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Recommendations on Model Fidelity for Wind Turbine Gearbox Simulations

Preprint

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Contents

1 Abstract

This work investigates the minimum level of fidelity required to accurately simulate wind turbine gearboxes using state-of-the-art design tools. Excessive model fidelity including drivetrain complexity, gearbox complexity, excitation sources, and imperfections, significantly increases computational time, but may not provide a commensurate increase in the value of the results. Essential design parameters are evaluated, including the planetary load-sharing factor, gear tooth load distribution, and sun orbit motion. Based on the sensitivity study results, recommendations for the minimum model fidelities are provided.

2 Introduction

Gearboxes in wind turbines do not always achieve their expected design life [SS13]; however, they commonly meet or exceed design criteria as specified in current standards by the gear, bearing, and wind turbine industries and by third-party certifications. The National Renewable Energy Laboratory's Gearbox Reliability Collaborative (GRC) was established by the U.S. Department of Energy in 2006. Its key goal is to understand the root causes of premature failures and improve reliability using a combined testing-and-modeling approach.

Modeling the planetary section of wind turbine gearboxes can be quite complex, and excessive fidelity can significantly increase computational modeling time to the point that it becomes detrimental to obtaining an understanding of the planetary behavior. The challenge, then, is to determine the level of complexity required to accurately simulate planetary load sharing and motions. Facets of model fidelity include:

- Drivetrain complexity: For example, is modeling the main shaft or main bearing(s) needed? Is modeling the intermediate and high-speed stages needed? Is modeling the gearbox trunnion mounts or generator coupling needed? Can the planetary section be decoupled from the rest of the system?
- Planetary system complexity: When modeling the planetary section, how many degrees of freedom, number of elements, or modes are needed? Is flexibility of the ring gear or housing needed? Is detailed gear and bearing microgeometry needed?
- Excitations: Are gravity, dynamic loads, or gear and bearing stiffness variations needed?
- Imperfections: Are shaft misalignment, bearing clearance, gear tooth position errors, runout, and eccentricity important? These quantities vary from one gearbox to another and over time. If they significantly affect the planetary load-sharing characteristics, then a probabilistic approach to gearbox modeling is required.

The objective of this report is to examine the effects of model fidelity on measured planetary loads and motions. Modeling results from multiple tools that have varying levels of complexity are compared to each other and to GRC experimental data.

3 Model Fidelity Study

Multiple gearbox modeling tools and a simple lumped-parameter model were used to simulate the GRC gearbox dynamics in this study. Additional details about each model are discussed in a lengthier paper [YG15].

Various decisions about model complexity, excitations, and imperfections were made while modeling the GRC drivetrain to accurately capture the gearbox response. During the decision-making process, extensive sensitivity studies were conducted to understand the individual and combined influences of model parameters on the aforementioned design parameters. These studies helped establish the appropriate model fidelity, boundaries, and excitation features.

3.1 Drivetrain Complexity

Studying drivetrain complexity can determine the appropriate model boundaries, such as whether the gearbox should be modeled alone or modeled with the connection to the upwind main shaft/bearing and downwind generator/generator coupling.

Image 1 shows the planet gear upwind bearing load calculated using the model with the gearbox alone and the model with the gearbox and main shaft/bearing together compared to the experimental results. Without the main shaft and bearing, the planet-bearing loads were nearly constant. With the main shaft and bearing, the calculated bearing load fluctuated and the amplitude better matched the experimental results.

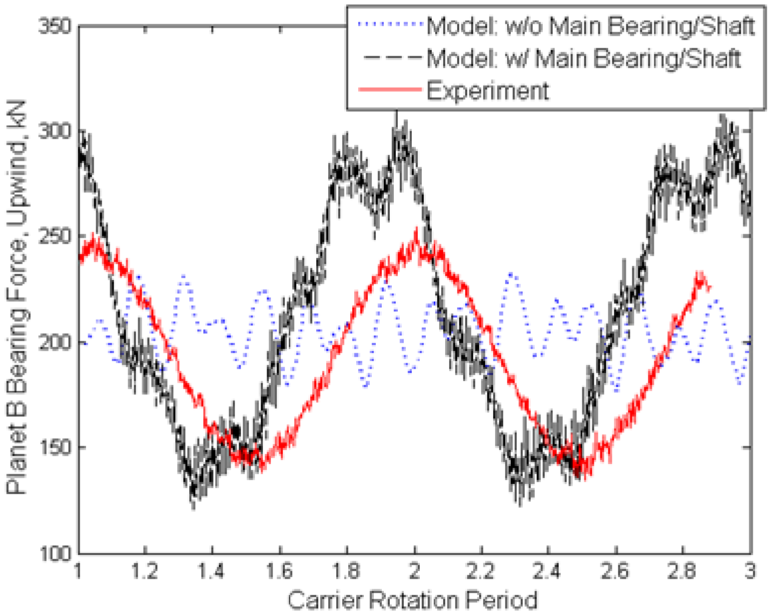

Image 1. Effect of main shaft/bearing modeling on upwind planet-bearing force

The bearing load fluctuations are caused by the rotor nontorque loads [YG141]. Including the main bearing/shaft is important to capture the nontorque load path to the gearbox. Further, the weight of the main shaft itself is an additional source of nontorque load that can affect the gearbox's internal response.

The gearbox trunnions support the gearbox weight and limit its torsional windup. At the same time, the trunnions allow axial and tilting motions of the gearbox. These trunnions are important elements to take into account in the model. The effect of generator shaft misalignment is being examined in current dynamometer testing.

3.2 Planetary Complexity

The study of planetary complexity focuses on gear tooth modifications (some simple gearbox models do not consider gear tooth modifications), ring gear flexibility, bearing clearance, and the spline connection. Image 2 compares the modeling results of the sun orbit amplitude calculated using the model with and without modifications and measured through experiments. The amplitude of the model without tooth modifications is more than double that with tooth modifications. The modeling results with tooth modifications are closest to the experimental results. The lack of modeling gear profile and lead tooth modifications leads to unexpected gear vibrations caused by an overestimated transmission error.

To model profile and lead modifications, gear tooth flanks in multibody dynamic models are typically divided into multiple slices. A convergence study was conducted to determine the optimal number of gear tooth slices. The optimum number of slices was determined to be 35, as shown in Image 3; however, using a large number of gear slices significantly increases the computational time.

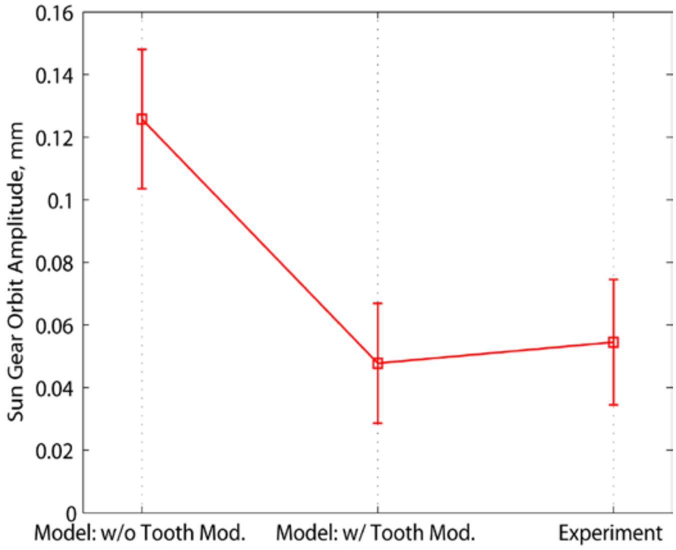

Image 2. Effect of the tooth microgeometry on sun gear motion amplitude

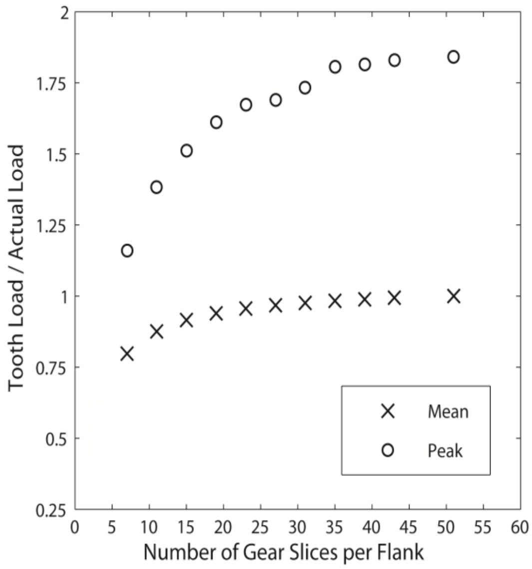

Image 3. Effect of the number of tooth slices on gear tooth load [WL12]

Accurate peak and mean loads are needed for a gear designer to measure the gear design parameter tooth load distribution factor, $k_{h\beta}$. Image 4 shows the calculated and measured $k_{h\beta}$ on the ring gear teeth at 0-, 120-, and 240-degree circumferential locations around the ring gear. The model includes 35 flank slices per gear. The model and the experimental data agreed reasonably well. Using a single tooth slice per gear (such as in a two-dimensional lumped-parameter model) leads to a line shape of $k_{h\beta}$ instead of the actual parabolic shape, as shown in Image 4. Therefore, it is important to include tooth microgeometry modifications to obtain accurate gear motion (Image 2) and tooth load distributions (Image 4). Futher, an accurate $k_{h\beta}$ is needed to calculate the planet-bearing load distribution among the upwind and downwind rows.

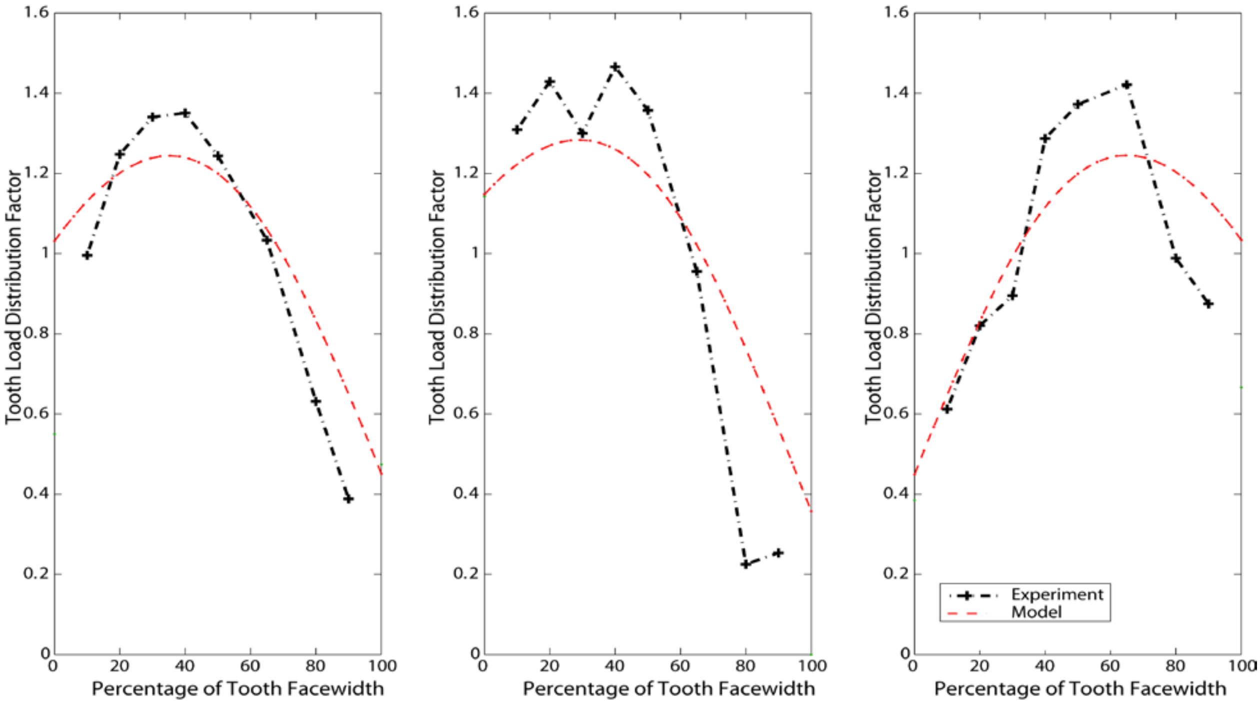

Image 4. Ring gear tooth load distributations at (left) 0-, (middle) 120-, and (right) 240-degree circumferential locations

Internal bearing clearance and preload are crucial modeling parameters that affect bearing load zone and stiffnesses. The effect of carrier-bearing clearance on planetary gear loads at several torque levels is shown in Image 5. Modeling results using analytical and computational approaches compared well to available experimental data. The carrier-bearing clearance was nondimensionalized by dividing it by the motion amplitude of the carrier when the carrier bearing had infinite clearance. The operating carrier-bearing clearance was 275 µm, and the planet-bearing clearance was not considered. Planet bearings were not in the nontorque transfer path to the gear teeth; therefore, the effects of planet-bearing clearance on load sharing were significantly less than those of carrier-bearing clearance. Increasing the carrier-bearing clearance clearly increased the load-sharing factor. When the carrier-bearing clearance became larger than the carrier motion with infinite clearance, the load-sharing factor converged to a constant. Additionally, the lower the torque, the more carrier-bearing clearance increased the load-sharing factor. At low torque, the load-sharing factor was high, because the nominal planet gear loads decreased with torque.

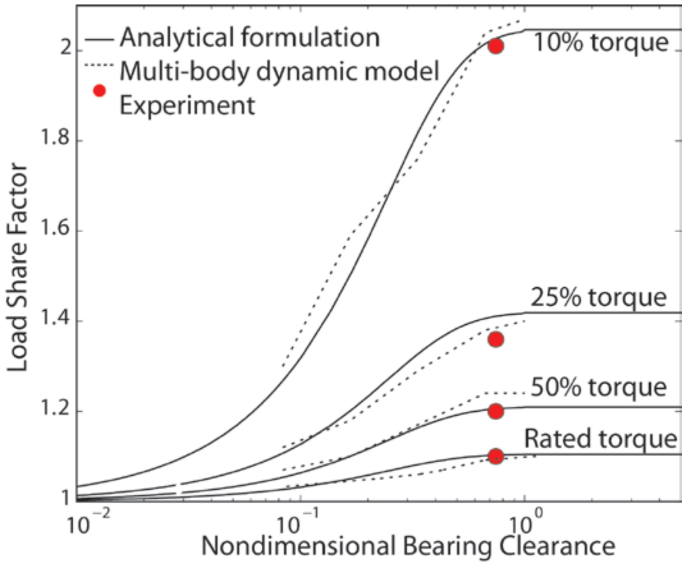

Image 5.Effect of carrier-bearing clearance on load-sharing factor [YG12]

3.3 Excitation Sources

Gearbox internal excitations are from the time-varying gear-meshing action and the stiffness variations of rolling-element bearings [YG142]. External excitations are primarily nontorque loads transmitted from the turbine rotor and gravity forces. Planetary load sharing is affected by bearing clearance, as described above, but even more so by nontorque loads (including gravity), as described in this section.

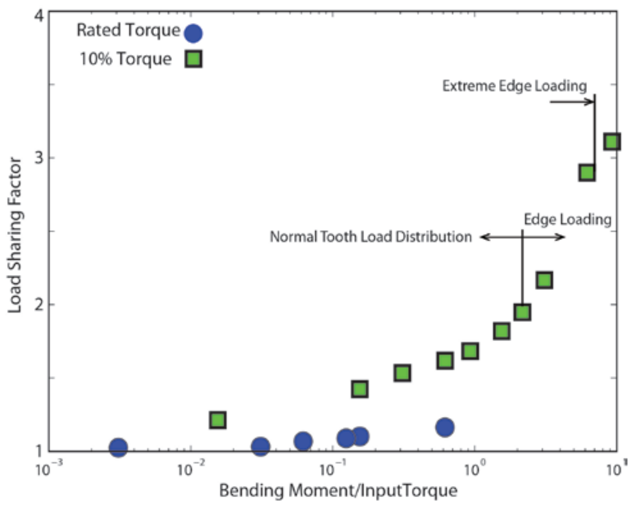

Image 6. Load-sharing factor sensitivity [YG12]

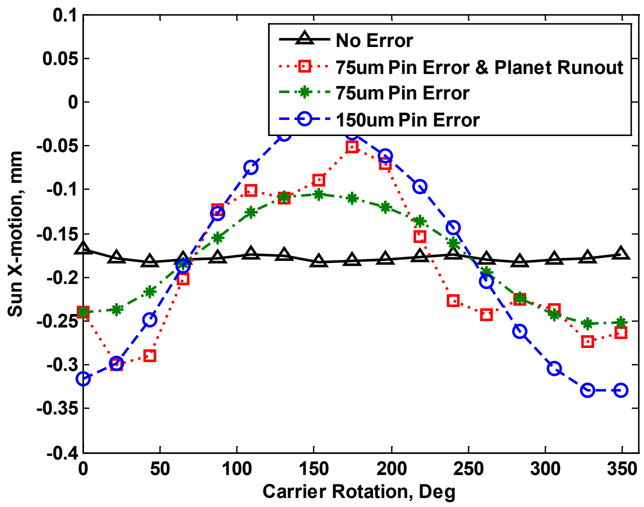

Image 7: Simulated sun motion caused by various manufacturing errors [JA13]

Image 6 shows the computational results of the planetary gear load sharing with various values of main shaft pitching moments. The carrier-bearing operating clearance was 275 μm, and the planet-bearing clearance was 0. The rotor-bending moment increased the load-sharing factor fairly gradually, until it was equivalent to the input torque. Beyond that, a large bending moment induced nonlinear tooth contact and sharply increased the load-sharing factor, particularly for the 10% torque condition. In that case, the gear tooth load contact pattern changed from a slightly disturbed load distribution when the nondimensional bending moment was less than 0.02, to edge loading when larger than 0.02, and eventually reversing contact when larger than 2.

3.4 Imperfections

This study investigated planet pin position error, planet pin looseness, and planet runout. The first two imperfections were measured on the GRC gearbox [AN14]. The sun gear orbiting motion is a combined effect of planetary load sharing, sun spline stiffness, and manufacturing and assembly tolerances. Among all tolerances, the errors that disturb the planet symmetry have a greater influence on sun motion—for example, the planet pin position error, planet runout, and carrier eccentricity. As shown in Image 7, the sun motion for the nominal case without errors had little variation.

Increasing pin position errors induced sun motion at a once-per-carrier revolution with increasing amplitude. There is a linear relationship between pin position error and sun motion. Planet runout adds an additional sun motion component at the frequency of planet rotation. This is superimposed on the motion because of the pin position error. Compared to Image 2, the motion variation without pin errors shown in Image 7 is much smaller, because the modeling tool used to calculate it solved for only the static response of the system and ignored the dynamics.

4 Summary

The recommended modeling approaches for major gearbox and drivetrain components are given in Table 1. Table 2 lists other important modeling elements, including gearbox internal nonlinearities, excitation sources, and manufacturing errors. These recommendations are based on the GRC gearbox and three-point drivetrain configuration, which is representative of a majority of drivetrains. Detailed results that support these recommendations will be included in a separate publication in 2015.

Component	Recommended Approach	Degree of Freedom Requirements
Rotor/hub	Rigid body with lumped weight	N/A
Main shaft	Flexible, FE beams	Determined by convergence study
Main bearing	Stiffness matrices	5 (exclude rotation)
Gearbox housing	Flexible, condensed FE	Determined by convergence study
Planetary carrier	Flexible, condensed FE	Determined by convergence study
Gearbox shafts	Rigid shaft	N/A
Gearbox support	Stiffness matrices	6
Gears	Rigid body with contact stiffness	6
Gearbox bearings	Stiffness matrices	5 (exclude rotation)
Spline	Stiffness matrices	2 (tilting)
Bedplate	Rigid body or condensed FE	N/A
Generator coupling	Stiffness matrices	5 (exclude rotation)

Table 1. Minimum model fidelity for the study of gearbox internal motion

Other parameters	Effects	Priority
Manufacturing tolerance	Affects component motions but has limited effects on loads	Medium
Bearing clearance or preload	Affects component motion and loads. Operational values with operating temperature are recommended.	High
Gear tooth micro-geometry	Affects frequency spectrum of component motions and gear tooth and bearing load distributions	High
Bedplate tilting angle	Causes gearbox axial loads because of gravity	Medium
Gravity	Affects component motion and loads	High
Nontorque loads	Affects component motion and loads	High
Gear mesh stiffness or transmission error	Affects frequency spectrum of component motions	Medium

Table 2. Other important modeling parameters that affect gearbox motion

5 Acknowledgements

This work was supported by the U.S. Department of Energy under contract number DE-AC36-08GO28308 with the National Renewable Energy Laboratory.

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