Efficient Modelling of the Drive Train Dynamics in Wind Turbines

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Summary

The dynamic behaviour of the drive train of a wind turbine is of mayor importance for the design loads and the verification of the machinery components. Multibody simulation tools (MBS) are widely used for this task. Models based on the MBS approach are used for the identification of eigenfrequencies and the resonance behaviour of the drive train. Using a generic wind turbine of the multi megawatt class, the impact of different modelling approaches and detail levels are investigated.

1. Introduction

The dynamic behaviour of the drive train of a wind turbine is of mayor importance for the design loads and the verification of the machinery components. Multibody simulation tools (MBS) are widely used for this task. Models based on the MBS approach are used for the identification of eigenfrequencies and the resonance behaviour of the drive train.

The possible level of detail for such a model ranges from simple torsional mass-spring-damper-systems with only a few rotational degrees of freedom to very complex systems containing flexible bodies and super-elements representing housings and foundations.

The determination of the required detail-level for each component in the model is of critical importance. A model with strongly reduced complexity will not yield to all required parameters or will result in insufficient precision for the predicted results. A MBS model of a drive train with a very high detail level will not provide more targeted information than a tailored model but will result in a high increase of the work load during the design process of the system.

Due to these contradicting factors it is necessary to identify the level of model complexity which is required for the sound prediction of the systems behaviour while at the same time keeping the effort on an acceptable level.

The aim of the work presented here is to identify the required level of complexity for components in the drive train model.

The quantity of components in the drive train and the arbitrary number of detail levels for each component lead to numerous individual models under investigation. In order to overcome this problem, a Design of Experiments (DoE) is set up in order to identify the most significant factors and to investigate the interaction between single components in an efficient way.

The work is carried out using the model of a generic wind turbine of the multi megawatt class with planetary and helical gear stages. The focus of the presented work lies on the drive train, i.e. the machinery components between the rotor hub and the generator. The influence of the rotors dynamic behaviour and the effect of the controller and generator are not taken into account.

2. Modelling Drive Trains

The current requirements for the modelling of drive trains for the certification of wind turbines are described in the GL Guideline for the Certification of Wind Turbines [1]. Furthermore, a Technical Note [2] provides information on the required detail level of the model.

2.1 Certification Requirements

The aim of the work defined in the guidelines is the analysis of the dynamic behaviour of the drive train using a detailed simulation model. Additionally, the model parameters assumed for the representation of the drive train in the global model are to be verified.

The current requirements towards the structure and the degrees of freedom (DoF) for the drive trains main components are given in the following table:

Drive Train Component	Minimum Component required	Minimum DOF required
Rotor blades	rigid body	edge wise and flap wise
Hub	rigid body	torsional
Main shaft	rigid body; elastic recommended	torsional
Low speed shaft coupling	rigid body	torsional
Gear box housing	rigid body	torsional
Planet carrier	rigid body	torsional
Gear box shafts	rigid bodies, elastic recommended	torsional; axial recommended
Gear box gears	rigid bodies	torsional; axial recommended
Elastic gear box support	rigid body	torsional
Brake disc	rigid body	torsional
Generator coupling	rigid body	torsional
Generator	rigid body	torsional
Elastic generator support	rigid body	torsional

Table 1: Modelling Requirements for the Certification of Drive Trains for Wind Turbines

2.2 Approach for Model Improvement

The target of an improved model quality shall not be pursued without limiting the modelling effort. In order to define a compromise between these contradicting targets makes it necessary to identify the required level of complexity for components in the drive train model. This level might vary from component to component. The driving factor is the influence of the individual component towards the overall result.

In order to identify the components which have to be subject to detailed investigation, a list comprising possibly influential factors is compiled. The list entries are ranked with respect to modelling effect and effort. This is based on the technique of pairwise comparison. The factors that combine high estimated effect with limited effort are selected for further investigation.

ID	Component / Model Approach
1	3 Point Suspension
2	Gearbox Output Shaft
3	Damping
4	Housing, Stiffness
5	Gearbox, 2nd Stage Shaft
6	Main Shaft
7	Coupling
8	Main Shaft Bearing
9	Bearing Characteristics
10	Gearbox, Floating Sun
11	Main Frame
12	Hub, Super Element
13	Planet Carrier, rot. Stiffness
14	Sun Wheel
15	ANOVA, Shaft Diameter (c, m, J)
16	ANOVA, Shaft Unbalance
17	Tower, Super Element
18	Gear Model, Constraint vs. Detailed Geometry Based
19	Gear Model, Tip Relief

Table 2: Components / Model Approach for model optimization prior to ordering

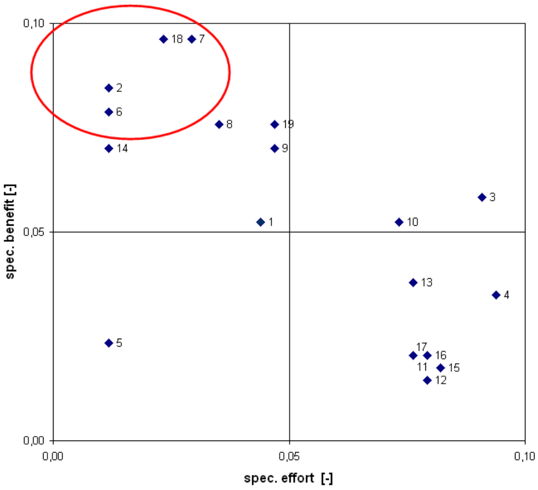

Figure 1: Benefit-vs.-Effort-Plot for Components. Selected Components in circle

3. Modelling Details

Based on the approach described above, the following components and modelling techniques are selected for further investigation:

- Main Shaft
- Gear Model
- High Speed Shaft / Generator Coupling

3.1 Main Shaft

In Multi-body-Systems, the main shaft of a wind turbine is typically modelled as a set of rigid bodies. The bodies are connected using joints and spring/damper elements. The joint type depends on the general modelling approach and the number of DoF in the model. For a model with only rotational DoF the connecting joint allows rotational motion around one axis, the remaining 5 DoFs are fixed. The spring/damper element represents a torsional spring with damping properties. In order to cover bending behaviour of the shaft, additional DoFs are needed.[3],[4]

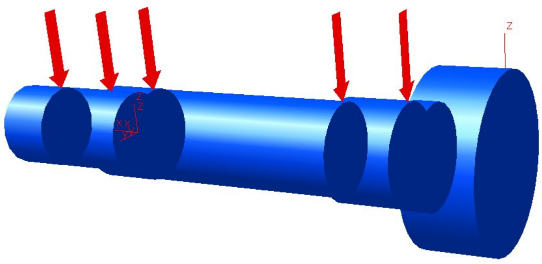

Figure 2: Multi-Body Model of a Main Shaft

More complex models of a shaft might use one of the following approaches:

- Bernoulli beam elements
- Timoshenko beam elements
- Structural solid elements

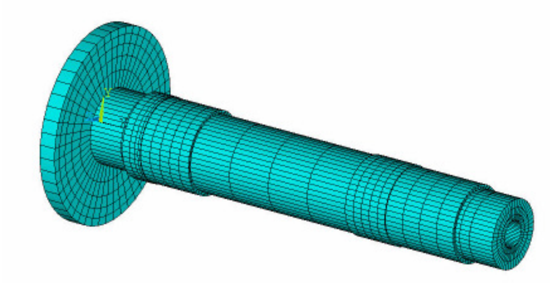

Figure 3: FEM Model of a Main Shaft

	Bernoulli beam	Timoshenko beam	Structural solid
1st bending	41.6 Hz	40.8 Hz	38.7 Hz
2nd bending	243.2 Hz	219.4 Hz	213.2 Hz
1st torsion	309 Hz	309 Hz	300 Hz
1st elong.	402 Hz	402 Hz	393 Hz
3rd bending	653 Hz	532 Hz	526 Hz
2nd torsion	776 Hz	776 Hz	740 Hz

Table 3: Eigenfrequencies of a Main Shaft for different Modelling Approaches

For the investigation of torsional modes the comparison shows that rigid bodies connected with spring/damper elements lead to satisfying results.

Since beam elements do not represent notch effects with respect to stiffness, the stiffness is higher compared to structural solid elements, leading to higher frequencies.

For the inclusion of axial modes Bernoulli-Beam elements should be used.

The effect of shear deformation on the bending behaviour can be found in the evaluation of bending modes. Elements that cover shear effects, i.e. Timoshenko-Beam and Structural Solids, lead to reduced bending stiffness.

For the consideration of shear effects and bending modes: Timoshenko beam elements should be the first choice. The use of solid elements requires a reduction of the mass and stiffness matrices, e.g. Guyan-Method or Craig-Bampton-Method. [5]

3.2 Gear Model

Two approaches for the modelling of meshing gears are investigated.

- Gear constraint with force element
- Purpose-built, time variant force element

The first approach combines a constrained equation for the speed ratio of two shafts with a force element. The force element represents a constant value for the meshing stiffness.

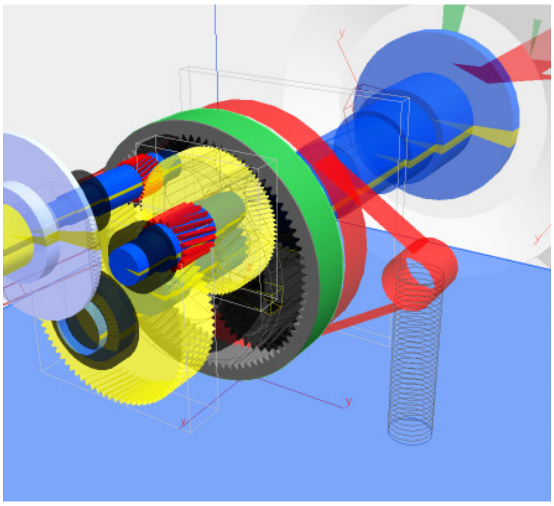

Figure 4: MBS Model of a Gearbox

The purpose-built force element allows the more realistic modelling of meshing gears. This element derives the meshing stiffness from the geometric data and the material properties of the gears. The influence of a variation in the gears centre and axial distance is included and has not to be additionally modelled. The influence of the variation of meshing stiffness is depicted in figure 5.

The analysis in frequency domain relies greatly on the correct description of the mass- and stiffness-distribution of the structure. Due to this fact, both gear models yield to similar results in the modal analysis.

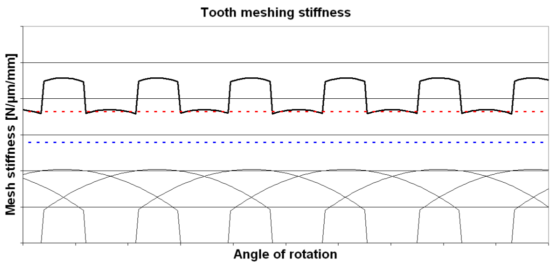

Figure 5: Meshing Stiffness as Function of Gear Rotation. Comparison of resulting stiffness with values derived from ISO 6336. [6]

The variation of the meshing stiffness results in an excitation of the drive trains torsional modes in the time domain. The result is depicted in figure 6.

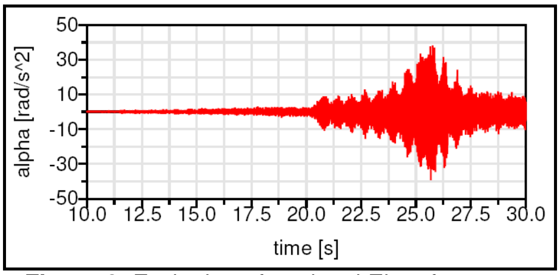

Figure 6: Excitation of torsional Eigenfrequency during Start-Up

The start-up of the turbines drive train leads to constant change of the excitation. At the intersection of excitational frequency and eigenfrequency, the resonance phenomenon can be observed.

3.3 High Speed Shaft / Generator Coupling

The output shaft and the generator coupling is the section of the drive train providing the smallest amount of torsional stiffness. Thus a large influence of these components on the first torsional eigenfrequency can be expected. The effect is investigated using a parameter variation.

Figure 7 displays the different frequencies of the drive trains eigenmodes. Modes which are not affected by the change of the coupling parameters remain on the diagonal. Figure 8 displays the residues for the comparison of the two configurations. Modes 1-6 are the first flap-wise and edge-wise modes of the rotor. These modes remain unaffected. Mode No. 7 is the first torsional mode of the drive train. The residue-plot (figure 7) emphasises this effect.

4. Conclusion

The analysis of different main shaft models yield that the use of rigid bodies and spring/damping elements lead to good results in the analysis of torsional modes.

The different gear models lead to a similar result quality in frequency domain for both approaches. The detailed gear model is favorable for transient analysis and for models with multiple DoF.

The torsional stiffness of the generator coupling has strong effect on the first torsional frequency of drive train. A detailed description of the couplings stiffness properties increases the models quality.

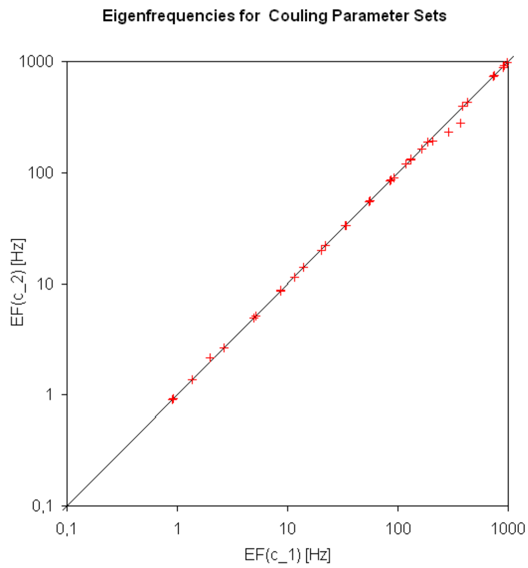

Figure 7: Eigenfrequencies of the Drive Train for two Sets of Coupling Parameter

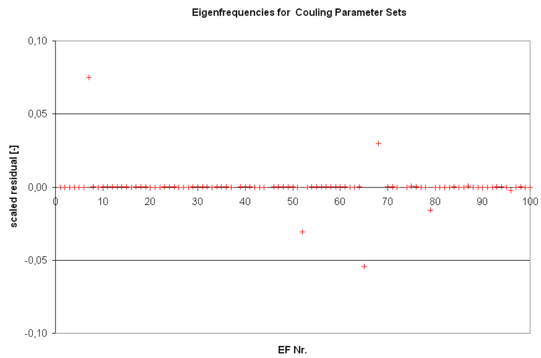

Figure 8: Eigenfrequencies of the Drive Train for two Sets of Coupling Parameter, Residues

5. References

[1] Germanischer Lloyd: Guideline for the Certification of Wind Turbines (Edition 2003 with Supplement 2004), 2004

[2] Germanischer Lloyd: GL Wind Technical Note 068: "Requirements and recommendations for implementation and documentation of resonance analysis", 2007

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