Gearbox Reliability Collaborative Analysis Round Robin

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Acronyms and Abbreviations

AGMA	American Gear Manufacturers Association
CLC	Calibration load case
FAST	Fatigue Aerodynamics Structures and Turbulence
FEA	Finite element analysis
GRC	Gearbox Reliability Collaborative
HS	High speed
IEC	International Electrotechnical Commission
INP_SH	Input shaft (main shaft)
IMS_SH	Intermediate speed shaft
LOA	Length of action
LS_SH	Low speed shaft
NREL	National Renewable Energy Laboratory
NWTC	National Wind Technology Center
PL	Planet gear
RING	Annulus or ring gear
SH	shaft
ST	Stage
V_{ave}	Annual average wind speed at hub height
V_{ref}	Referenced wind Speed averaged over ten minutes

Introduction

The National Renewable Energy Laboratory's (NREL) National Wind Technology Center (NWTC) has embarked on the task of determining the causes and loading conditions that result in the premature failure of wind turbine gearboxes. The NWTC approaches this challenge by bringing together the parties involved in the gearbox-design process to achieve the common goal of the improving the lifetime of gearboxes. This is being accomplished through the Gearbox Reliability Collaborative (GRC).

In addition to this collaborative effort, the GRC seeks to achieve its goal by exploring three avenues of research: drivetrain numerical analysis and modeling, full-scale dynamometer testing, and field testing. These different avenues are correlated and iterated to obtain the closest representation of actual load behavior in a controlled environment.

The numerical analysis plays a key role in the design process of the gearbox; thus, the "Gearbox Reliability Collaborative, Analysis Round Robin" presented here seeks to examine various analytical approaches to gain insight in the validity and level of fidelity required to satisfy the design practice. The analysis is based on modeling techniques that range from simple torsional models to models that account for gear tooth contact and gear tooth load distribution, as well as bearing compliances and roller load distribution. The results from the GRC Analysis Round Robin will be validated with experimental data, and eventually will be a valuable data set that can be used for the validation of analysis tools.

Industry designers typically have used different design approaches, but there is some consensus based on AGMA 6006 [1]. It would be useful to establish a definition of this consensus as a baseline and then compare it against a broad range of analytical approaches. The GRC Round Robin could provide an objective database of analytical results as well as experimental validation to aid the evaluation of the consensus.

The main focus of this document is the analytical approaches overseen by the Round Robin. It also gives a brief overview of the GRC and describes the gearbox redesigns, instrumentation, and testing, as well as the GRC Round Robin. Next is a description of the different analytical approaches presented by the participants, and discussion of the levels of fidelity implemented. The load cases and the parameters provided to the participants are presented along with the results obtained.

Gearbox Reliability Collaborative—Overview

This section presents a brief overview of the GRC effort. It describes the steps followed to bring the gearbox up to today's design practices and manufacturing standards. It also discusses the instrumentation utilized and experimental effort. Lastly, it gives a description of the GRC Analysis Round Robin.

Gearbox Redesign

The GRC gearbox was redesigned so that its configuration and characteristics are consistent with the current industry standards. Additionally, the redesign reflects the

configuration of a 1.5-MW gearbox, which is an integral part of the currently installed fleet. The redesign of the gearbox also enables NREL to provide the GRC participants with the design information needed to perform the analysis. This is an important aspect of the GRC, because it is quite difficult to obtain manufacturing drawings, material properties, and solid models due to the proprietary nature of the information. Further, one of the goals of the GRC is to produce a reference data set that is substantiated by reliable data and can be used for the validation of new analytical tools.

The redesigned gearbox included the addition of a floating sun configuration. This configuration allows the sun to move freely between the planets, which improves the planet load-share characteristics of the gearbox. Bearing configuration was changed in the planets, abandoning the old standard of spherical roller bearings. The bearing configuration of the parallel shaft stages also was updated to use tapered roller bearings. The internal lubrication system was upgraded from a splash-lubricated system to a jet-lubricated system adding lubricant distributions to each individual planet. The filtration system also was updated to reflect the highest standard in the industry by the addition of a self-standing kidney loop and desiccant filters. The gearing microgeometry was updated using up-to-date analytical tools to compensate for deflections and misalignments.

Instrumentation

The redesigned and rebuilt process noted above was performed for two independent gearboxes. The first is the test subject for the field and the other is used in the dynamometer. Both gearboxes are heavily instrumented, and include more than 150 channels to capture the internal behavior of the gearbox. Among the parameters to be evaluated are the bearing roller load distribution in the planet bearings as well as the planet bearing load share. Planet bearings are a common source of failure in wind turbine gearboxes [2]. These measurements will help validate the analytical models and shed some light on the uncertainty associated with planet bearing behavior.

The bearing raceway temperature differential also will be monitored, which will help in calculating bearing operating clearances—an important component in the bearing-life calculations. Sun motion with respect to the planet carrier shows the behavior of the sun as it adjusts for the changing planet load share among the planets. Planet carrier misalignment will be measured and its effects on the planet load share will be evaluated through measuring the root stresses on the planet ring mesh. These parameters represent a portion of what is evaluated in the experimental side of the GRC, and will be used to validate the analytical branch of the GRC.

Dynamometer and Field Testing

The GRC testing will operate in the field and in the dynamometer in parallel, allowing an important interaction and cross feeding between both tests. This interaction will help determine how the field loading can be simulated in the dynamometer. The control environment of the dynamometer will serve as an important validation tool for the analysis, and the parallel field test will ensure that the validation is representative of the loading experienced on the field. Entire load cases generated in the field will be fed to the validated models, so that real turbine behavior can be simulated.

GRC Analysis Round Robin

The GRC Analysis Round Robin is intended to evaluate the different analytical tools and approaches utilized in the development and design of horizontal axis wind turbine drivetrains. The advantage of the round-robin approach is that it brings together many experts. This pool of expertise foments progressive discussion surrounding the assumptions and analytical discrepancies produced by the round robin. Additionally, the round-robin configuration allows the evaluation of different analytical approaches as well as different levels of complexity, exploring a broad gamut of tools used in the industry. Lastly, the participation of industry partners' consulting companies as well as software development companies ensures that the feedback of the knowledge gained by the GRC will be rapidly absorbed by the industry, thus inherently improving drivetrain reliability.

The GRC Round Robin utilizes the GRC drivetrain to provide its participants with a comprehensive characterization of the gearbox and other drivetrain components. The participants perform their analyses using this information. All analysis results are compared, based on a platform provided by GRC. The platform includes parameters with a narrow definition and provides assumptions to avoid discrepancies in the results as well as specific load conditions.

The parameter comparison (noted above) initially evaluates parameters of low complexity which are independent of other parameters. Subsequently, the parameter complexity is increased to evaluate the capabilities of the higher-fidelity models. The progressive comparison approach, as well as the different levels of model fidelity, result in a comprehensive evaluation of the different analytical tools.

The parameters are obtained under certain loading conditions which follow a progressive load spectrum. Simple load cases—such as steady state or static loading—are initially evaluated. Subsequently, load cases of more complexity—such as dynamic and transient events—are evaluated. The combination of the progressive modeling and loading conditions prevents rootless discrepancies on highly compounded models with complex dynamic transient loading.

GRC Round robin Participants

The GRC participants that have provided results for the round robin are the following CENER, DONG Energy, GE Transportation, KiSSsoft, Ricardo Inc., and Romax Technology. The number of participants involved in the round robin is more extensive than the previously mentioned. However participants that are currently building the models or tuning already existing models to provide agreeable results have not been included in the list. Nevertheless we feel confident that the participants involved in the project are representative of the analytical tools currently used in the industry.

Analysis Description

This section describes the different analytical approaches applied to the different areas of the gearbox. A brief description of the level of fidelity implemented by the different

participants also is presented. Figure 1 shows an exploded view of the different components of the GRC gearbox.
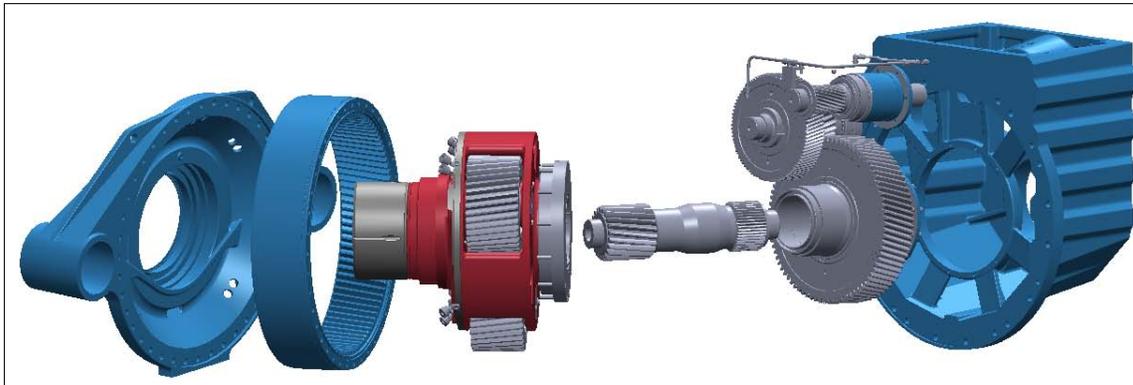
Figure 1. Exploded view of the GRC gearbox

Preliminary Analysis

As an initial step, some participants generate a free-body diagram. This approach gives a first glimpse of the load path traveling through the gearbox. This initial calculation is fairly simple; however, it provides a good guideline of the loading and the expected results throughout the analysis. As the analysis increases in complexity a number of assumptions must be made. The initial free-body calculation becomes a reference to validate these assumptions. Figure 2 shows the layout of the free-body diagram for the gearbox.
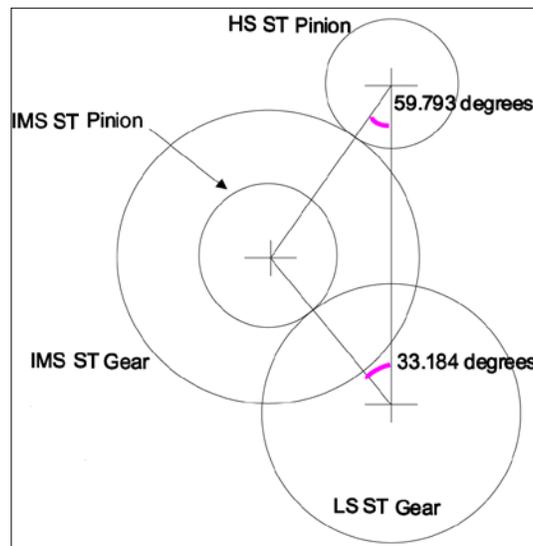
Figure 2. Gearbox free-body diagram (GE Transportation)

Gearing Analysis

The simplest approach to the gearing behavior used by the participants included the basic calculation of the tooth interaction forces. These values are dependent on the helix angle and the pressure angle, which is assumed to be constant. This type of approach is used to obtain the loading conditions in the previously mentioned free-body diagram.

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Gear mesh stiffness also has been evaluated using a number of techniques. The simplest assumes that the gear mesh behaves as a cantilever beam. This approach assumes a constant mesh stiffness throughout the mesh cycle [3]. Additional modeling uses two-dimensional analytical modules to estimate the gear mesh stiffness. The gear behavior is dependent on the gears' center distances and the gear tooth characteristics. This type of modeling approximates the influence of microgeometry by including the tip relief characteristics of the profile. This approach approximates the gear mesh stiffness through the mesh cycle with a parabolic function [4]. More comprehensive models also have been used; they generally were finite element analysis (FEA) based, however they specifically were tailored and designed for the gear analysis. This type of modeling encompasses all the corresponding tooth geometries, including lead modification as well as crowning and tip relief. This type of analysis generates a series of stiffness values throughout the mesh cycle, giving an accurate representation of the gearing behavior.

Gear contact stress was evaluated by some of the participants; however, there was no simplified modeling to obtain the stresses. Generally, FEA-based and highly specific codes were used for this task. The difference in the analytical approach is in the integrated nature of the modeling. Some of the participants presented decoupled models and others presented a fully coupled modeling approach. Gear contact stress is highly dependent on the misalignments generated by the compliance of the gearbox components.

The decoupled approach brings together different tools that calculate the misalignment generated by the various components of the gearbox. The contributions of the components are added to obtain the effective misalignment experienced by the gear mesh. The information is input into the stress-specific codes to calculate the contact stresses. The downfall of this approach is that different codes use different sign conventions and coordinate systems. Consequently, the addition of the misalignment should be evaluated carefully to ensure that the final misalignment used for the calculation is correct. Some coupled codes also were used in this approach. The integration of a number of tools into a single code eliminates the need for the manual addition of deflections. Generally, to decrease computational time this approach reduces the fidelity of some of its components. Nevertheless, some of the participants have generated fully coupled models with very high fidelity. Figure 3 shows a fully coupled model capturing the entire gearbox behavior and able to calculate gear tooth contact stresses.
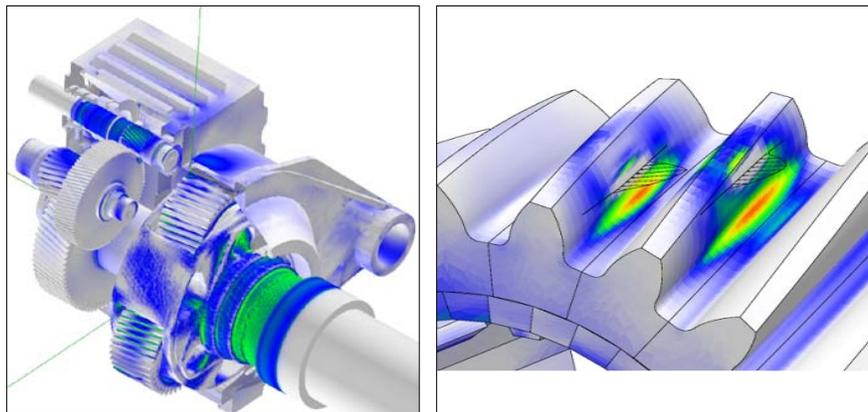

Figure 3. Detailed integrated model (Ansol)

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Shaft Analysis

Shaft modeling has taken several approaches. The simplest approach has ignored the shaft deflection, assuming that the shafts are infinitely stiff. A simplified version of the shaft behavior also has been included by considering only the torsional contribution of the shafts. Generally, the assumption of a linearly elastic body has been used and the shaft geometry simplified such that the torsion of circular bars mechanics of solids can be used. The more comprehensive analysis used the finite element analysis of the shaft to predict the torsional and bending behavior. Figure 4 shows the combined torsional compliance of the low speed shaft components.
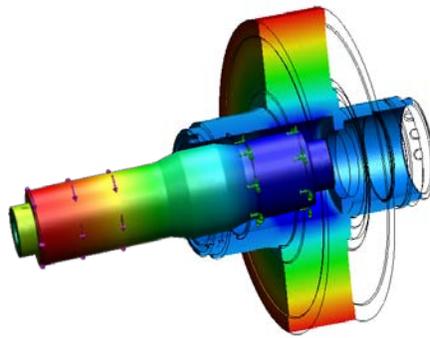
Figure 4. Shaft torsion FEA (NREL)

Planet Carrier Analysis

Due to its complicated geometry, the planet carrier was not simplified so that a basic analytical calculation could be used. Most of the analysis used FEA to account for the deflections generated at the planet carrier. These deflections are very influential in the evaluation of the microgeometry and the tooth stresses. Due to the dependence (noted above) the misalignment is measured at the planet pin so that its translation to planet misalignment easily could be taken into account.

The main difference in the analytical approach was in the assumptions of the models. For instance, some analyses modeled the planet carrier pin interactions by the use of contact elements. This approach resulted in a more compliant planet carrier with greater misalignments. Others assumed an infinitely stiff joint increasing the stiffness of the components. Figure 5 shows the reference planes used to measure the planet pin misalignment (on the left) and a linear approximation of the pin misalignment (on the right).
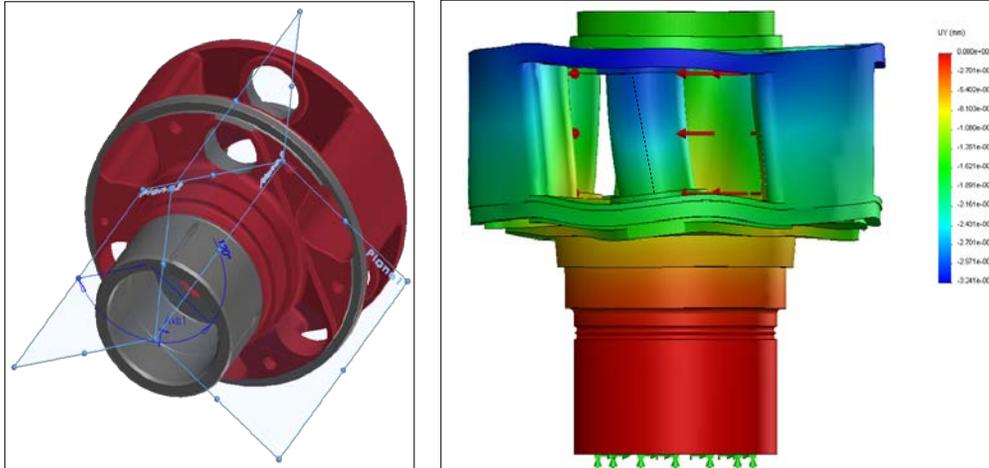
Figure 5. Planet carrier pin misalignment (NREL)

Housing Analysis

The main approach taken to simulate the housing behavior has been the FEA method. Housing deflections also are of great importance because they produce misalignment that affects the gearing performance. Additionally, the interaction between the housing and the bearings was modeled by some participants. This interaction can have a significant effect on the internal clearance of the bearings, which is a determinant factor in bearing life. Furthermore, misalignment (discussed above) also has a significant impact on the bearing life. For the dynamical behavior of the gearbox housing, the participants performed virtual modal simulations which are exported to multibody dynamics codes to capture the dynamic behavior of the gearbox. Figure 6 shows housing stress characteristics for the housing underrated torque.
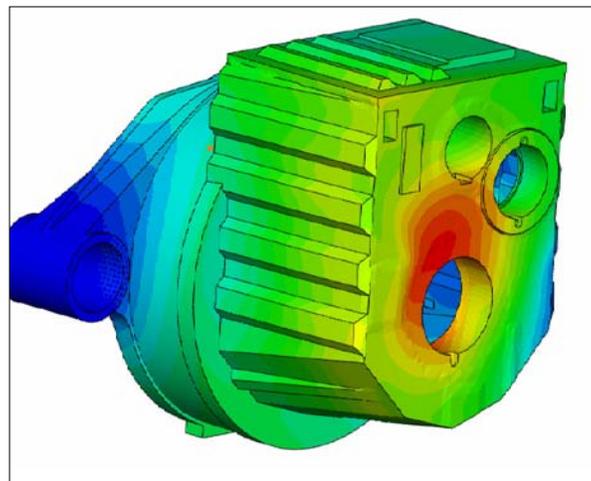
Figure 6. Housing analysis (Ricardo)

Bearing Analysis

Modeling the bearing behavior is quite complex. Bearing response is dependent on factors including the loading conditions, temperature gradients, misalignment, rotational velocity, preload and operating clearance, off-roundness, and defects of the housing. For the bearing models to account for the previously noted parameters, the entire gearbox generally is modeled. Simplified gearing gives estimated loading to the shaft and housing, which interacts with the bearings. The bearing portion of the models generally is of very high resolution with sophisticated contact elements that account for the interaction of all the bearing components. These models may be used to calculate a linearized bearing stiffness, which may then be applied to simpler dynamic models (examples of this type of modeling can be found in reference [5]), this simplification is appropriate for transient time domain simulations. The detailed non-linear models are required for calculating accurate bearing life and roller contact stresses. Figure 7 and Figure 8 show the shaft bearing configuration, roller stress characteristics and roller contact stress for all rollers. Here the calculation accounts for bearing misalignment, variation in contact angle with misaligment, increase in contact area and number of rollers contacting with load and the microgeometry of the bearing, such as roller crowning. When modeling the whole gearbox the contact and deflection calculation incorporates the elasticity of the surrounding structure, this is important to correctly predict misalignment, which directly effects the calculation for stress and component life [6]
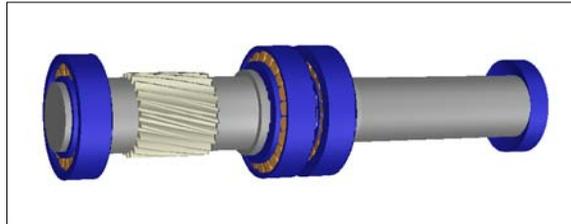

**Figure 7. Shaft bearing configuration**
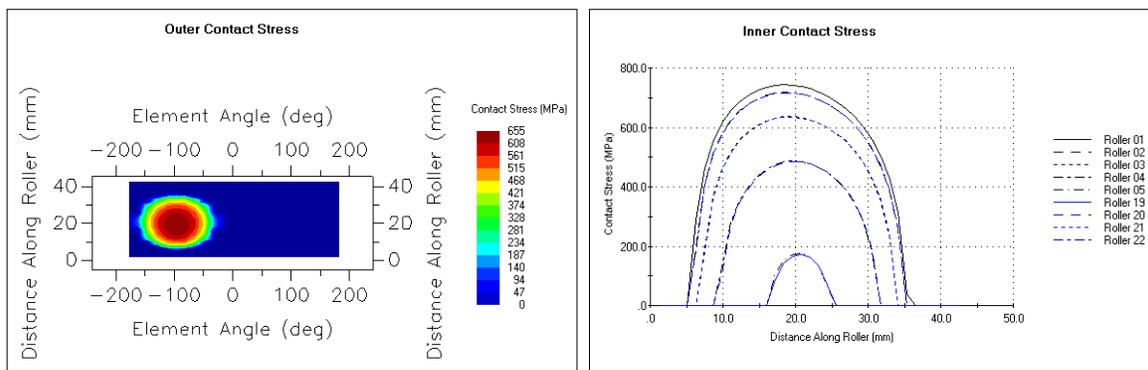
Figure 8. Bearing analysis illustration (Romax)

Loading Analysis and Results Comparison

This section briefly describes the load cases that are used in the GRC Round Robin. The load cases range from simple static load cases to complex and dynamic load cases. The

simple static-load cases are used as a calibration tool for the analysis. More complex load cases follow the guidance of current standards used in the traditional design process. Also discussed are load cases including time series from field measurements which are representative of loading conditions experienced by the turbine under normal operation.

A brief description of the parameters evaluated in the GRC Round Robin effort also is presented. The parameters increase in complexity as the comparison process progresses. The simple independent parameters are evaluated first and subsequently are increased to parameters that are dependent on previously evaluated parameters. As the project progresses, the new parameters are added to better correlate the analytical effort to the experimental data. The results presented in this section are the result of the calibration load cases.

Load Cases Evaluated
Calibration Load Cases
The calibration load cases (CLC) are the simplest load cases evaluated by the GRC Round Robin. These CLC cases include static-load cases as well as constant-speed load cases. This subdivision was created to accommodate the analysis, which only can implement static, semi-static, or dynamic-load cases.

The calibration load cases range from 25% to 100% rated power in magnitude. The 100% load case gives good insight on the deflections and general behavior that is seen by the drivetrain under normal operation. The 25% rated load case serves as another point of comparison for the study. The operation of the drivetrain at 25% also utilizes the four-pole configuration of the generator operating at a lower constant speed. This enables testing of the models under different conditions. A 39.5% rated power case also is evaluated. This case is representative of a 100% rated load for the four-pole generator. This load precedes dynamic load cases such as generator shifting events, therefore an understanding of the behavior of the gearbox prior to these events could prove valuable. The table below shows a summary of the load cases noted above.

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Table 1. Calibration Load Cases

Classification	Load Case	Description	RPM	Torque	Units
Static	CLC 1.1	25% rated	N/A	120.77	kNm
	CLC 1.2	100% rated	N/A	322.61	kNm
Dynamic	CLC 2.1	25% rated	14.824	120.77	kNm
	CLC 2.2	39.5% rated	14.824	128.83	kNm
	CLC 2.3	100%rated	22.199	322.61	kNm

</div>

Design Load Cases
The design load cases are more complex than the CLC cases included here, and they follow the guidance of the IEC 61400-1 3rd ed. standard [8]. These design load cases are representative of the conditions used in the current industry design process. To generate the loading conditions the entire turbine must be modeled. In this case, the simulation

code FAST was used. FAST is an aeroelastic simulation code capable of simulating two- and three-bladed horizontal axis turbines. The code can simulate the transient dynamics of the entire system from which the fatigue and extreme events can be obtained. FAST uses a multibody and modal approach, incorporating rigid and flexible bodies to represent the turbine [7]. The entire drivetrain, however, is represented by a two-degrees of freedom system connected by a linear stiffness torsional spring; hence, there exists a need for further exploration of the drivetrain.

The IEC standard provides guidelines for the inflow characteristics that should be applied to the models, such as turbulence intensity, wind speed, and wind-shear conditions, depending on the turbine classification. The standard also gives guidelines for the fault conditions that would generate extreme loading conditions and which should be included in the design. The table below shows the characteristics of the GRC turbine class [8].

Table 2. Gearbox Reliability Collaborative Turbine Class

GRC Turbine Characteristics	
Wind Turbine Class	II
V_{ref} (m/s)	42.5
V_{ave} (m/s)	8.5
Turbulence Class	B
Turbulence Intensity at 15 m/s	0.14

The guidelines in combination with the aeroelastic model were used to generate the fatigue load spectrum and ultimate strength time series. The fatigue load spectrum is used to calculate the predicted life of the different gearbox components. The ultimate strength time series along with the fatigue time series is used to correlate the simulated prediction of the design process to the behavior observed in the field. These time series prove valuable because they can be fed into the models without the inherent noise existent in experimental data. Figure 9 shows the time at level load spectrum (left) for torque and a time series for a torsional extreme event (right).
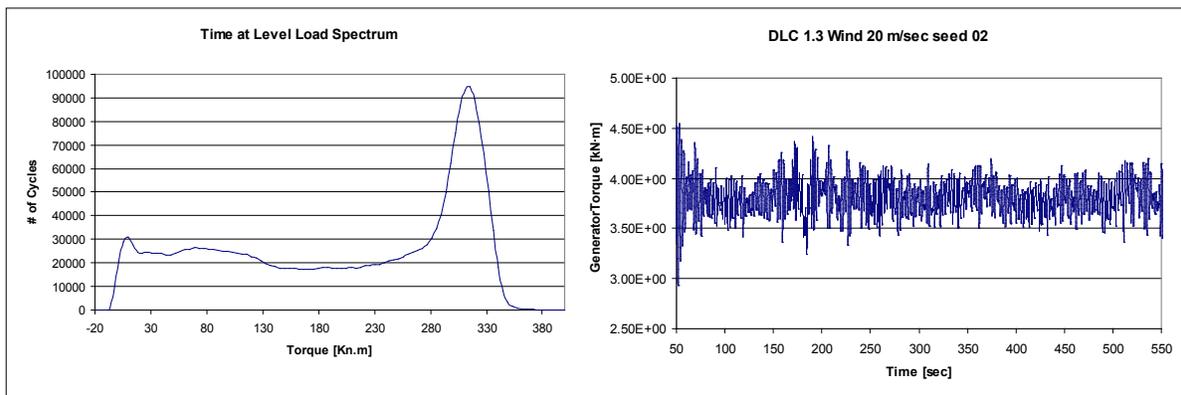

Figure 9. Fatigue load spectrum and simulated transient

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Preliminary Field Transients

An initial set of experimental data also has been included in the GRC load envelope. The purpose of this data is to validate the response of the entire drivetrain. Characteristics such as bulk stiffness and damping can be extrapolated from this information. The purpose of the data set is the validation of the models, therefore it is not very extensive; however, it is comprised of specific events that aid the validations. Among these are braking events, start-up, and generator-shifting events. Figure 10 shows torque at the main shaft of the turbine for an emergency braking event at low angular velocity and a startup event to rated power.
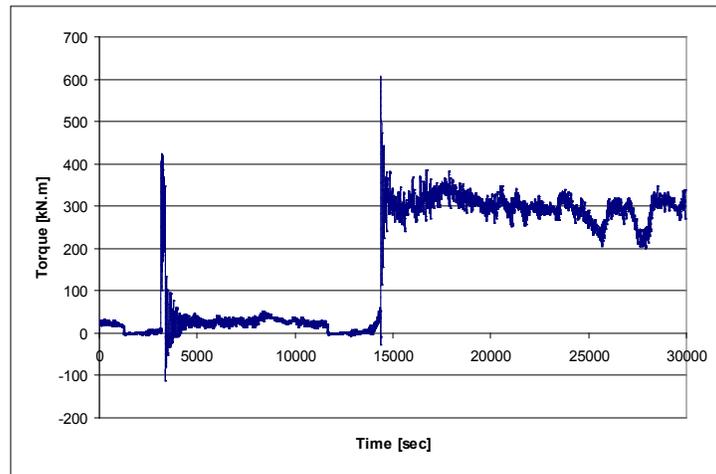

Figure 10. Torque time series from field testing

Parameter Evaluation and Results
Torque Distribution

Torque distribution can be defined as the torque entering each of the stages of the gearbox. It mainly was used for the initial comparison of the calibration load cases. Although it is a simple calculation it serves as a check for the models generated. Differences in gear ratios at any of the stages yield different results. Calculation of the torque distribution also ensures that the different participants are using the same sign convention and units. The result for the torsional distribution also is used as an input for a more-detailed analysis in models which are not fully coupled. Figure 11 shows the comparison of the different participants through the different stages of the gearbox.
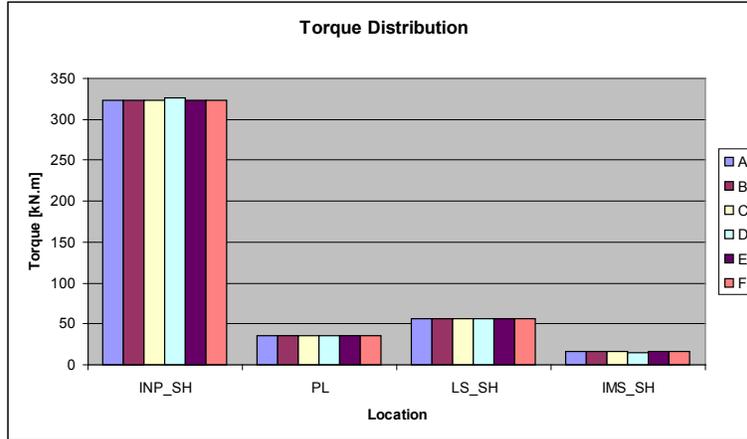
Figure 11. Gearbox torque distribution results

Gear Mesh Stiffness

The stiffness along the line of action through the mesh cycle is evaluated. The average throughout the mesh cycle is used as the means of comparison. The gear mesh stiffness is dependent on the gearing material properties, geometry, and the center distance. For the purpose of the CLC comparison, the center distance is assumed to be nominal; therefore, no housing compliance or misalignment is included. Figure 12 below shows the result for the ring planet mesh stiffness.
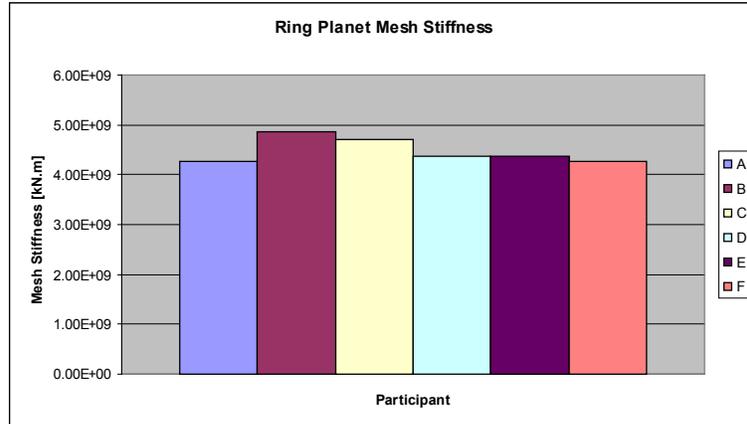
Figure 12. Planet ring gear mesh stiffness results

Torsional Compliance

The shaft torsional deflection is quantified throughout this parameter. The previously described torsional distribution is used as an input parameter to calculate the deflection on each shaft of the gearbox. This approach can be used to calculate the torsional stiffness needed for the generation of torsional models (Figure 13). This model approach is required by some certification agencies such as Germanischer Lloyd WindEnergie [9].
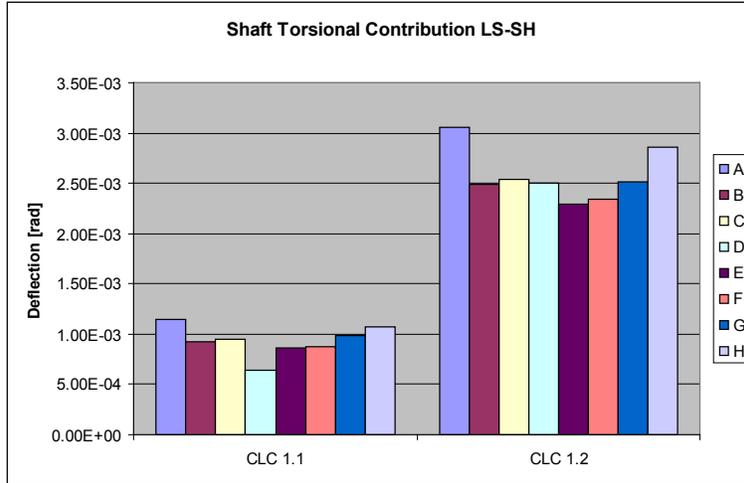
Figure 13. Shaft torsional compliance results

Gear Tooth Stress

The gear tooth contact stress is evaluated throughout the mesh cycle. Microgeometries such as tip relief crown and lead modifications are include in the calculation. For the initial comparison phase, misalignment has been ignored; hence, housing deflections, bearing compliance, and clearance are ignored. As the comparison process progresses these values will be validated and included in the stress calculations. Figure 14 shows the ring planet stress. Figure 15 shows a "slice" comparing the results for all the participants.
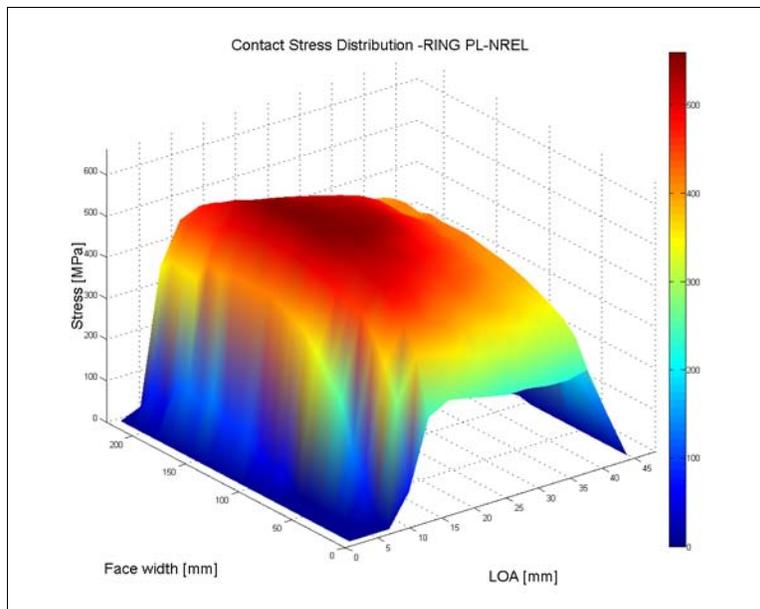
Figure 14. Gear tooth contact stress
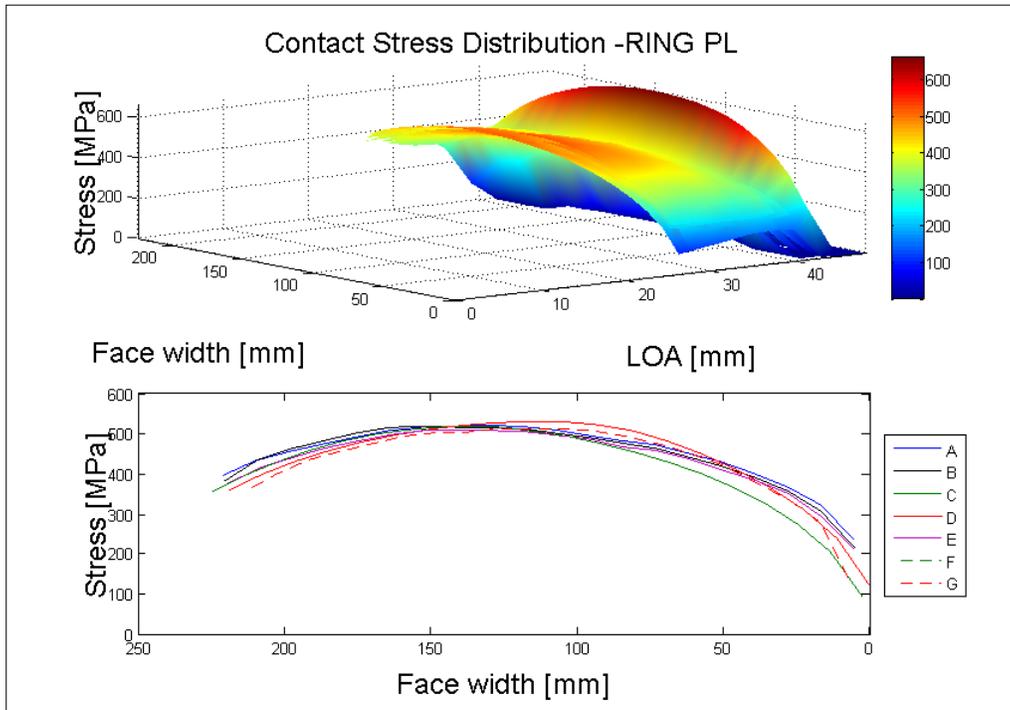
Figure 15. Contact stress distribution and participant comparison

Gear Tooth Loading

In this case, gear tooth loading refers to the load generated by the gear mesh. Wind turbine gearboxes generally have helical teeth producing a radial, tangential, and axial component as reaction forces from the gear mesh. These forces impact the gearbox shafts, bearings, and housing, producing deflections which result in misalignments—which emphasizes the importance of the gear tooth loading. Figure 16 shows the forces generated by the planet annulus mesh.
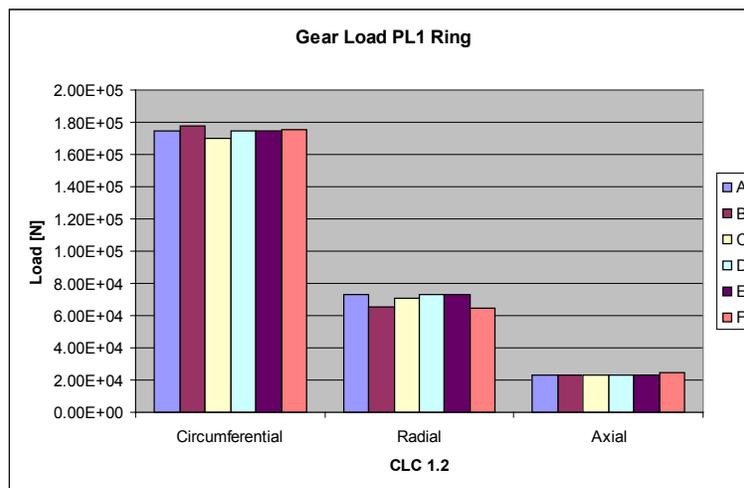
Figure 16. Planet ring gear loading result

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The following parameters currently are being evaluated; therefore, the results are not presented in this document.

Planet Carrier Torsional Misalignment

As the planet carrier undergoes torsional loading it twists slightly; this twist is one contributor to planet misalignment. This parameter quantifies the angular misalignment of the pin responsible for holding the planet. These values will be included in contact stress calculations and the evaluation of the microgeometry.

Planet Load Share

Planet load share can be defined as the load carried by each planet in the planetary stage. Theoretically, it should be equal for all planets; however, there is debate with respect to the load equalizing capabilities of the floating sun configuration under dynamic conditions. This parameter is intended to clarify some of the existing doubts regarding the planet load share with validation by means of experimental data. To date, the main focus of comparison has been static cases, so there have not been any major deviations from the theoretical approach.

Sun Motion

This parameter is closely correlated to the previously noted load share. It specifically examines the motion of the sun with respect to the planet carrier. The correlation of the motion and the load share might give better insight into the behavior of the internal components of the gearbox.

Gear Tooth Contact Pattern

The gear tooth contact pattern is closely related to the tooth contact stress and microgeometry. The contact patterns will be monitored in the initial stages of the dynamometer testing as the run-in procedure is followed. This possibly will provide an approximate dimension and location of the contact patch for several loading conditions. This information will be used to validate the global deflections of the gearbox, especially in the planetary stage.

Shaft Bending

Shaft bending, which is evaluated under the loading conditions generated by the gears, is of great importance to the misalignment of the internal components. For the purpose of the GRC Round Robin, specific locations were chosen along the shaft so that they can be compared among the different participants.

Bearing Stiffness

The highly dependent nature of the bearing stiffness on misalignment and loading requires modeling that captures the integrated behavior of the gearbox. For the purpose of the GRC Round Robin, specific load cases which exemplify the nonlinear behavior of the bearings have been chosen for the comparison. The calculated stiffness from the high-fidelity models will be simplified and used for the dynamic models.

Bearing Roller Load Distribution

This parameter refers to the number of rollers in contact as well as the load carried by each roller. The parameter is highly dependent on the preload of the bearing, which changes the operating clearances. It is of particular relevance in the planetary stage, where the planet generally has a very thin wall so as to reduce overall weight. The thin wall of the planet deforms under the load, leading to an oval shape and thus changing the loading characteristics of the load zone. Roller contact stress also will be evaluated.

Housing Deflections

The housing behavior will be evaluated by monitoring its influence on other parameters. This includes the change in gear center distance due to housing compliance as well as its influence in the bearing and shaft behavior.

Future Work

The GRC Round Robin is a work in progress. At this point, the main focus has been the calibration load cases and the less-involved parameters. Regardless of the simplicity of the load cases, a great deal of progress has been made in the comparison, initially presenting large discrepancies in the results. As the project progresses, the importance of the assumptions made for the different analytical approaches and their impact on the results will be very informative. Additionally, the evaluation of the strengths and weaknesses of different tools and their partial overlap will prove valuable for their implementation in the different stages of the design process.

The GRC Analysis Round Robin will continue the comparison of the analytical parameters, advancing to the more complex parameters with a higher dependence on the global behavior of the gearbox. As the analysis and the experimental testing progress, the parameters evaluated will be adapted to better reflect and evaluate the internal behavior of the gearbox. Finally, the parameters evaluated will expand to include the behavior of the entire drivetrain, thereby truly capturing its dynamics.

As the data from the field and dynamometer test is generated, the validation of the analysis will take place. This process will include the reevaluation of the analytical assumptions and the final tuning of the models. The GRC Round Robin also will provide feedback to the dynamometer testing so that specific load cases that are tailored to the validation of the models can be input into the GRC test subject.

It is hoped that the GRC Analysis Round Robin as an integrated part of the GRC will shed light on the root causes of the premature failure of gearboxes. This should provide guidance in the use of analytical tools for the design process, and point out the impact of the assumptions inherent in the implementation of these tools.

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