Gearbox Reliability Collaborative Phase 1 and 2: Testing and Modeling Results

Preprint

J. Keller, Y. Guo, W. LaCava, and H. Link
National Renewable Energy Laboratory

B. McNiff
McNiff Light Industry

To be presented at the 2012 Leuven Conference on Noise and Vibration Engineering
Leuven, Belgium
September 17-19, 2012

Available electronically at http://www.osti.gov/bridge

Available for a processing fee to U.S. Department of Energy
and its contractors, in paper, from:

> U.S. Department of Energy
> Office of Scientific and Technical Information
> P.O. Box 62
> Oak Ridge, TN 37831-0062
> phone: 865.576.8401
> fax: 865.576.5728
> email: mailto:reports@adonis.osti.gov

Available for sale to the public, in paper, from:

> U.S. Department of Commerce
> National Technical Information Service
> 5285 Port Royal Road
> Springfield, VA 22161
> phone: 800.553.6847
> fax: 703.605.6900
> email: orders@ntis.fedworld.gov
> online ordering: http://www.ntis.gov/help/ordermethods.aspx

Cover Photos: (left to right) PIX 16416, PIX 17423, PIX 16560, PIX 17613, PIX 17436, PIX 17721

Printed on paper containing at least 50% wastepaper, including 10% post consumer waste.

Gearbox Reliability Collaborative Phase 1 and 2: Testing and Modeling Results

J. Keller[1], Y. Guo[1], B. McNiff[2], W. LaCava[1], H. Link[1]
[1] National Renewable Energy Laboratory
Golden, CO 80401-3305 USA
email: **jonathan.keller@nrel.gov**

[2] McNiff Light Industry
Harborside, ME 04642 USA

Abstract

One activity of the Gearbox Reliability Collaborative (GRC) investigates root causes of wind turbine gearbox premature failures and validates design assumptions that affect gearbox reliability using a combined testing and modeling approach. Knowledge gained from the testing and modeling of the GRC gearboxes builds an understanding of how the selected loads and events translate into internal responses of three-point mounted gearboxes. This paper presents some testing and modeling results of the GRC research during Phase 1 and 2. Non-torque loads from the rotor including shaft bending and thrust, traditionally assumed to be uncoupled with the gearbox, affect gear and bearing loads and resulting gearbox responses. Bearing clearance increases bearing loads and causes cyclic loading, which could contribute to a reduced bearing life. Including flexibilities of key drivetrain subcomponents is important in order to accurately simulate the measured gearbox response during the tests (using modeling approaches).

1   Introduction

Wind turbine gearboxes are experiencing premature gearbox failures often well short of their intended design life. The cost of gearbox rebuilds, as well as the down time associated with these failures, has significantly elevated the cost of wind energy. Some gearbox problems present in older, smaller turbines still exist in multi-megawatt machines.

The National Renewable Energy Laboratory (NREL) established the Gearbox Reliability Collaborative (GRC) to investigate the wind turbine gearbox design process and identify what may be preventing gearboxes from reaching the expected design life. The GRC project has five major goals: 1) establish a collaborative of wind turbine manufacturers, gearbox designers, bearing experts, universities, consultants, national laboratories, and others to jointly investigate issues related to wind turbine gearbox reliability and to share results and findings, 2) conduct field and dynamometer tests using two redesigned and heavily instrumented wind turbine gearboxes to build an understanding of how selected loads and events translate into internal bearing and gear response, 3) evaluate and validate current wind turbine, gearbox, gear, and bearing analytical tools/models and develop new tools/models as required, 4) establish a database of gearbox failures to aid in root cause analysis, and 5) investigate condition monitoring methods to improve availability [1].

Non-torque loads such as shaft bending and thrust have historically been assumed to be uncoupled from the gearbox. In a three-point suspension drivetrain, a main shaft bending moment causes radial reaction forces that pass through the gearbox. Ideally, these radial reactions pass from the main shaft, through the gearbox carrier bearings, the gearbox housing, the gearbox trunnions, and into the bedplate [1]. However, these reaction forces could cause gear and bearing misalignment and change the loads in the low-speed stage (LSS) of the gearbox. The GRC has measured a bending moment on the main shaft that is mainly caused by the rotor weight and aerodynamic forces. This bending moment has the same order of

magnitude as the rated torque [1] [2]. Other wind turbine designs adopt a two main bearing configuration to reduce such non-torque loads; however, these turbines still have a small amount of bending moment transmitted to gearboxes. The bending moments transmitted from the rotor are fundamental vibration sources for the gearbox LSS and affect its internal responses [3]. When input torque is low, the bending moment plays a dominant role in gearbox responses [3]. Aerodynamic forces on the blades result in thrust on the main shaft. In three-point drivetrain suspensions, this thrust loading has historically been assumed to be transmitted through the main bearing, routing the forces through the nacelle to the tower without affecting the gearbox. Any main shaft motion due to displacement of the bearing or nacelle is thought to be compensated by the gearbox trunnion mounts. However, due to the axial play and low axial stiffness of the main bearing, this shaft thrust causes drivetrain axial motion. This paper investigates these non-torque load design assumptions by studying the influences of main shaft bending moment on the LSS internal load sharing and load distribution, and the influences of shaft thrust on drivetrain axial motions.

Identification of the necessary component flexibilities is paramount to reducing the cost of system dynamics modeling, and establishing the correct model reduction procedure is critical for maintaining accuracy. Gearbox housing flexibility has generally been considered necessary due to its influence on shaft bore and ring gear misalignment [4]. Carrier flexibility is also considered necessary due to its effect on planet pin alignment [5]. However, a robust assessment of the accuracy of various modeling techniques versus computational effort has not been conducted, nor has the influence of flexibility of other subcomponents been analyzed with test data correlation. In this paper, a guideline for the required modeling fidelity of wind turbine drivetrains is also provided.

2   Field and Dynamometer Testing

The GRC gearbox is mounted in a three-point suspension drivetrain configuration with supports at the main bearing and gearbox trunnions. The gearbox includes a helical planetary gear and two parallel stages [1][6][7].

Gearboxes are designed to transmit input torque to the output shaft, with reacting forces that are transferred through the internal bearings and gearbox trunnions. Non-torque loads have not been included in the dynamometer test validation process in the past. However, for wind turbines with three-point gearbox mounting, non-torque loads could be a major contributor to gearbox problems. In order to validate this design assumption, non-torque shaft loads and dynamic load application capabilities [8][9] are included in the NREL dynamometer testing as shown in Figure 1.

Figure 1: GRC static non-torque loading arrangement (*NREL/PIX-19222*)

2.1   Internal Response to Bending Moments

Main shaft bending loads up to 180 kiloNewton-meter (kNm) (70% of the rated torque) were applied to simulate bending moments caused by rotor overhung weight. Results indicated that bending moments affect tooth contact patterns in the low-speed stage, which could shorten gearbox life [8]. Field testing has shown larger than anticipated variations in upwind and downwind load share between the paired planet bearings compared to the dynamometer testing, as shown in Figure 2. Upwind and downwind bearing loads are out of phase in the dynamometer tests. The planetary load sharing factor measured in the field and dynamometer tests varies with shaft rotation well beyond the ideal value of 1.0. Fully representative dynamic moment and torque variations measured in the field are not reproduced in typical dynamometer testing, which may contribute to the differences in the field and dynamometer testing. These variations can often be reproduced in the dynamometer, but limitations in the test equipment must be considered [8]. The effects of bending moments on tooth load distribution of the ring gear are shown in Figure 3. Bending moments increase tooth edge loads significantly. Tooth load distribution varies when the shaft rotates, which suggests the cyclic loading on gear teeth.
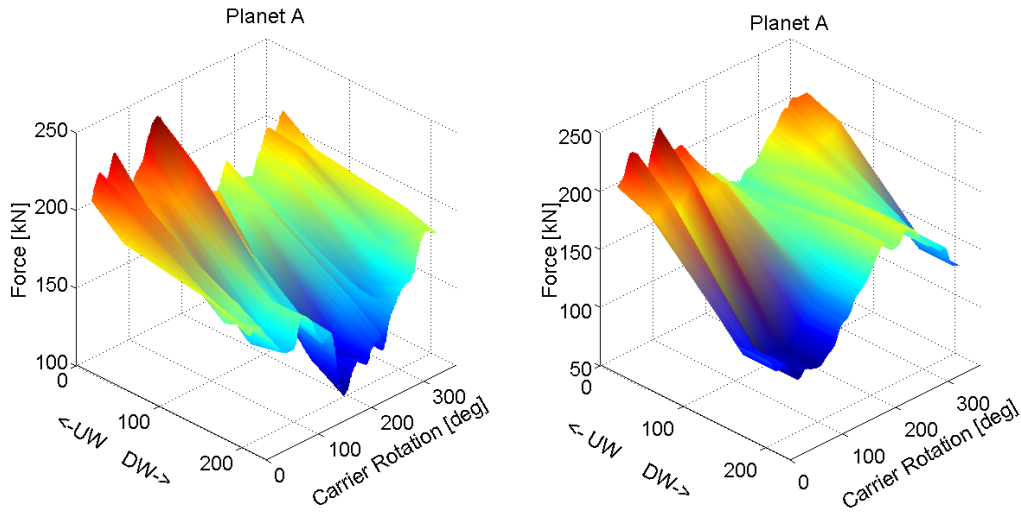

Figure 2: (left) Field and (right) dynamometer upwind and downwind planet-bearing loads over a carrier rotation (*Source: NREL*)
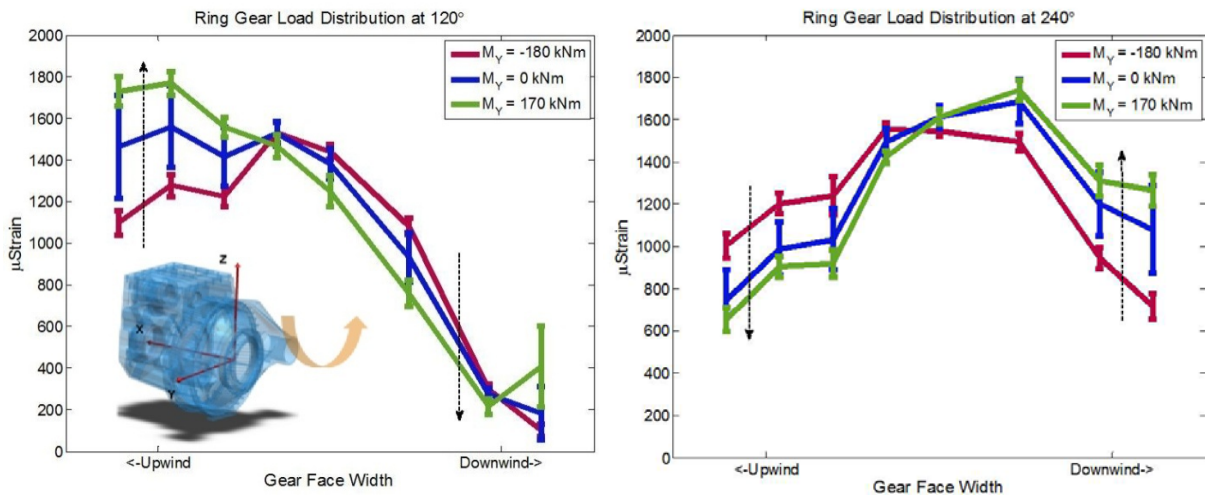

Figure 3: Ring gear load distribution at 120 (left) and 240 (right) degrees with various bending moment (*Source: NREL*)

3

2.2   Internal Response to Thrust Load

Thrust corresponding to the equivalent of 20 meters per second (m/s) wind was applied using the non-torque loading mechanism in the NREL dynamometer (Figure 1) with various levels of torque to evaluate its effects on gearbox motions. Figure 4 (left) shows that the main shaft moves relative to the bedplate 800 micrometers (μm) with 100 kiloNewtons (kN) thrust. The main bearing, a double row spherical roller bearing, is assumed to react the thrust but this axial motion is still allowed. Figure 4 (right) shows that the carrier moves up to 63 μm relative to the gearbox housing, a deflection similar to the 60 μm of axial clearance in the planet bearings. The planets move downwind approximately the same amount as the planet carrier, suggesting that they are constrained to move together by the planet bearings. This indicates that the axial clearance in the planet bearings has been removed, and that forces between the ring gear and planets act to restrict motion. The carrier axial motion of 63 μm has a negligible effect on gear loads because it is much smaller than the gear facewidth of 230 μm. It does not create meaningful axial loads on carrier and planet cylindrical roller bearings because there is sufficient clearance and low friction in gearbox trunnions. However, the drivetrain internal axial motions and loads might become significant when thrust is reversed or clearance in the trunnions is small. Furthermore, the planets aren't designed to handle significant axial loads. Axial loads beyond about 10% of the radial loads can cause end contact between the rolling elements and flange faces, misaligning the inner and outer races and potentially causing unanticipated damage [1]. The aforementioned discussions suggest that non-torque rotor loads affect internal responses of three-point suspension gearboxes.
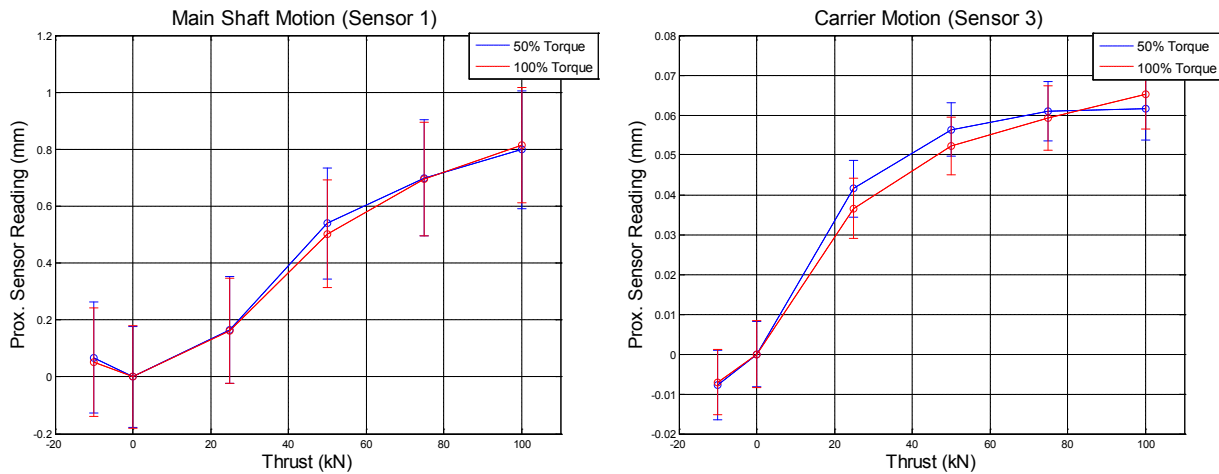

Figure 4: Axial motion of the main shaft relative to the main frame (left) and carrier motion relative to the gearbox housing (right) versus thrust (*Source: NREL*)

3   Modeling and Analysis

NREL GRC and modeling partners have adopted a selection of drivetrain programs and in-house codes with different levels of complexity to simulate wind turbine drivetrain dynamics. These tools consist of lumped-parameter, rigid multi-body, flexible multi-body, and finite element models.

3.1   Importance of Model Fidelity

The GRC conducts model-to-model comparisons to identify the critical parts to include in design specifications and considerations. The level of model fidelity is important in predicting the responses of gearbox components as captured in testing. A trade-off exists between simulation time and accuracy. Four drivetrain models with different levels of model fidelity are listed in Table 1 and are compared against the

dynamometer testing data. As shown in Figure 5, Model 1 assumes that the housing, carrier, gear bodies, and bedplate are all rigid. The torsional stiffness of each stage is represented with a spring placed in the midsection of each shaft. In Model 2, finite element representations of the planet carrier and housing are imported as flexible bodies into the model using modal condensation [10]. In Model 3, flexible shafts are included. Two levels of bearing model fidelity are considered: A) a diagonal stiffness matrix, and B) a diagonal stiffness matrix with bearing clearances in the planetary stage.

| | | Drivetrain Model | | |
		Model 1: Rigid 6 Degrees of Freedom (DOF)	Model 2: Flexible Housing and Carrier	Model 3: Fully Flexible
Bearing Model	A. 6x6 Diagonal Matrix	M1A	M2A	
	B. 6x6 Diagonal Matrix with clearance	M1B	M2B	M3B

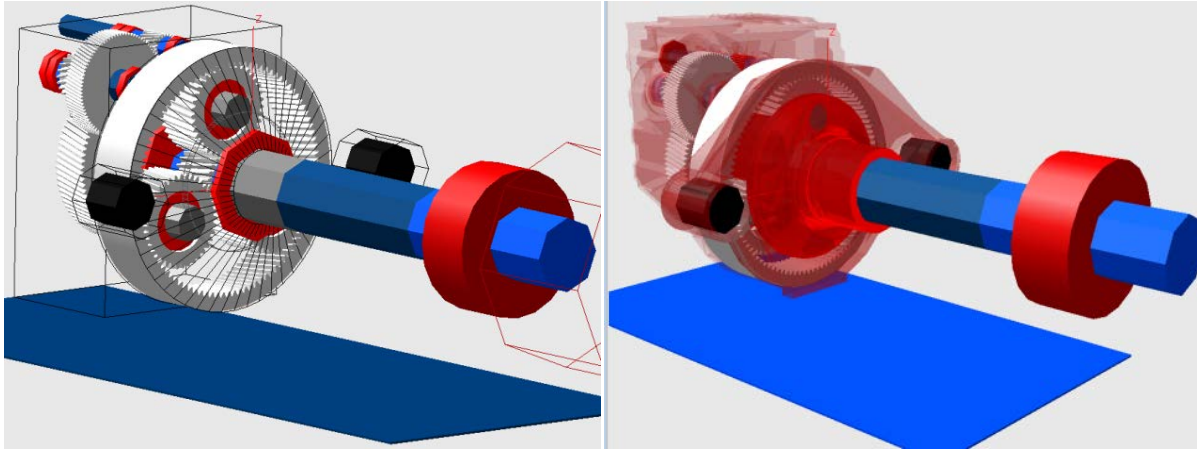**Table 1: Definitions for the model fidelity comparison**

Figure 5: Model 1 (left) and Model 2 (right) of the gearbox (*Source: NREL)*

Planet C upwind loads are shown in Figure 6 (left). Model 1A and 1B prediction of the once-per-revolution cyclical load sharing of the bearings compares well with test results with a mean value prediction within 10%–20% of the test mean load. Model 2 predicts the maximum load within 5% for the upwind and downwind bearings. Both models over-predict the mean downwind bearing load by 20%. A better characterization of the rotor-side bending loads and pin-to-carrier stiffness would rectify this discrepancy. Bearing clearances in the planetary stage have a negligible effect on the output value. A computational effort comparison in terms of number of degrees of freedom (DOF) and computation time among these models is shown in Figure 6 (right). There is a small improvement in accuracy between Model 1 and 2, whereas computation time increases by nearly 400%. For test cases that show similar upwind and downwind loading of the planet bearings like the field test shown in Figure 2 (left), the rigid model is therefore an effective option. Because of the large numbers of modes retained for the carrier and housing in Models 2 and 3 (with flexible shaft and planet pins), and the low number of modes retained for beam elements in Model 3, there is only a 26% change in the number of DOF between Model 2 and 3.
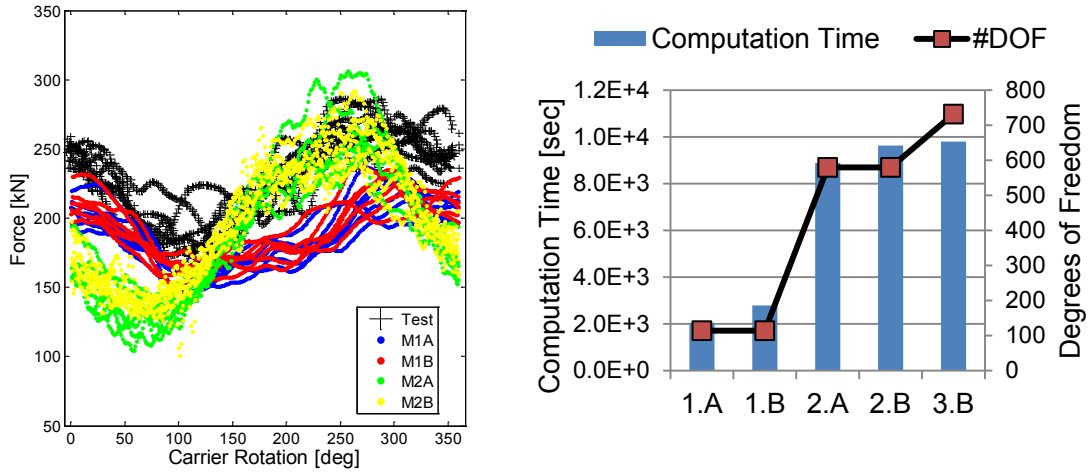
Figure 6: Field test planet C upwind bearing load comparison (left) and computation time comparison of different models (right) (*Source: NREL*)

In summary, model flexibilities in the planetary carrier, support structure, gears, and bearings are important. Planetary load sharing varies within each revolution; thus, the planetary elements should be modeled together to capture these effects [10].

3.2   Importance of Bearing Clearance

In three-point mount drivetrains, the carrier bearings support the carrier weight and the non-torque loads caused by the rotor overhung weight and aerodynamic forces. Ideally, carrier bearings with high load capacities carry the majority of these loads so that only a small fraction of the non-torque loads are transmitted into the gears. Therefore, in the ideal case, the gear teeth only carry the input torque. Figure 7 compares the tooth load distribution with and without bearing clearance in the carrier supports at the rated torque. Pure torque and gravity effects are considered. Without clearance, a time-invariant, parabolic tooth contact pattern is present. When clearance is included, the ideal tooth load distribution is disturbed and the contact pattern varies periodically because the gear teeth carry the carrier weight and input torque simultaneously. Gravity induces excitations in the rotating carrier frame, resulting in cyclic loading on gear teeth as shown in Figure 7 (right). This cyclic loading becomes significant when bending moments are included. Increased loading on bearings may be a contributing factor to shorter bearing life than expected.
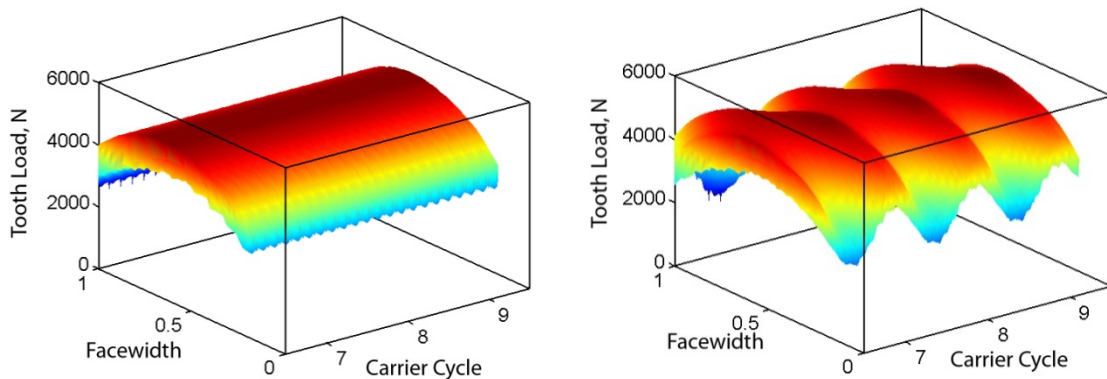

Figure 7: Tooth load distribution on the first sun-planet mesh without (left) and with (right) carrier bearing clearance [3]

3.3   Importance of Non-Torque Loads

GRC results indicate that non-torque loads cause increased gear tooth edge loading and unequal load sharing among planet gears [3][5]. The simulated tooth load distribution of the first ring-planet mesh at 10% torque is shown in Figure 8 (left). The gear tooth face has edge contact towards the upwind end and partial contact loss towards the downwind end. Micropitting on the upwind face of the ring gear teeth believed to be caused by this edge contact was observed after the experiments and is shown in Figure 8 (right). This suggests that bending moments are transmitted into the gear meshes and is an important step towards validation of the drivetrain model. Results also show unequal load sharing between upwind and downwind support bearings caused by bending moments, especially at low torque.
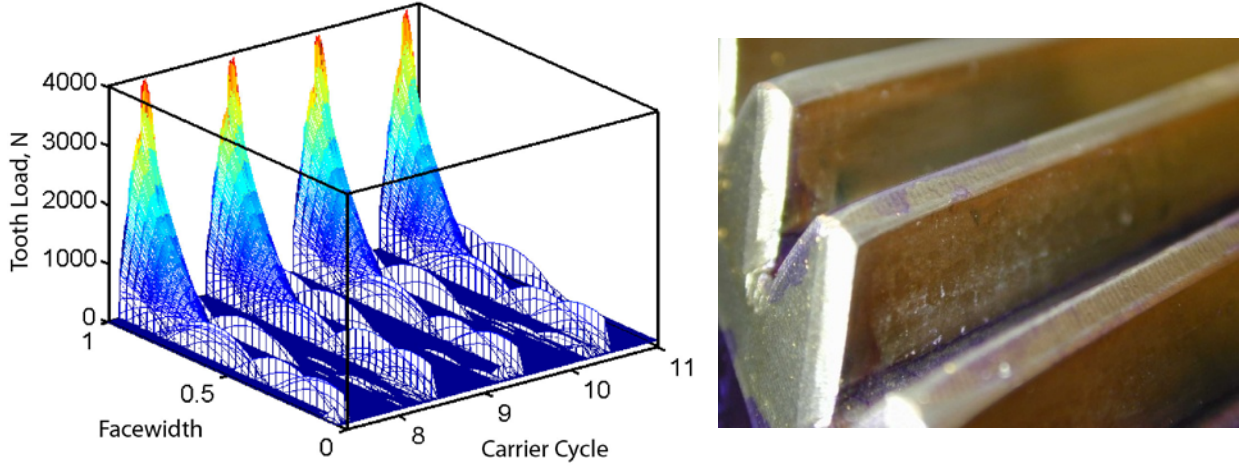

Figure 8: Tooth load distribution when input torque equals 10% of the rated torque (left); Micropitting on ring gear teeth caused by edge loading (right)

The effect of bending moments on planetary gear load sharing depends upon input torque. A nondimensional quantity M_y/T_{in} is introduced to address the combined loading, where M_y is the bending moment and T_{in} is the input torque. Figure 9 shows the planetary gear load sharing at 10% torque and rated torque. The load-sharing function, k_γ, is defined as:

$$k_\gamma(t) = \frac{1}{\text{mean}\left\{\text{mean}\left[f_p^i(t)\right]\big|_i\right\}\big|_t} \max\left[f_p^i(t)\right]\big|_i, \quad i = 1,...,N \tag{1}$$

Where f_p is the bearing force on the ith planet and N is the number of planets. The load-sharing function is a function of time over one revolution of the carrier. In order to compare steady-state solutions at different torque levels, a maximum load-sharing factor, k_γ^*, is defined as

$$k_\gamma^* = \max\left[k_\gamma(t)\right]\big|_t \tag{2}$$

The load-sharing factor at 10% torque is higher than at rated torque because the influence of gravity on the load-sharing factor is relatively higher at low-input torque [1]. Adding bending moment increases the load-sharing factor nearly linearly when the moment is moderate. The relation between the load-sharing factor and bending moment becomes nonlinear when M_y/T_{in} is large because of nonlinear tooth contact caused by the high bending moment. In addition, the bending moment affects the tooth load distribution. When the bending moment increases, the gear tooth load contact pattern changes from a slightly disturbed load distribution when $M_y/T_{in} < 0.02$ to edge loading when $M_y/T_{in} < 2$ and eventually reversing contact when $M_y/T_{in} > 2$ at 10% torque. Shaft bending moment affects planetary gear load sharing and load distribution, which agrees with the observations during the experiments as shown in Figure 2 and Figure 3. Both testing and modeling results suggest that non-torque loads affect gearbox internal responses.

Therefore, the design assumption that non-torque rotor loads are uncoupled from the gearbox is not valid for three-point mount gearboxes.
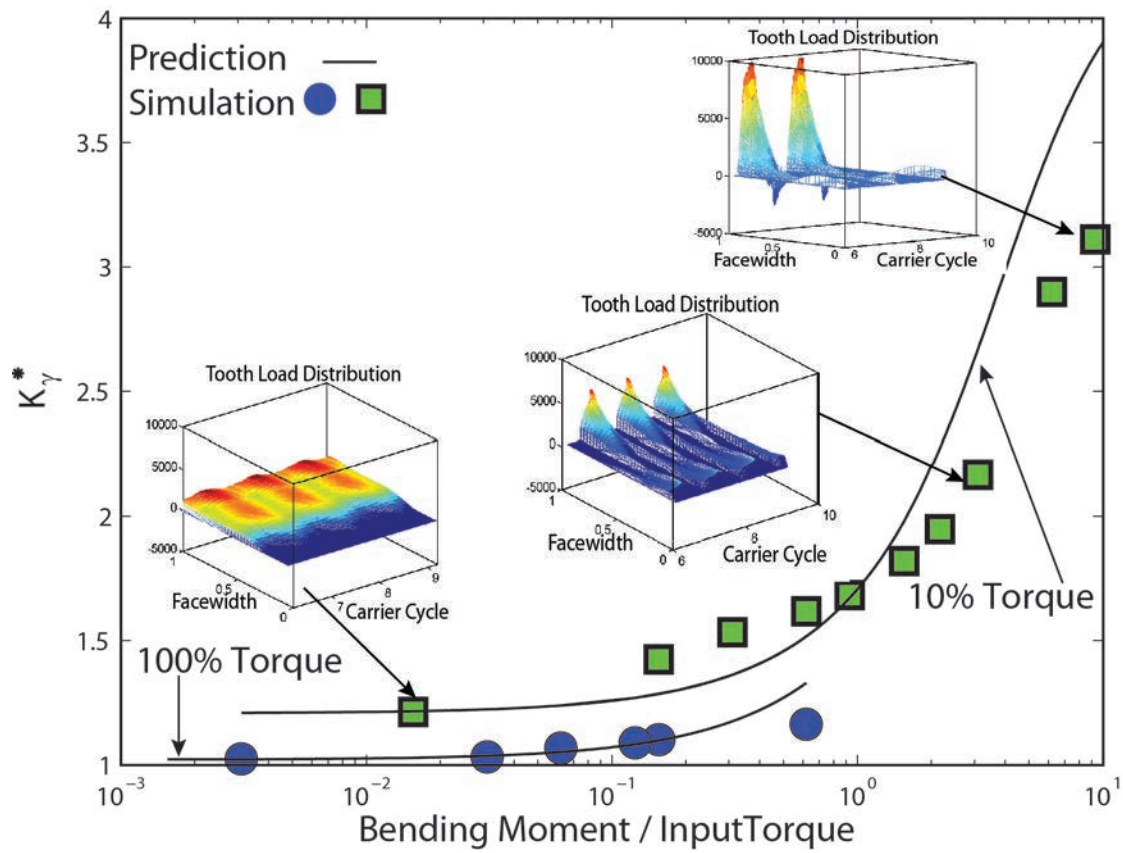

Figure 9: Load-sharing sensitivity to bending moment [3]

4   Conclusions

The GRC uses a combined testing and modeling approach to investigate root causes of gearbox premature failures and design assumptions that contribute to reduced gearbox life. Key findings of this work include the following:

- A fully flexible gearbox model simulates the dynamometer loading conditions very well. However, in the field measurement data when planet-bearing loads are in phase, a rigid model should be considered because of the advantage in computational effort.
- Clearance in carrier bearings affects the gearbox load sensitivity to non-torque loads. It disturbs the symmetry of planets, leading to asymmetric bearing loads.
- Bending moments affect gearbox internal response and play an important role in gearbox reliability. Bending moments induced by the blades and hub weight cause unequal loads between upwind and downwind carrier and planet bearings. The upwind planet bearings could develop cumulative Hertzian fatigue damage due to loads that are high in magnitude and cyclical in nature. These unequal bearing loads also lead to gear tooth edge loading and potential fatigue damage failures.
- Main shaft axial motion caused by thrust has limited influence on gearbox internal loads. However, with excessive friction or inadequate clearance in the trunnion supports, the planet carrier could be loaded axially due to its limited axial play, in particular when thrust is reversed.

5   References

[1] H. Link, W. LaCava, J. V. Dam, B. McNiff, S. Sheng, R. Wallen, M. McDade, S. Lambert, S. Butterfield and F. Oyague, "Gearbox reliability collaborative project report: findings from phase 1 and phase 2 testing," NREL technical report: NREL/TP-5000-51885, 2011.

[2] J. V. Dam and E. Overly, "Torque measurements on GRC test turbine 1," NREL, 2007.

[3] Y. Guo, J. Keller and W. LaCava, "Combined Effects of Input Torque, Non-Torque Load, Gravity, and Bearing Clearance on Planetary Gear Load Share in Wind Turbine Drivetrains," in *AGMA fall technical meeting*, 2012.

[4] F. Oyague, "NREL Gearbox Reliability Collaborative Analysis Round Robin," NREL/CP-500-45325, 2009.

[5] F. Oyague, D. Gorman and S. Sheng, "NREL Gearbox Reliability Collaborative Experimental Data Overview and Analysis," in *Windpower Conference and Exhibition*, Texas, 2010.

[6] B. McNiff, J. V. Dam, J. Gorman and H. Link, "Gearbox Reliability Collaborative Phase I Test Plan," NREL Report No. TP-500-47842., Golden, CO, 2012.

[7] J. V. Dam, "Gearbox reliability collaborative bearing calibration," NREL/TP-5000-47852, 2011.

[8] W. LaCava, B. McNiff and J. van Dam, "Comparing In-field Gearbox response to Different Dynamometer Test Conditions," in *AWEA, Windpower 2011. NREL Report CP-500-51690.*, Anaheim, CA, 2011.

[9] P. Veers, H. Link, B. McNiff, W. LaCava, S. Lambert and R. Wallen, "Augmenting dynamometer testing to improve wind turbine gearbox design validation," in *EWEC*, Brussels, 2011.

[10] W. LaCava, Y. Xing, Y. Guo and T. Moan, "Determining wind turbine gearbox model complexity using measurement validation and cost comparison.," in *EWEA*, 2012.