of a wind turbine, the 'wind' side of the test rig is speed controlled, whereas the 'generator' side is torque controlled. Test gearbox 2 is driven at a certain time varying speed - corresponding to variable wind speeds - and loaded with a certain time varying torque by the 'generator' – corresponding to the loading from the grid. Therefore the test conditions on the test rig will be very similar to wind turbine operating conditions. For reproducing effects of wind and grid dynamic loads on the test gearbox, typical phenomena have been identified and translated into 11 parameterised load cases, being "Start cases", "Run cases", "Stop cases" and one "Special case".

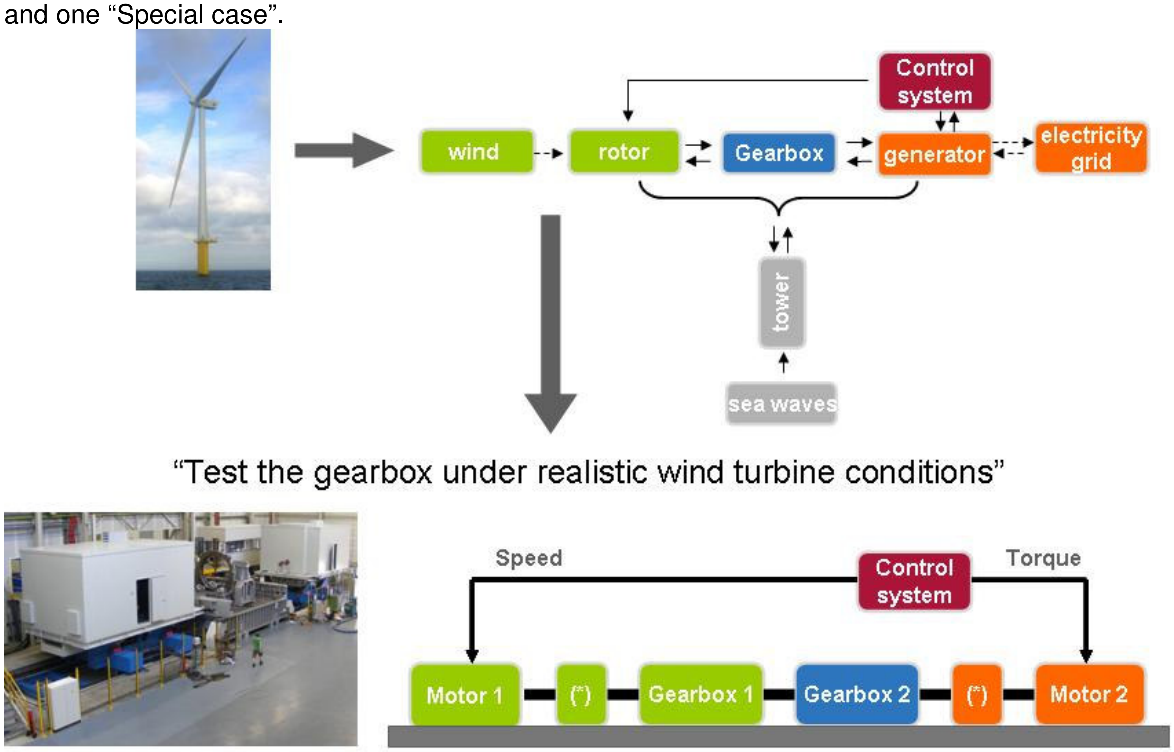

Figure 3: Test rig functionality

An extensive measurement campaign on the test facility with gearboxes in back-to-back setup provides the necessary experimental data to feed the validation of the multibody gearbox models. During these tests more than hundred sensors are used in order to yield sufficient insight in the actual gearbox dynamics. Instrumentation includes speed, torque, displacement, strain and acceleration sensors on all relevant and accessible components, see Figure 4. The measurement campaign consists of two major parts:

1. (quasi)–static measurements comprising speed and torque run-ups and run-downs as well as measurements at constant speed and torque. This approach results in a complete torque - speed 'matrix' which covers the complete operational range of the gearboxes under 'normal' operating conditions.

2. dynamic measurements comprising various start, run and stop cases completed with special cases representing typical loads in wind turbines. These measurements will not only cover 'normal' operating conditions but also various transient phenomena - typical for wind turbine operation - such as generator short circuits, grid disturbances or emergency shut downs leading to torque peaks and torque reversals.

The measurements provide the necessary input for [5]:

1. the validation of model parameters: measured torque run-ups are used for the extraction of torsional stiffnesses and bearing stiffnesses which can be directly compared with the multibody model
2. the validation in the frequency domain: mostly measured speed run-ups are used and transformed into the frequency domain for:
    a. eigenfrequencies from the multibody simulation model with the experimentally identified eigenfrequencies
    b. vibration amplitudes and phases from the simulated eigenvectors and the experimental data at the identified eigenfrequencies
3. the validation in the time domain: direct comparison of measured and simulated loads, speed and displacement signals are used for tuning the damping and clearances in the multibody simulation model

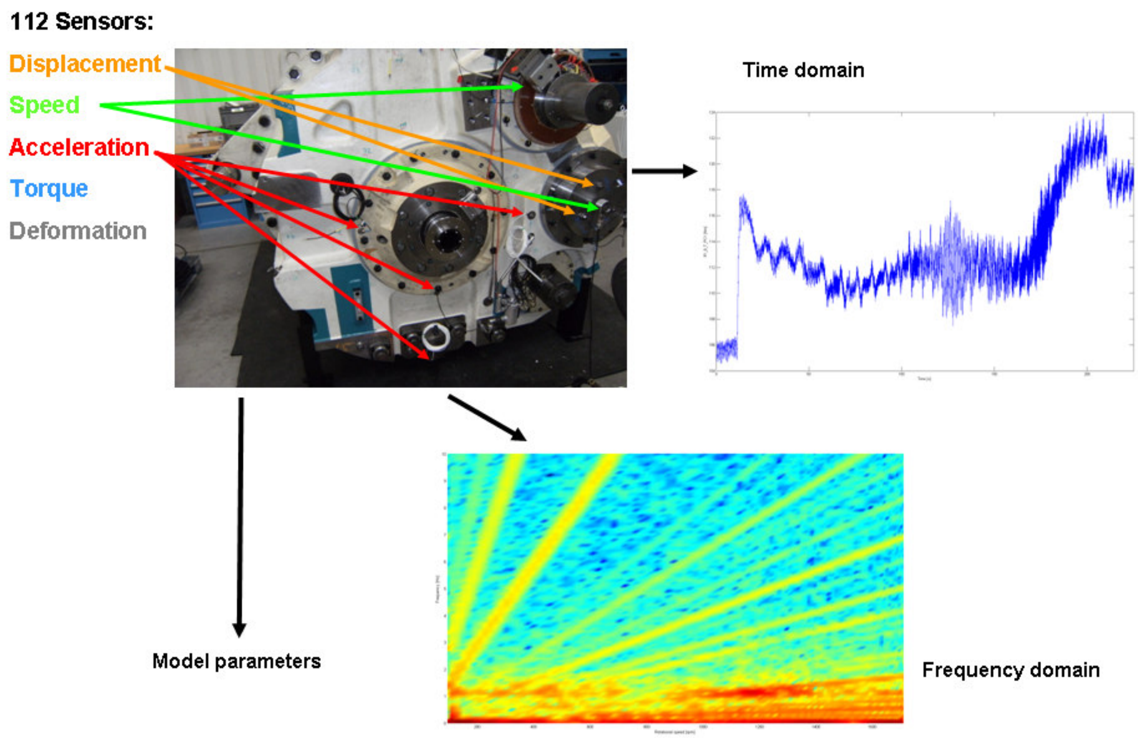

Figure 4: The foundations of the validation campaign

For straightforward comparison of simulated and measured results and consequent tuning of the numeric gearbox models, the multibody simulation model is built up representing the test conditions in a realistic way, i.e. a model including two gearboxes in a back-to-back set-up on the test rig, including the controller.

The validation results

This section covers the challenges and some initial results of the model validation process on the dynamic test facility. This section is divided in three different parts. The first part focuses on model validation by means of parameter comparison, the second part focuses on model validation in the frequency domain, and the third part covers model validation in time domain. In practice these three different methods are used to update the multibody model in this sequence, the first to verify / modify some model parameters by direct measurement, the second to modify model parameters to match the measured modal behavior, and the last to update model parameters that effect nonlinear behavior.

Validation of model parameters

For some model parameters, mainly stiffnesses, it is possible to measure them on the dynamic test facility. These measured values can be compared with the corresponding parameters that were used in the simulation model or otherwise can be extracted from the simulation model without the need to make long simulation runs in the time domain. Torsional stiffness is an excellent example of a model parameter which can be extracted from the measurements of the rotational deflection and torque during a torque run-up. The torsional stiffness can then be calculated as:

$$K = \frac{\Delta T}{\Delta \alpha} \qquad (1)$$

where ΔT is the difference in torque and $\Delta\alpha$ is the difference in rotational deflection. As an example the torsional stiffness measurement of the entire drive train is shown. The increasing difference in rotational deflection as well as the increasing torque during a torque run-up can be seen in Figure 5.

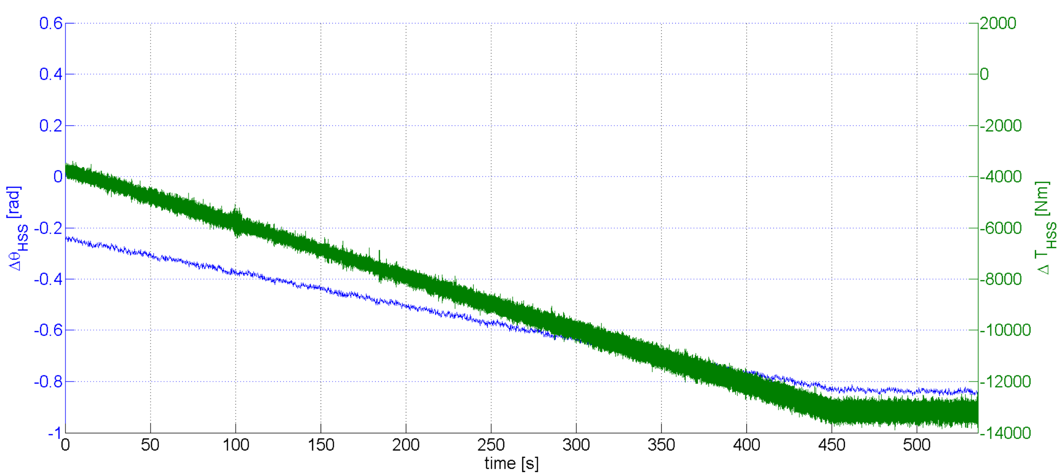

Figure 5: Torsion angle and torque level during torque run-up

The torsional stiffness of the entire drive train can then be calculated from Figure 5 and formula (1) and is 16153 Nm/rad. The torsional stiffness of other parts of the drive train can be estimated in a similar way. The individual and global stiffnesses can be used to update the simulation model or to place bounds on model parameters or on a combination of model parameters during an update process in the future

Validation in the frequency domain

Model verification in the frequency domain is used to match the modal behavior of the real gearbox on the dynamic testing facility with the modal behavior of the simulation model of the gearbox on the dynamic testing facility. A big challenge is to accurately extract the measured eigenfrequencies and mode shapes from the measured data such that they can be compared directly with the eigenfrequencies and mode shapes calculated from the multibody simulation model. In this section, a new approach to do peak picking is suggested. Instead of FFT's, used in the traditional peak picking approach [6], Campbell diagrams are used to find possible eigenfrequencies. Campbell diagrams are plots of vibration amplitude versus frequency and RPM. For each RPM, the contribution of the frequency in the ordinate is indicated using a color scale. In practice, such Campbell plots are measured by performing an RPM run-up and taking the FFT at every steady state. The FFT is plotted as a color contour on a vertical line inside the Campbell. The frequency of an eigenmode remains constant during run-up, resulting in a horizontal line, whereas the frequency of internal excitation depends on rotational speed, creating an inclined line. This makes it possible to separate eigenfrequencies from internal excitations. For each measured signal, a separate Campbell diagram was constructed and investigated, see for example the left figure in Figure 6. Since the identification of eigenfrequencies from a Campbell diagram is sometimes subjective, all the eigenfrequencies that have been found are given a confidence level ranging from "very unlikely" to "very likely". The resulting list of eigenfrequencies was mapped on a 2 dimensional color plot with in the ordinate the frequency and in the abscissa the corresponding measurement sensor where the eigenfrequency was found, the color for a specific eigenfrequency – sensor combination indicates the confidence level. The use of such a map has two main advantages:

1. Eigenfrequencies that were localised in different Campbell diagrams (i.e. different sensors) at the same frequency can be visually combined into one eigenfrequency. Finding an eigenfrequency at the same frequency in several Campbell diagrams increases the confidence in this eigenfrequency.
2. The dominant modal deformation locations of the corresponding modes can be derived from the 2D frequency – sensor map and gives a first impression of the mode shapes.

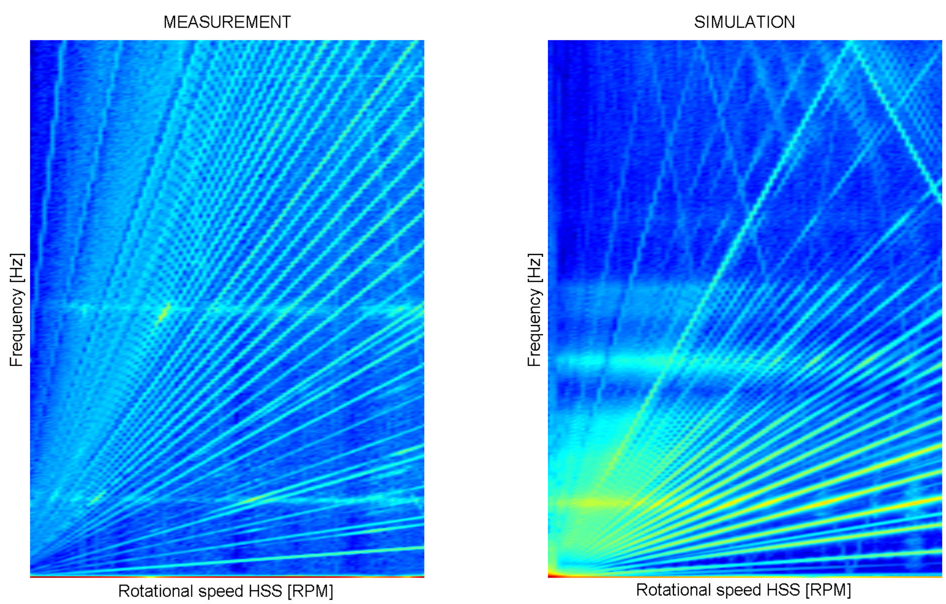

Figure 6: Comparison of Campbell diagrams from measurements and simulations

The experimentally identified eigenfrequencies can be used to validate the multibody model describing the test facility with two back-to-back gearboxes. The following approach is suggested to obtain a systematic and objective comparison. Based on the 2D frequency - sensor map it is known in which sensors the mode shape corresponding to the experimental eigenfrequency is mainly manifested. At the degrees of freedom of the model corresponding to these sensor locations auto-FRF's are simulated. With the use of these simulated auto-FRF's at the sensor locations a similar 2D frequency – sensor map for the simulation model can be made. Simulated and experimental eigenfrequencies can then be matched based on these two 2D frequency – sensor maps. The first results of this approach are shown in table 1, giving the percentage difference in eigenfrequency between the simulation and the experimental model. Due to confidentiality reasons the exact eigenfrequencies cannot be presented. Instead a frequency range for the eigenfrequencies is indicated. Based on this first comparison it can already be concluded that the multibody model shows good potential to describe the dynamic behavior of a gearbox. Another approach that has been used is the comparison of measured and simulated Campbell diagrams as shown in Figure 6. This figure shows very good correspondence for the lower eigenfrequency (4%) and less good correspondence for the higher eigenfrequency (21%). The correspondence between the measured and the simulated eigenfrequency for mode 2 and mode 6 can be improved by updating the multibody simulation model of the test rig with axial and radial stiffnesses.

NR.	FREQUENCY RANGE [HZ]	DIFFERENCE [%]
1	0-50	2
2	0-50	22
3	50-100	4
4	50-100	6
5	100-150	5
6	150-200	21
7	200-250	4
8	250-300	2

Table 1: Comparison of eigenfrequencies for experimentally identified and simulated modes

Nevertheless more detailed analysis of the measurement data is needed. Therefore future work will focus on using advanced experimental and operational modal analysis techniques to facilitate more detailed model verification.

Validation in the time domain

Model verification in time domain is used in a last step to verify or to tune model parameters that are difficult to verify or tune using any of the above methods. Often these model parameters result in nonlinear behavior that typically appears during load and speed variations. The backlashing due to clearance between the gears is such an example. During normal operation it is impossible to measure the backlash or any effect caused by it. However, during dynamic loading such as a torque reversal, loss of contact between two meshing gears will occur leading to backlash. Such torque reversals occur in wind turbines during idling or during an emergency shut down. In this section an emergency stop of the dynamic testing facility is compared with the multibody simulation results of an emergency stop. When an emergency stop occurs during normal testing on the test rig, torque is removed instantaneously from the two gearboxes which results in a large oscillation through the clearance of the entire test rig, mainly at the first torsional resonance. The corresponding mode shape damps out and the rotational speed of the gearboxes decreases due to the damping between the gears and in the bearings inside the gearboxes and test rig. The main parameters that govern the response in this load case are inertia, stiffness, damping and clearance values. Stiffness and inertia are mainly determined by the first torsional eigenfrequency and are well represented by the simulation model since there is good correspondence between the simulated and experimentally identified first torsional eigenfrequency (cfr. Table 1). The damping values and damping distribution determine how the average rotational speed and superimposed oscillation are damped out. The amount of clearance and the clearance distribution determine the duration of the loss of contact. By comparing measurements from this emergency stop with a simulated emergency stop, both rotational damping as well as backlash can be further tuned to improve the simulated model response. Figure 7 compares measured torque and rotational speed at one of the electrical drives during an emergency stop with the simulation model.

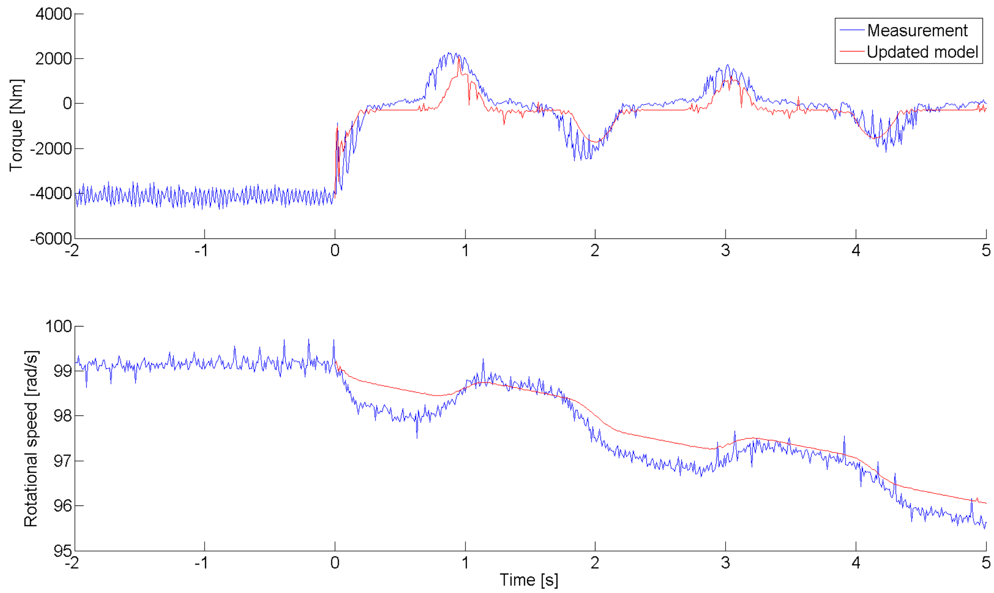

Figure 7: Comparison of measurements and simulations for torque and rotational speed during an emergency shut down

During the test, electrical drive 2 was outputting a torque of -4000Nm at a constant velocity of 100 rad/s. At t=0s, the emergency stop was performed, removing torque from both electrical drives instantaneously. The simulation started at t=0s with the correct initial boundary conditions. From the comparison of the measurements and the updated simulation model it can be seen that the updated simulation model already shows the correct torque amplitude and the correct amount of clearance (zero torque) as well as the correct speed slope for the deceleration. Further simulation model updates will even improve the correlation.

Conclusions

A validation methodology is shown which is applied to multibody simulation models of gearboxes. This methodology is based on validation of the model parameters, validation in the frequency domain and validation in the time domain. For this purpose measurements are performed on a specially developed highly dynamic 13.2 MW test facility on which it is possible to test gearboxes under realistic wind turbine conditions.

The initial validation results show good results of the Hansen multibody simulation approach. This demonstrates the potential of the use of multibody simulation models in gearbox design. Further experimental validation of the simulation models will be part of the future work.

The use of these multibody simulation models will help the development of reliable and innovative drive train products, help to decrease the kWh cost of renewable energy and lead to lower noise and vibration levels in the gearbox.

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