A Multi-Component Analysis of a Wind Turbine Gearbox using A High Fidelity Finite Element Model

THESIS

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Abstract

In this study, a high fidelity finite element model of a wind turbine gearbox is constructed in an advanced gear analysis program. The model is used to analyze motions and loads of the gearbox components. The model is validated using experimental data provided by the National Renewable Energy Laboratory (NREL). Carrier motion measurements, planet motion measurements, and planet bearing strain measurements are all simulated and compared to the experimental data. Manufacturing errors and their effect on system components are evaluated with specific detail given to the planetary stage bearings and the spline on the sun gear shaft. Modification of the high speed shaft bearing raceways as a way to provide strain amplification is also modeled and used to validate particular groove geometry.
Dedication

This thesis is dedicated to my parents, Jay and Joyce Austin.
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CHAPTER 1

INTRODUCTION

1.1 Background, Motivation and Literature Review

Energy prices, supply uncertainties and environmental concerns have driven investment in sources of clean renewable energy. Additionally, the United States is working toward achieving energy independence and relying increasingly on domestic resources. A diverse energy portfolio was developed under President Bush in 2006. Under this model wind would provide 20% of the nation’s electricity by 2030 [1]. Significant challenges face the wind energy industry if this demand is to be met. The United States has offered production tax credits (PTCs) which have accelerated growth and investment in the industry. The tax credit is 2.3 cents a kilowatt-hour for electricity produced over a wind farm’s first 10 years of operation, which brings the price of wind power close to that of conventional fuels [2]. New installations have been largely dependent on the availability of the PTCs and in order for wind energy to be independently sustainable and attractive to investors, the cost of energy production must
come down. Operations and maintenance contribute a significant amount to the total cost of wind energy production [1]. Gearbox reliability is a major issue and the replacement of this component is quite expensive. Initially, failures were due to fundamental design errors compounded by under-estimation of operating loads [3]. Over the past two decades wind turbine design standards have been developed, however, the design life goal of twenty years is still not achieved on many designs. These high failure rates increase maintenance costs and also impact the sale price of gearboxes as manufacturers add contingencies to the price to cover warranty risks.

There are alternative wind turbine designs that employ direct electricity generation and eliminate the gearbox. However, their initial costs are far higher which has prohibited these designs from large scale implementation [4]. Geared turbine designs remain as the most cost effective model despite the current reliability issues.

The National Renewable Energy Laboratory (NREL) has made a commitment to address gearbox reliability as a part of its research agenda. A gearbox reliability collaborative (GRC) was developed to address major gearbox issues with the goal of increasing overall reliability of wind turbines. Many gearbox problems may be a result of poor communication and feedback during design, operation, and maintenance of turbines [3]. The GRC is aimed at bringing together these pieces of the design process and to encourage collaboration and sharing of information. A 750 kW turbine design was chosen by a committee of experts and gearbox consultants hired by NREL under the GRC. This design is chosen to be representative of many turbines currently in service.
It is also speculated that many of the problems seen in wind turbine gearboxes are a result of common causes among all power levels.

NREL utilizes a 2.5 MW dynamometer test facility as part of the GRC testing procedure. The GRC gearbox is instrumented to better understand loads and system dynamics of various components. This instrumentation is aimed to address specific components which have seen high rates of failure. Through use of the dynamometer new configurations can be evaluated quickly compared to field testing and instrumentation is far easier to install and maintain. Field testing is also a part of the GRC agenda in order to get a better understanding of a typical load spectrum for an operational gearbox. These loads can also be replicated within the dynamometer for more in depth analysis. The experimental efforts of the GRC are critical to understanding system loads and establishing design requirements. This data can also be used for validation of drive train models. Model validation and drive train analysis comprise another portion of the GRC agenda. Validating new and existing models is an important step in improving future designs. Modeling practices can be developed from this effort and applied to future designs.

The development of a failure database is necessary to help identify lapses in the design process, which is a goal of the GRC – to create a public database and encourage the sharing of information and knowledge within the industry. The majority of industry failures have been seen in the bearings of various stages in the gearbox [3] [5]. There are three critical bearing locations that have been identified:
1. Planet Bearings
2. Intermediate shaft bearings
3. High-speed bearings

The majority of observed failures appears to initiate at bearing locations, and are not a result of gear failures or gear-tooth design deficiencies. The failures appear to initiate at bearing locations and then propagate into other components of the gearbox. Additional failures and component damages were observed in a GRC gearbox which was inspected by The Gear Works. There were a number of component failures seen in the disassembly of the gearbox, many of which were attributed to lubricant starvation. Components damaged in this gearbox included the high speed gear set, sun spline, and planet bearing [5].

In order to reach the energy production goals, an effective modeling approach must be developed. It is the goal of this project to use the experimental data produced by NREL to validate and improve new and existing models. In doing so, modeling best-practices can be developed and applied to future designs. A full finite element model such as the one used in this study are essential to the modeling efforts. Transmission3D, the finite element program used, allows for a full gearbox analysis with accurate gear and bearing contact evaluation [6]. The objectives of this study are as follows:

- Further validate and improve the existing Transmission3D model with additional experimental data
- Evaluate the load sharing capacity of the GRC gearbox.
• Analyze the sun shaft spline and evaluate the effect of spline lead modifications on sun float, spline stiffness, spline tooth load distribution, and planet load sharing.

• Evaluate the effect of manufacturing errors on planet load sharing.

• Aid in the development of instrumentation and experimental procedures.

1.2 GRC Dynamometer Set Up

The GRC gearbox is tested using the 2.5 megawatt dynamometer. The load on the gearbox is supplied by a 750 kW generator which would be used on the actual turbine. The gearbox is mounted at a 5° angle as it would be in real world operation. Wind turbine gearboxes experience a wide range of non-torque loads (NTLs) during field operation due to winds perpendicular to the axis of rotation. Non-torque loads are exactly as the term implies - any load that is not generated by the torque of the rotor blades. In the dynamometer these can be applied using two opposing hydraulic actuators. These two actuators are positioned at a 45° angle from the floor and tilted 5° to match the angle of the gearbox, as seen in Figure 1.1. There is also a thrust actuator that is capable of applying axial forces.

The gearbox is instrumented with 126 channels of various transducers which include strain gages, thermocouples and displacement probes. Some of the primary channels will be discussed in further detail. The transducers are sampled at 100 Hz. The
instrumentation was designed to analyze the components which were observed to have the highest failures, with much of the instrumentation being applied to the planetary stage.

![Image of NREL 2.5 MW dynamometer with non-torque load actuators](image)

Figure 1.1: NREL 2.5 MW dynamometer with non-torque load actuators [12].

1.3 Physics of the GRC Gearbox

Wind turbines use a multi-stage gearbox to transmit power from the rotor to the generator at the proper speed and torque. The rotor speed is between 15 and 22 rpm while the required output speed to the generator is 1800 rpm. This is a requirement of the generator to produce electricity at the proper electrical grid frequency. The GRC gearbox is representative of a typical configuration present in most turbines today and is shown in
Figure 1.2. The design is a speed increaser that includes a planetary stage followed by two parallel shaft stages. The gearbox has a power capacity of 750 kW with an overall speed increase ratio of 1:81.491. The nominal input speed is 22 rpm and at this ratio it is increased to an output speed of 1800 rpm. A 323 kN-m input torque is required on the rotor blades to operate the generator at full rated power.

Figure 1.2: Exploded view of example gearbox. [7]

The following nomenclature will be used throughout the rest of this document when referencing internal components of the gearbox. Upwind components are those located on the entrance side of the stage before power has passed through the gear. These may also be termed rotor side, as they are located closer to the rotor. Downwind components are those located on the aft end of the shaft. These components may also be termed generator side as they are closer to the generator. The power flow begins at the rotor blades which introduce torque to the system through the low speed input shaft. This input shaft is connected to the planet carrier. The planetary system is a three planet fixed
ring layout with a floating sun configuration. A three planet system with one floating member should not have issues with load sharing amongst the planets [8]. This particular design does not allow pure radial float of the sun gear, but rather it is cantilevered from a splined connection at the downwind portion of the shaft. The spline is crowned heavily and this is meant to provide freedom of the sun shaft at the spline location. The load sharing capability of this configuration is evaluated in subsequent chapters. The planet gears are supported by two single row cylindrical roller bearings. The planet carrier is supported upwind and downwind by full complement cylindrical roller bearings.

The parallel shaft stages are comprised of three separate shafts, each supported by three bearings. The bearing configuration of each shaft consists of a cylindrical roller bearing on the upwind portion of the shaft (LS-SH-A, IMS-SH-A, HS-SH-A) and a pair of tapered roller bearings on the downwind portion of the shaft. Bearing locations and naming convention can be seen in Figure 1.3. The transmission is contained in a two piece housing that is bolted to the ring gear. The torque reaction arms encase rubber bushings to dampen vibrations of the gearbox and these bushings are mounted to the bedplate that is connected to the tower. The input shaft is supported by a spherical roller bearing to mitigate NTLs from entering the gearbox.

1.4 Transmission3D Model

Transmission3D is a linear finite element contact analysis program developed by Dr. Sandeep Vijayakar [6], specifically designed for analysis of multi-mesh gear drive trains. The program permits the user to model the housing, bearings, shafts, planet
carrier, and gears as deformable bodies. Gear micro-geometries can be included as well as assembly and manufacturing errors. Bearings are generated similar to gears and can include crowning on the rollers and races and can include clearance and preload.

![Gearbox Bearing Locations and Nomenclature](image)

**Figure 1.3: Gearbox bearing locations and nomenclature [7].**

The true benefit Transmission3D is in the contact algorithm. Gear, bearing, and spline stresses are localized and are influenced by micro-geometries which are on the micron scale. Traditional finite element programs are ill suited for this application as the width of the contact line between gear teeth is orders of magnitude smaller than the dimensions of the gear teeth. This would require a highly refined finite element mesh
over the entire contact zone or would require the localized contact elements to be remeshed at each position of the mesh cycle [9]. This would greatly increase computation time. Transmission3D uses a unique contact analysis solver, Calyx, which utilizes a hybrid algorithm of finite elements to predict far field displacements and an elastic half space model to predict relative displacements local to the contact region [10]. In doing so, Calyx does not require highly refined a finite element mesh.

There are two assumptions made by Calyx, the first assumes that the finite element solution predicts deflections well for points far enough away from the contact region. Second, the solver assumes that the contact region is sufficiently smaller than the dimensions of the gears themselves and employs the semi-analytical technique that uses the surface integral form of the Boussinesq solution. This is done by overlaying a contact grid, separate from the finite element mesh. The finite element solution and semi-analytical solution are then matched at a subsurface interface [10]. The stiffness terms from each solution are combined into the overall compliance matrix and a simplex-type solver is used to evaluate the correct load distribution. This is performed simultaneously at all contacting interfaces including gears, bearings, and splines.

Quasi static analyses were employed in which a static solution for each time step at multiple instances through time. Mesh size is determined based on the criticality of the particular component with more critical components given a finer mesh. Meshes may also be adapted to locate specific nodes if data is needed at a specific location, to simulate a sensor output for example. Bearings can be modeled using stiffness matrices or as rolling elements in contact. Rolling element bearings are modeled investigating contact
at every roller location and the user is able to specify all geometric, preload, and clearance properties of the bearing. In general, stiffness bearings are preferred due to the significant decrease in computation time. The semi-analytical technique at gear and bearing contacts along with the methods employed to increase computational efficiency are what make Calyx and Transmission3D unique and so well suited for this application.

The Transmission3D model used in this study was initially built by Dr. Sandeep Vijayakar and was later updated by Aaron Thaler [11]. The model was revised and updated during this study to improve accuracy and to provide a better match with experimental data.

1.5 Thesis Outline

Chapter 2 provides an overview of the experimental proceedings and modeling results and comparison. Chapter 3 details the effect of carrier and planetary gear manufacturing errors on sun float, planet load share, and gear load distributions. Chapter 4 provides a comparison of two spline modeling approaches and validates the Transmission3D spline model for use in spline stiffness and spline tooth load intensity prediction. In Chapter 4, the effect of spline tooth lead modifications on sun float and planet load share are evaluated. Chapter 5 details the validation of proposed bearing race groove geometry for the purposes of strain amplification. Major conclusions and suggestions for future work are presented in Chapter 6.
CHAPTER 2

MODEL BUILD-UP, RESULTS AND EXPERIMENTAL COMPARISON

2.1 Model Overview

The following chapter includes the results gathered from the Transmission3D model and the comparison to experimental data. There were a number of signals used in the GRC instrumentation and those that could be simulated using the model will be compared in this section. Previously, the input shaft strains and ring gear strains were modeled and validated by Thaler [11] using the experimental data gathered through NREL testing [12]. This chapter will focus on experimental data released after June 2011. The parameters studied include; i) carrier rim deflection and carrier motion, ii) planet rim deflection, iii) planet bearing race strain, and iv) sun motion.

Most of the subsequent results are gathered from the planetary stage. The primary time reference scale will be carrier rotation and many of the signals will be plotted with respect to carrier position. The carrier rotates clockwise when viewed from upwind and zero degrees corresponds to the point when Planet A is located at top dead center (TDC) as shown in Figure 2.1. All figures from the Transmission3D model will be drawn with
the actual finite element mesh that was used in the simulations. Any figure with a color contour will show stressed parts, however, the maximum stresses are usually at contact locations that are not always visible.

Figure 2.1: Planetary gear set coordinate system and planet gear labeling convention
2.2 Planetary Stage Overview

The planetary stage comprised a large portion of the modeling effort. As mentioned earlier, the planetary stage and more specifically the planetary bearings were identified as one of the critical regions. The components in this stage of the gearbox constitute a significant portion of the gearbox related failures. The gearbox was heavily instrumented to measure a number of parameters and they are outlined in Table 2.1. There were six proximity sensors mounted on the housing, measuring the carrier rim with the intent of deducing carrier motion. Six additional proximity sensors were placed on the carrier to measure the displacement of the planet gear rims. Measurements were taken on two planets each using three sensors. Strain gages were mounted on grooves that were machined into the inner diameter of the inner race of the planet bearing. Three grooves were machined into each bearing and two gages were mounted in each groove for a total of 6 gages per bearing, 12 gages per planet (6 on the upwind bearing and 6 on the downwind), and 36 gages in the whole planetary system. Two proximity sensors were also mounted on the carrier to measure the upwind portion of the sun shaft in order to extract sun motion. Strain gages were also mounted on the input shaft to measure the magnitude of NTLs. Strain gages were used to measure the ring gear strain in order to extract load distribution of the gears. The input shaft strain and ring gear strain were simulated prior to this document by Thaler [11]. Proximity sensors and bearing strains will be discussed in detail in this chapter.

The gearbox utilizes a three planet carrier system at the input stage, as shown in Figure 2.2. The power flows from the input shaft to the carrier and is output at the sun.
Figure 2.3 shows the gears of the planetary system. The ring gear is bolted to the housing and the sun gear is floated by a spline on the downwind portion of the sun shaft, as seen in Figure 2.4. The spline is heavily crowned to maximize the “float” of the sun and in turn create optimal load share between the planets. The planet carrier is supported by two cylindrical roller bearings, one on each side of the carrier and there are two cylindrical roller bearings located under each planet, as shown in Figure 2.5. The planet bearings are supported at their inner diameter by a pin which then mates with the carrier.
<table>
<thead>
<tr>
<th>Purpose</th>
<th>Sensor Location</th>
<th>Sensor Type</th>
<th>Model Comparison</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measure carrier rim deflections to</td>
<td>Carrier Rim Sensor</td>
<td>Proximity Sensor</td>
<td>Yes</td>
</tr>
<tr>
<td>evaluate carrier motion.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Measure planet gear rim deflections to</td>
<td>Planet Rim</td>
<td>Proximity Sensor</td>
<td>Yes</td>
</tr>
<tr>
<td>evaluate planet gear motion.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Measure planet bearing race strain to</td>
<td>Planet Bearing Inner</td>
<td>Strain Gage</td>
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</tr>
<tr>
<td>evaluate bearing loads.</td>
<td>Race</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Measure sun motion.</td>
<td>Planet Carrier</td>
<td>Proximity Sensor</td>
<td>Yes</td>
</tr>
<tr>
<td>Measure of input shaft strain to evaluate</td>
<td>Input Shaft</td>
<td>Strain Gage</td>
<td>No [11]</td>
</tr>
<tr>
<td>magnitude of NTLs</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Measure ring gear root strain to evaluate</td>
<td>Ring gear</td>
<td>Strain Gage</td>
<td>No [11]</td>
</tr>
<tr>
<td>load distribution.</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2.1: Sensor Overview
Figure 2.2: Planet carrier and planet gears.
Figure 2.3: Ring, planet, and sun gears
Figure 2.4: Mid-plane cut-away of planetary stage, showing sun spline location.
2.3 Carrier Motion Measurement

The displacement and deflection of the planet carrier is measured using six non-contact proximity sensors, detailed in Figure 2.6 and Figure 2.7. The coordinate system is shown in Figure 2.8 and Figure 2.9. The sensors are attached to the housing and measure the carrier rim, which is machined to give an adequate measurement surface.
Figure 2.6 shows the positioning of the axial rim sensors which are mounted on the housing and measure the relative displacement of the carrier rim with respect to the housing. The carrier rim measurement surface is located 412 mm radially from the center of the carrier. There are four sensors spaced 90 degrees from one another and from these the axial displacement and tilt of the carrier about the X and Y axes can be obtained. Figure 2.7 shows the positioning of the radial rim sensors which are mounted on the housing and measure the relative displacement of the carrier with respect to the housing. The two sensors are spaced 90 degrees from one another and from these radial displacements the carrier X and Y displacement can be obtained. There are six carrier rim sensors in total:

1) Sensor_047 – Axial rim sensor (Figure 2.6).
2) Sensor_137 – Axial rim sensor (Figure 2.6).
3) Sensor_227 – Axial rim sensor (Figure 2.6).
4) Sensor_317 – Axial rim sensor (Figure 2.6).
5) Radial_040 – Radial rim sensor (Figure 2.7).
6) Radial_310 – Radial rim sensor (Figure 2.7).

Through the use of these six sensors the carrier displacement with respect to the housing can be determined in the 5 degrees of freedom of interest. Downwind displacement of the carrier rim, in the positive Z-direction, creates a positive sensor output.
Figure 2.6: Planet carrier rim axial sensor and deflection measurement system, figure courtesy of NREL [12].

Figure 2.7: Sensor locations for carrier radial motion, figure courtesy of NREL [12].
Figure 2.8: Transmission3D planet carrier model and finite element mesh.
2.4 Transmission3D Model

The Transmission3D model includes all gear stages and includes contact at all gear mesh locations. Bearing stiffness matrices, provided by Romax [14] were used to increase computational efficiency with the exception of the planet carrier bearings (PLC-A and PLC-B). The planet carrier bearings were modeled as journal bearings, see Figure 2.11. It is imperative to include clearance in the planet carrier bearings as this effects the
relative displacement of the carrier with respect to the ring and sun gear. Without these clearance values included, the planet bearing load profile changes dramatically. Stiffness bearings are not sufficient because a stiffness matrix does not include the non-linearity caused by the clearance. In reality, the bearing has zero stiffness until the clearance is “used up” and contact is initiated. Stiffness matrix bearings do not capture this non-linearity.

Journal bearings were used because the planet carrier bearings are virtually unloaded and the journal bearings allow for clearance while maintaining computational efficiency. The alternative is to fully model the cylindrical roller bearings. These have significantly more elements and contact points, as seen in Figure 2.11, thereby increasing computation time by nearly three times.

The carrier and housing models must be built outside of the Transmission3D environment and converted to a bulk data file before importation to T3D. The carrier and housing models were built and meshed using SolidWorks. Care should be taken to ensure proper aspect ratio of the elements, an aspect ratio of <5 was achieved for all elements of the carrier and housing. The meshes are then exported from SolidWorks to a Nastran file and later converted using the cvtbdf utility included within T3D.

For purposes of data extraction, a compliant shaft piece (E = 10GPa) was attached to the carrier rim, as shown in Figure 2.10. This applied a more uniform mesh at the point of interest which allowed for more effective post processing. Due to the low modulus of elasticity, the displacement of the carrier rim was negligibly effected by this
shaft segment. The post processing function, Get_FEDisplacement, was used to extract the nodal displacement at each position around the circumference of this shaft. The displacement at each sensor location was then extracted and compiled to create the simulated sensor output. The housing displacement at each sensor mount location was also extracted and removed from the carrier rim values; this provided the relative displacement which was the measured quantity. The simulated signals include both the rigid body motion and finite element deformations.

Figure 2.10: Carrier rim shaft segment.
Figure 2.11: Journal (left) and full cylindrical roller (right) carrier support bearing.
2.4.1 Carrier Axial Motion Simulation

The method used by NREL for obtaining carrier axial displacement and rotation about the X and Y axes is determined from the four axial proximity sensor measurements discussed earlier and shown in Figure 2.6 [12]. These sensor outputs were simulated in the Transmission3D model for validation purposes. Figure 2.12 shows the simulated sensor outputs from Transmission3D. These sensor outputs are the absolute displacements of the carrier at the sensor locations, meaning they are not relative displacement and housing displacement has not been removed. The proximity sensors measure both the deformations and rigid body displacements of the carrier. The model provides the ability to separate these two components. The deformation of the carrier rim is measured by the proximity sensors but is not an indication of the carrier displacement. The deformation of the carrier and carrier rim are exaggerated and shown in Figure 2.13 where the “rippling” of the carrier rim can clearly be seen. The rigid body motion is the parameter of interest and what is intended to be calculated from the sensor outputs. The exaggerated rigid body displacement of the carrier is shown in Figure 2.14.

In order to provide a basis for comparison, the housing displacement of the model at the sensor locations must also be extracted. Figure 2.15 shows the displacement of the housing at the sensor locations. The mean offset of the signals is due to the rigid body motion of the housing and positive sensor output indicates downwind motion. There is also a small three per revolution variation in each sensor output. This is due to the planet gear passing the sensor location, as the carrier rotates and the planet gear passes, the
Deflection of the housing is affected. The sensors are out of phase by 90 degrees as would be expected due to their spacing being 90 degrees from one another.

The GRC experimental setup measures the relative displacement of the carrier rim with respect to the housing. Figure 2.16 shows the simulated relative displacement of the carrier with respect to the housing which is obtained by subtracting the housing sensor displacement from the carrier sensor displacement. This is the basis for comparison with the experimental data. Downwind deflection is positive.

![Graph showing sensor displacement vs. carrier rotation](image_url)

**Figure 2.12:** Transmission3D carrier rim sensor axial displacement measured in fixed frame.
These are the large three per revolution peaks seen in the simulated signal.

Figure 2.13: Carrier rim deformation (no rigid body displacements shown) with pin contact included, color contours are Von-Mises stresses.
Figure 2.14: Exaggerated carrier tilt (rigid body motion only), color contours show Von-Mises stresses.
Figure 2.15: Transmission3D housing axial displacements at sensor mount locations.

Figure 2.16: Transmission3D axial carrier sensor output.
2.4.2 Carrier Pin Interface

Clearance at the carrier-pin interface greatly impacts the carrier rim deflection [15]. The results of this paper showed that the carrier rim deflection was heavily dependent on the carrier pin interface. The same results were seen in the Transmission3D model as well. Two pin interfaces were explored, the first being a rigid “welded” connection where the pin is constrained to the carrier bore in all degrees of freedom. The second pin interface allowed for clearance and contact between the pin and carrier bore a diametric clearance of 15 micron. Figure 2.17 shows the carrier and pin model and finite element mesh and Figure 2.18 shows the isolated carrier pin. The raised bars in the figure represent the contact pressures at each contact point when pin clearance and contact are included in the model. Figure 2.19 shows the axial carrier rim displacement at the Sensor_047 location for each pin interface model. The rigid interface model has six distinct peaks. The carrier has six features that affect rim deflection. There are three planet pins and there are three additional stiffening features which are intended to stiffen the carrier in torsion. The six peaks correspond to the passing of each of these features. The rigid interface adds torsional stiffness to the carrier and this creates more distinct peaks with lower amplitudes. The clearance fit has three dominant peaks with higher amplitudes. The peak associated with the planet pin passing the sensor is greatly decreased and the peak associated with the other torsional stiffening feature is increased by nearly three times. Therefore, for good sensor comparison the pin interface must be accurately modeled. These two interface conditions represent two ends of a spectrum; in reality the pin interface is likely somewhere between the two. In the pin
contact model there is no friction at the interface. This is not an available option for pin
contacts in Transmission3D and hence there are no interfacial shear forces between the
two surfaces. The rigid interface represents the opposite end in which the maximum
shear forces will be generated. The GRC gearbox has clearance at the pin interface but
there will be friction which may have an effect on the rim deflections.

Figure 2.17: Carrier pin connection, color contours show Von-Mises stresses.
Figure 2.18: Carrier pin modeled with bore clearance, contact pressures are shown (color contours show Von-Mises stresses).

Figure 2.19: Transmission3D comparison of Sensor_047 axial displacement for both carrier-pin connection models.
The carrier-pin interface condition greatly impacts the torsional stiffness of the carrier and therefore it impacts the pin alignment. The model with pin contact has greater pin misalignment and this impacts the gear mesh alignment as well. Figure 2.20 shows the Planet A upwind bearing load plotted against carrier rotation for three model conditions. The first is a rigid pin interface with no gravity included, the second is pin clearance with no gravity, and the third is pin clearance with gravity included. Gravity causes the carrier to tilt with respect to the housing, this causes a misalignment at the gear meshes and will show up as a once per revolution variation in the planet bearing loads. This is why clearance had to be modeled at the carrier bearings - PLC-A and PLC-B - as mentioned earlier. Without clearance these gravitational effects are not captured because misalignment of the carrier is constrained. The misalignment due to gravity favors the upwind side of the face width through half of the carrier rotation and the downwind side of the face width through the other half. Figure 2.20 shows the bearing loads for the upwind bearing of Planet A for these three conditions. Through the addition of pin contact there is an increase in upwind bearing load over the rigid pin condition and the addition of gravity introduces a once per revolution variation to the planet bearing load. As previously stated, the planet gear is supported by two cylindrical roller bearings. The load on these two bearings is caused by the loads generated at the ring-planet and sun-planet meshes. The ratio of the upwind bearing load to the total load carried by the planet provides an indication of the load distribution. Perfect load share, a value of 0.5, would indicate that there is equal load on the upwind and downwind bearing. This would potentially indicate that the load is equally distributed across the face width and the
resultant load is located at the midpoint of the face width. A value greater than 0.5 indicates that the load distribution favors the upwind side of the face width and a value less than 0.5 indicates that the load distribution favors the downwind side of the face width. Figure 2.21 shows the Planet A upwind bearing load share for the various conditions. The rigid pin interface with no gravity condition shows near perfect load share, but is slightly greater than 0.5 indicating that the load distribution slightly favors the upwind side of the face width. This is due to the “twist” or torsional deformation of the carrier causing a pin misalignment. The pin contact condition increases the upwind bearing load share but has no impact on the shape of the signal, however, when gravity is added the mean remains the same yet there is a once per revolution variation added to the signal due to the tilt of the carrier. The upwind bearing load share shows how the gear load distribution changes as the carrier rotates through time and the effect of gravity can be seen. As mentioned previously, the bearing loads are caused by the gear load distributions. Figure 2.22 shows the load distribution on Planet A at the planet-ring mesh for the three model conditions with the carrier at the zero degree location. It can be seen that the inclusion of pin clearance causes a slight upwind shift in load distribution and the inclusion of gravity causes a far greater upwind shift in load distribution. This is consistent with what was seen in the upwind bearing load share seen in Figure 2.21. Figure 2.23 shows the load distribution on Planet A at the planet-sun mesh for the three model conditions with the carrier at the zero degree location. There is less change in load distribution between the rigid and pin clearance conditions. The addition of gravity
causes a slight upwind shift in load distribution but not as significant as was seen in the planet-ring mesh.

Figure 2.20: Planet A upwind bearing load for various pin modeling conditions.
Figure 2.21: Planet A upwind bearing load share for various pin modeling conditions, this is the ratio of the upwind bearing load to the total load carried by the planet.
Figure 2.22: Load distributions on Planet A gear at the planet-ring gear mesh for various pin conditions. These are for the carrier at the 0 degree position.
Figure 2.23: Load distribution on Planet A gear at the planet-sun gear mesh for various pin conditions. These are for the carrier at the 0 degree location.


2.5 Comparison of Carrier Rim Sensor Results

Figure 2.24 shows the comparison of the simulated Transmission3D data and experimental data, for Sensor 047. The experimental signal has a six, three, and a once per revolution component. As seen in the previous section the pin interface has a large impact on the rim deflection. The rigid interface caused six distinct peaks in the carrier rim and the clearance interface had three distinct peaks. The experimental signal has both suggesting that the interface would be somewhere in between the rigid and clearance interfaces. This effect could potentially be caused by friction at the pin-carrier interface and the shear forces generated thereby. The simulated signal uses the clearance pin interface and compares quite well with the exception of the once per carrier revolution signal component. There are a couple of potential sources of this component. One suggested cause was presented by Crowther [16], in which a machining tolerance was cited. This would essentially act as face run-out of the measured surface and would, therefore, be included in the signal at once per revolution. This would be a relatively simple inspection and would perhaps be worthwhile to provide additional modeling insight.
Figure 2.25 shows the sensor output that would be caused by the face run-out of the carrier rim. It is a sinusoidal output with 60 micron amplitude. This machining tolerance is superimposed on the original simulated signal and the summation of these two signals is shown in Figure 2.26 and compared to the experimental data again. With the addition of the runout signal, the data compares is very well.
Figure 2.25: Once per carrier revolution sensor output, due to carrier rim runout.
A second potential source of the once per revolution component would be a rotating misalignment of the carrier, due to an assembly error. This error will generate the same sensor response as the face run-out, a once per revolution sinusoidal variation, but it may have other system effects. The rotating misalignment is actually a system parameter rather than just a measurement effect. The face run-out is completely benign and will have no impact on the other components of the system, whereas the misalignment could potentially affect bearing loads and gear load distributions. This error was simulated in Transmission3D by misaligning the carrier with respect to the...
housing and the sensor output comparison is shown in Figure 2.27. The once per carrier rotation component can be clearly seen in the simulated data. The phasing of the misalignment would need to be adjusted slightly to achieve the optimal correlation. However, it is clear that this signal component can be caused by manufacturing error as well as carrier rim run out. It would be worthwhile to inspect the carrier rim surface so run out could be verified or eliminated as the cause.

Figure 2.27: Sensor 047 zero-mean comparison for simulation with rotating carrier misalignment.
2.6 Radial Carrier Sensors

The radial rim sensors are mounted on the housing in two locations and used to measure the relative radial displacement of the carrier with respect to the housing. Figure 2.28 and Figure 2.29 show the radial sensor comparison of the simulated Transmission3D model and the experimental data. Once again, the signals compare quite well to the experimental data. The phase and frequency content compares very well between the two signals, however, the model has lower amplitudes. The six peaks correspond to the passing of each of the six features on the carrier, similar to what was seen in the axial sensor. The lower amplitude could be caused by differences in the pin interfaces or could be caused by the Transmission3D carrier model being slightly stiffer than in reality. There is once again a small once per carrier revolution component in the measured signal. The previous sources of face run-out and carrier misalignment could be applied to this component as well. These two signals are used to determine the radial motion of the carrier and Figure 2.30 shows the exaggerated radial displacement of the Transmission3D carrier.
Figure 2.28: Sensor 040 radial displacement comparison, zero-mean.

Figure 2.29: Sensor 310 radial displacement comparison, zero-mean.
2.7 Planet Bearing Load

One of the targeted measurements for the collaborative was the load distribution of the planet bearings. The planet bearings were identified as a critical component due to the number of failures seen in operational gearboxes. A measurement scheme was developed to evaluate the load on the planet bearings. To achieve this, strain gages were mounted on the inner race of the planet bearings. There are six total planet bearings, two under each planet. Each bearing has three axial grooves machined in the inner diameter of the inner raceway. One groove is located at top dead center (TDC) on each bearing. This is in the heart of the load zone and gives an indication of the tangential load on this
bearing. There are strain gages placed at two axial locations in each groove, located at roughly 25% and 75% along the length of each raceway. The groove locations are shown in Figure 2.31, Figure 2.32, and Figure 2.33. The strain gages are intended to provide an indication of bearing load magnitude and the size of the load zone. The bearing load is obtained through a calibration procedure outlined in [17]. This document reviews the calibration rig and procedure used to calculate applied load from strain. The gages also allow for the calculation of bearing load share, both between planets and between rows.

Figure 2.31: Planet A bearing groove locations, picture courtesy of NREL [17].
Figure 2.32: Planet B bearing groove locations, picture courtesy of NREL [17].

Figure 2.33: Planet C bearing groove locations, picture courtesy of NREL [17].
2.7.1 Measured Race Strain Results

The strain gage measures the hoop strain of the bearing race. Figure 2.34 shows the bearing strain measurement for the TDC strain gage of Planet A. The high frequency content is due to the roller passing over the groove location and occurs at 9.9 Hz or 27.1 roller passes over one carrier revolution. A peak indicates that a roller is directly over the strain gage, whereas a valley indicates that the strain gage is equidistant between two rollers. Figure 2.35 shows the bearing strains for multiple carrier rotations. It can be seen that an envelope is created, the upper boundary being established when rollers are directly over the gage and the lower boundary is established when rollers straddle the gage. The once per carrier revolution variation is due to the tilt of the carrier caused by gravity and NTLs as were seen in the previous section.
Figure 2.34: Bearing strain measurement for AU_00_25, signal shows measurement for one carrier rotation.

Figure 2.35: Bearing strain measurement for multiple carrier rotations.
2.7.2 Transmission3D Model of Bearing race Strains

It was desired to compare simulated strain data to the measured strain data. With a model of this size, care should be taken to properly manage computation time. For this sensor comparison, the planet bearings needed to be fully modeled, this alone increases computation time three times. The fully modeled Transmission3D planet bearings are shown in Figure 2.36; these bearings have a radial manufacturing clearance of 141 micron as provided by NREL. The upwind and downwind bearing roller load distributions can be seen in Figure 2.37 and Figure 2.38. In addition, the roller pass frequency is 9.71 Hz; therefore, many more time steps must be included in the simulation to avoid aliasing of this signal. In situations like this, it is beneficial to devise solutions which provide the desired data in an efficient manner. As mentioned previously and as seen in Figure 2.35, there is a strain envelope developed by the roller pass. The modeling approach was then to create this envelope using the model. The procedure was to run three time steps with a roller directly over the gage location and another three with the gage location between two rollers. This would in essence produce the simulated strain envelope.
Figure 2.36: Planet gear with fully modeled planet bearings, minimum principal stresses are shown.
Figure 2.37: Planet A upwind bearing. Fully modeled, showing contact load distribution on rollers. This is with the carrier at the 0 degree location.
Figure 2.38: Planet A downwind bearing. Fully modeled, showing contact load distribution on rollers. This is with the carrier at the 0 degree location.

The bearing groove allows for placement of the strain gages and also provides strain amplification. The Transmission3D model does not include a groove in the raceway, so it is expected that strain levels will be a bit lower than the measurements, but trends should be similar. Assumptions must be made to allow for data comparison under these conditions. The first being that the groove geometry merely amplifies the strain by a constant, and that the amplification constant is the same for both peaks and valleys.
The second is that the strain amplification factor is constant for this range of bearing loads. As shown in Figure 2.35, there is a once per carrier revolution variation in the planet bearing strains. If the groove assumptions are upheld, the ratio of the maximum peak to peak strain value to the minimum peak to peak value should be comparable to the experimental data. Figure 2.39 shows this ratio is equal to 1.37 for the simulated bearing strains. The ratio gathered from experimental strain data in Figure 2.35 is 1.33, which compares very well to the simulated data. This would indicate that the bearing load variation is very similar in both the experimental gearbox and Transmission3D model. Figure 2.40 shows the calibrated planet bearing loads for Planet A from the measured strain data. Figure 2.41 shows the simulated planet bearing loads for Planet A. The simulated bearing loads compare very well to the calibrated bearing loads in mean, peak to peak amplitude, and phase. The once per carrier revolution load variation is seen in the bearing loads just as it was seen in the bearing strain. Once again, this is due to the tilt of the carrier from gravity and NTLs. The upwind and downwind bearings are out of phase by 180 degrees and as the carrier rotates the load is transferred from the upwind bearing to the downwind bearing. The ratio of upwind bearing load to downwind bearing load is an indication of the load distribution on the planet gear. If the upwind bearing carries more load than the downwind bearing it suggests that the load distribution favors the upwind side of the gear tooth. There is a mesh frequency bearing load variation as well but it is not seen in the simulated data. This is because the simulation was sampled such that at each time step the planet gears were at the same location in the mesh cycle. In essence, this filters out the mesh frequency effect. A calibration technique can also be
devised from the model. If the bearing load is divided by the instantaneous peak to peak strain value a calibration factor can be created and used with measured strain values to get bearing loads.

Figure 2.39: Simulated strain envelope for AU_00_25.
Figure 2.40: Planet bearing loads calibrated from measured strains, data is shown for two rotations of the carrier.

Figure 2.41: Planet A tangential bearing loads (simulated).
2.8 Planet Rim Motion and Deflection Sensors

The axial motion of two of the planet gear rims, Planet B and Planet C, were measured at three positions [18]. The sensor locations are shown in Figure 2.42. The proximity sensors were mounted on the carrier, see Figure 2.43, so all measurements are relative to the carrier motion. There are three sensors mounted approximately 90 degrees from each other. Downwind deflection is positive.

![Figure 2.42: Planet gear rim sensor locations, picture courtesy of NREL [18].](image-url)
2.8.1 Transmission3D Model of the Planet Rim Displacement

The measurements were made at a distance of 175mm from the planet center, see Figure 2.44. In the model, this distance is in the middle of the tooth mesh which is not ideal for data extraction. In the model, a compliant shaft segment was placed over this portion of the gear in order to attain a more uniform finite element mesh for data extraction. This was a very similar approach as used in the carrier motion study. The planet rim and mesh are shown in Figure 2.45.
Sensor mounts were built into the Transmission3D carrier model as shown in Figure 2.46. A script was written to extract data at each node on the planet rim. The data extracted included displacement information as well as nodal position in the fixed frame. A second script was written to extract the same data at the carrier sensor locations. A search routine was used to find the node on the planet rim closest to the sensor of interest. The difference between the axial displacement of the planet rim and carrier sensor provide the simulated sensor output.

Figure 2.44: Planet rim measurement locations, picture courtesy of NREL [18].
Figure 2.45: Transmission3D planet gear, showing rim segment where measurements were made.
Figure 2.46: Transmission3D planet carrier, showing sensor mount locations.
2.8.2 Comparison of Experimental and Predicted Planet Rim Displacement

The zero-mean simulated and experimental sensor outputs are shown in Figure 2.47, Figure 2.49, and Figure 2.50. The data compares quite well with the exception of the PlanetB_90 sensor as seen in Figure 2.49. This could be attributed to differences in the sensor mounting locations. This measurement is a relative displacement, therefore, the deflection of the sensor mount and sensor mount locations are included in the measurement. There are peaks in the experimentally measured signal that occur at the frequency of the planet gear revolution which is 56.6 rpm. The carrier rotates at 22.3 rpm meaning there are 2.54 planet rotations for every carrier rotation. Figure 2.48 shows the sensor output at the PlanetB_00 location for four full rotations of the planet carrier. There are ten such peaks in this signal and there would have been ten full rotations of the planet gear as well. It is likely that these are due to a nick or imperfection on the measurement surface at a point on the planet rim. If this were due to a physical change in the system it would be expected to see the peaks occur at the same location for every rotation of the carrier. However, this is not the case and they repeat every two carrier rotations. This also points to a surface flaw because due to the ratio of the planet with respect to the carrier being 2.54, the planet is at the same position with respect to the carrier every two rotations as this provides the lowest integer multiple of planet rotation. The high frequency content seen in the simulated data is an aliased mesh frequency effect which is at 36.3 Hz. As the gear rolls through mesh, the load distribution changes and the location of the resultant tooth force moves along the face width of the tooth, this has been
labeled as shuttling force [14]. This is what induces the mesh frequency variations in the measured and simulated signals and is seen in both the bearing loads and planet motion.

Figure 2.47: Zero–mean planet rim sensor comparison for PlanetB_00.
Figure 2.48: Experimentally measured planet rim displacement as measured by sensor PlanetB_00. Four full carrier revolutions are shown.

Figure 2.49: Zero-mean planet rim sensor comparison for PlanetB_90.
Figure 2.50: Zero-mean planet rim sensor data comparison for PlanetB_180.

These three sensors were used to calculate the planet motion relative to the carrier. This can be useful to ascertain the misalignment at the gear meshes. The planet motion is calculated as per the equations below [18].

\[
U_z = \frac{\text{Sensor00} + \text{Sensor180}}{2} \tag{2.1a}
\]

\[
d\theta_x = \arctan \left[ \frac{\text{Sensor00} - \text{Sensor180}}{2R} \right] \tag{2.1b}
\]

\[
d\theta_y = \arctan \left[ \frac{U_z - \text{Sensor90} + R\cos(80) \cdot \tan(d\theta_x)}{R\sin(80)} \right] \tag{2.1c}
\]
The angular misalignment is the main parameter of interest as this will have the greatest impact on gear tooth load distributions and planet bearing loads. The misalignment about the x-axis, $d\theta_x$, will have the greatest impact on load distributions as the majority of this misalignment is in the line of action. The calculated planet motion from both the experimental and simulated is shown in Figure 2.51, Figure 2.52, and Figure 2.53. The experimental data shows data from multiple rotations of the carrier. The signals all compare well, the main differences are seen in the phase of the signals.

![Figure 2.51: Planet axial displacement comparison.](image)
Figure 2.52: Planet angular misalignment comparison about the y-axis.

Figure 2.53: Planet angular misalignment comparison about the x-axis.
Conclusions

The model performs very well in matching the experimental signals. There are certain discrepancies which may be attributed to sensor mounting differences between the model and the test gearbox. As mentioned earlier, the planet bearing loads were a critical component, therefore, it was important to match data well in this regard. Many of the signals are affected by the same system parameters, and the alignment of the planetary system components. After performing this model validation, it is reasonable to assume that the other model results are indicative of real gearbox phenomena.
CHAPTER 3

SUN MOTION DUE TO CARRIER MANUFACTURING ERRORS AND THE EFFECT ON PLANET LOAD SHARE AND GEAR LOAD DISTRIBUTION

3.1 Carrier Motion Measurement and Calculation Validation

The motion of the GRC gearbox carrier was obtained through measurements of the carrier rim, as described in Chapter 2. Since the sun gear proximity sensors are mounted on the carrier, the carrier motion must be removed to get the absolute motion of the sun gear. There are deformations of the carrier rim due to the large torque in this section of the gearbox. These deformations are measured by the proximity sensors; however, they are not indicative of the motion of the whole carrier. Since only two sensors are used to calculate the carrier motion, it is possible that these deformations are not properly removed from the calculation. If this is the case, spurious carrier motions will be obtained from the experimental data. The model can be used to test this hypothesis. Two carrier motion measurement methods are compared in the following section.
3.1.1 Method 1 for Carrier Motion Measurement and Calculation

Method 1 obtains carrier motion using the simulated sensor displacements as discussed in Chapter 2. This method follows the NREL methodology outlined in the GRC documentation [13]. Figure 3.1 shows the radial measurements which were used for calculating the radial carrier motion in the X and Y axes. It was shown in Chapter 2 that the measured and simulated carrier sensor data compared very well. The carrier motion is obtained from the sensor output through use of the equations below.

\[
U_x = \frac{\text{Radial310} \cdot \cos(40) - \text{Radial040} \cdot \cos(50)}{2} \quad (3.1a)
\]

\[
U_y = \frac{\text{Radial310} \cdot \sin(40) + \text{Radial040} \cdot \sin(50)}{2} \quad (3.1b)
\]
3.1.2 Method 2 for Carrier Motion Measurement and Calculation

As discussed previously, there was a compliant shaft segment placed at the carrier rim measurement surface in the model. This shaft segment was used to provide an axisymmetric finite element mesh for more efficient data extraction. Method 1 averages the two sensor outputs to obtain the radial motion of the carrier. As discussed previously, the carrier will deform significantly under the applied torque and will experience deformation as well as rigid body displacements. Both of these are picked up by the sensor, however, only the rigid body motion is of interest. Ideally the deformations will
be averaged out however, since there are only two sensors used to calculate radial carrier motion in Method 1, this may not be the case. In the model, a shaft segment was placed on this part of the carrier, as seen in Figure 3.2. The displacement of every node around the circumference of the shaft was extracted and averaged. This is in essence the same approach as Method 1, however, there are 96 virtual sensors in this method such that when the carrier rim displacements are extracted and averaged the deformations will truly be averaged out and the true radial displacement of the carrier will be all that remains.

Figure 3.2: Shaft segment used for Method 2.
Method 2 predicts the true motion of the carrier in the model, and it can be seen in Figure 3.3 and Figure 3.4 that the two methods produce very different results. Figure 3.3 shows the displacement of the carrier in the X-axis. Method 1 and Method 2 both have a 3 per revolution component; however Method 1 has much lower amplitude and includes other higher order frequency content. Figure 3.4 shows the carrier displacement in the Y-axis. Method 1 shows a distinct six per revolution component whereas Method 2 has only a three per revolution component. The three per revolution component seen in Method 2 has a physical explanation and is due to the planet pass. The six per revolution component in Method 1 is caused by the deformations of the carrier rim added to the planet pass. Recall that the radial sensors had a similar six per revolution component that was due to the deformation of the carrier rim, shown again in Figure 2.28 and Figure 2.29. These are not rigid body displacements of the carrier. However, since there are only two sensors used in Method 1 they are not averaged out as they are in Method 2. Through this exercise it has been shown that the experimental method employed may lead to spurious carrier motions. Two radial sensors are not sufficient for averaging out deformations of the carrier rim. Therefore, it may be beneficial to devise another measurement technique or add additional sensors.
Figure 3.3: Predicted carrier motion comparison, X-motion.

Figure 3.4: Predicted carrier motion comparison, Y-axis.
3.2 Sun Motion Measurement

The gearbox rebuild included the implementation of a floating sun configuration in the planetary stage. The intended goal of this modification was to improve planet load sharing. For a three planet system, such as the GRC gearbox, manufacturing errors should be perfectly shared as long as the sun is able to float enough to compensate for the errors. Due to the high loads in this stage of the gearbox it is possible that flexibility of the structures may impact load sharing.

The sun gear orbit was measured in dynamometer testing. The measurement scheme employed two proximity sensors that were mounted on the carrier and measured the most upwind portion of the sun gear shaft. Figure 3.5 and Figure 3.6 show the sensor location and measurement location of the sun gear shaft. The measurements are taken at 90 degrees to one another.
Figure 3.5: Proximity sensor locations, picture courtesy of NREL [19].

Figure 3.6: Picture of sun gear proximity sensors [19].
The sun motion is measured relative to the carrier. As stated earlier, the three planet system will share load equally between the planets as long as the sun gear is able to float enough to compensate for manufacturing and assembly errors. The model was used to input manufacturing errors and to evaluate the induced sun motion and its effect on planet load sharing. Two types of errors were used for this purpose, pin position error and planet gear run out. Pin positioning error is placed tangent to the carrier rotation where it is expected that the sun will need to reposition itself to compensate for this error. Tangential pin errors cause one planet to come into contact earlier than the other planets in the system, thereby carrying a larger portion of the load. A radial position error could also have been used, but it will have negligible effect on sun motion and planet load share. Radial pin position errors simply shift the contact up the tooth flank but do not cause the teeth to come into contact early unless the radial pin error is exceptionally large which is not expected. Figure 3.7 shows the measured sun motion, showing a peak to peak motion of 150 micron. The goal of the following analysis was to input an error which would induce sun motion of this magnitude and evaluate the effect on planet load share. If sun motion of this magnitude can be induced without impacting planet load sharing, then the floating sun arrangement has achieved its goal.
Figure 3.7: Experimentally measured sun motion provided by NREL.

The sun is able to float by the use of a heavily crowned spline at the downwind portion of the shaft, as shown in Figure 3.8. This spline mates with the hollow shaft. The large lead crown magnitude reduces the bending stiffness of the spline coupling, allowing the sun to float as needed. Figure 3.9 shows a close up of the spline model. A detailed analysis of this spline is performed in the next chapter.
Figure 3.8: Transmission3D planetary stage components.

Figure 3.9: Transmission3D sun gear, shaft and spline
3.2.1 Sun Motion Simulation

As mentioned earlier, manufacturing errors were introduced into the model to assess sun motion and planet load share capability. Pin position error was placed on Planet A as shown in Figure 3.10. Tangential errors of 75 micron and 150 micron were both simulated; both were placed on Planet A. Recall that the measured peak to peak sun motion was approximately 150 micron which would correspond to an error in the range of 75 micron. The goal of this analysis was not to match the specific frequency content but rather to match the extremes of the sun motion. If the modeled sun motion is able to encompass the measured sun motion and the load sharing in the model is not affected, then it can be assumed that the load sharing of the actual gearbox is not affected either. Radial pin position errors were not considered as these have a negligible effect on planet loads and sun motion. A simulation with both 75 micron pin position error and a 50 micron planet run out were also considered. The error positioning is shown in Figure 3.11, the pin position error was placed on one planet (Planet A) and run out was put on a second planet (Planet B).
Figure 3.10: Pin position error.
Figure 3.11: Pin position error and planet runout.

As seen in Figure 3.12 and Figure 3.13, the sun motion for the nominal case with no manufacturing errors has little variation. There is a mean displacement due to the rigid body motion of the entire gearbox, but there is no variation as the carrier rotates. The pin position errors induce a once per carrier revolution sun motion. The peak to peak sun motion for the pin error simulations are summarized in Table 3.2.
<table>
<thead>
<tr>
<th>Pin Position Error Magnitude</th>
<th>Peak to Peak Sun Motion</th>
</tr>
</thead>
<tbody>
<tr>
<td>75 µm</td>
<td>184 µm</td>
</tr>
<tr>
<td>150 µm</td>
<td>357 µm</td>
</tr>
</tbody>
</table>

Table 3.2: Peak to peak sun motion for pin position error simulations.

The peak to peak sun motion increases by 94% when the pin position error is increased by 100%. This would indicate a nearly linear relationship between sun motion and pin position error, however, additional error magnitudes should be run before drawing far reaching conclusions. Planet runout adds a sun motion component at the frequency of planet rotation. This is superimposed on the motion due to the pin position error.
Figure 3.12: Simulated sun motion due to various manufacturing errors, x-axis.

Figure 3.13: Simulated sun motion due to various manufacturing errors, y-axis.
An ideal floating sun configuration would allow unconstrained radial motion of the sun in order to achieve equal planet load sharing. This is not the case for the GRC gearbox. As stated earlier, the sun is cantilevered from the spline on the downwind portion of the shaft. Radial motion of the sun is achieved by rotation/tilt at the spline location. This is not pure radial motion and introduces angular misalignment of the planet-sun meshes. The experimental set-up is not capable of determining the angular misalignment, but this data can be extracted with ease from the Transmission3D model.
3.2.2 Sun Displacement Method

The nodal displacement was extracted at every node around the circumference of the inner diameter of the sun shaft. These were then averaged to obtain the displacement of the centroid. Figure 3.15, shows the two axial locations where the displacements were extracted. The total motion of the sun gear can then be determined using the below equations. The simulated sun gear motion for various manufacturing errors is shown in Figure 3.16.

Figure 3.15: Sun gear displacement.
\[ U_x = \frac{U_{x_1} + U_{x_2}}{2} \]  

\[ U_y = \frac{U_{y_1} + U_{y_2}}{2} \]  

\[ d\theta_x = \sin^{-1}\left(\frac{U_{y_2} - U_{y_1}}{FW}\right), \quad \text{FW} = \text{Facewidth} \]  

\[ d\theta_y = \sin^{-1}\left(\frac{U_{x_2} - U_{x_1}}{FW}\right), \quad \text{FW} = \text{Facewidth} \]

---

Figure 3.16: Angular misalignment of the sun gear about the X-axis for various manufacturing errors.
3.3 Planet Load Share

The floating sun arrangement was implemented with the intent of equalizing load share between the planets. Planet load share is quantified as the ratio of total load on the planet to average planet load.

\[
\text{Planet Load Share} = \frac{\text{Planet Load}}{\text{Average Planet Load}}
\]  

(3.3)

Figure 3.17 shows the load share for Planet A plotted versus carrier rotation. It can be seen that there is a once per carrier revolution variation in load share. This is independent of manufacturing errors and is due to the static misalignment of the sun gear and carrier. This effect is likely due to gravity and non-torque loads.

![Figure 3.17: Planet A load share for various manufacturing errors.](image-url)
The manufacturing errors cause little change from the nominal case. After 270 degrees of carrier rotation, Planet A carries its largest load. The max load carried by the Planet A is 7% greater than would be seen in with perfect load share. The peak to peak and average load share are summarized for each manufacturing error in Table 3.3. The average load share increases as the pin position error increase. The increase is only 2% for a pin position error of 150µm.

<table>
<thead>
<tr>
<th>Error</th>
<th>Peak to Peak</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>0.1307</td>
<td>0.9998</td>
</tr>
<tr>
<td>75 µm Pin Error</td>
<td>0.1311</td>
<td>1.009</td>
</tr>
<tr>
<td>75 µm Pin Error &amp; Runout</td>
<td>0.1302</td>
<td>1.009</td>
</tr>
<tr>
<td>150 µm Pin Error</td>
<td>0.1287</td>
<td>1.018</td>
</tr>
</tbody>
</table>

Table 3.3: Summary of Planet A load share for various manufacturing errors.

The load share for Planet B and Planet C, as seen in Figure 3.18 and Figure 3.19, exhibit the same behavior as Planet A. Once again, there is a large once per carrier revolution load share variation. Recall, Planet B is the gear with the run out error. It can be seen in Table 3.4 that for the case of run out error, the peak to peak load share is affected on the gear which has the run out. However, the increase in peak to peak load share is only 1.5% greater than the nominal case.
Table 3.5 summarizes the load share characteristics of Planet C for the modeled manufacturing errors.

Figure 3.18: Planet B load share for various manufacturing errors.

<table>
<thead>
<tr>
<th>Error</th>
<th>Peak to Peak</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>0.1297</td>
<td>0.998</td>
</tr>
<tr>
<td>75 µm Pin Error</td>
<td>0.1305</td>
<td>0.994</td>
</tr>
<tr>
<td>75 µm Pin Error &amp; Runout</td>
<td>0.1434</td>
<td>0.994</td>
</tr>
<tr>
<td>150 µm Pin Error</td>
<td>0.1308</td>
<td>0.990</td>
</tr>
</tbody>
</table>
Table 3.4: Summary of Planet B load share for various manufacturing errors.

<table>
<thead>
<tr>
<th>Error</th>
<th>Peak to Peak</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>None</td>
<td>0.1312</td>
<td>1.0017</td>
</tr>
<tr>
<td>75 µm Pin Error</td>
<td>0.1307</td>
<td>0.9963</td>
</tr>
<tr>
<td>75 µm Pin Error &amp; Runout</td>
<td>0.1298</td>
<td>0.9975</td>
</tr>
<tr>
<td>150 µm Pin Error</td>
<td>0.1286</td>
<td>0.9913</td>
</tr>
</tbody>
</table>

Figure 3.19: Planet C load share comparison for various manufacturing errors.
Table 3.5: Summary of Planet C load share for various manufacturing errors.

In summary, the floating sun arrangement utilized in the GRC gearbox is able to mitigate the manufacturing errors. The simulations show that there is enough freedom for the sun to accommodate a manufacturing error of 150 µm and still maintain nearly the same load share as the nominal case. The floating sun does not address the issue caused by the misalignment due to weight and non-torque loads.

The peak to peak sun motion due to the 75 µm pin position error is 158 µm compared to the measured peak to peak sun motion of 150 µm. This would indicate that the system is operating within the boundaries of tolerable manufacturing error. Since a sun motion of 158 µm can be induced without adversely affecting load share, it is expected that the GRC gearbox will not have load share issues since the sun motion is within this range.

The once per carrier revolution load share variation could be addressed by using more rigid carrier bearings (PLC-A and PLC-B) a factor that will be addressed in the next gearbox redesign. The bearings at this location will be changed from cylindrical roller bearings to tapered roller bearings. Thereby removing the clearance and decreasing misalignment between the sun and the carrier.

3.4 Manufacturing Error Impact on Gear Load Distribution

It has been shown that the planetary set up can accommodate pin errors as large as 150 micron without adversely affecting the planet load sharing. However, this is not to say that the manufacturing errors do not affect the load distribution on the gears. As
shown previously, the ratio of the upwind planet bearing load to the total planet load is an indication of the load distribution on the gear teeth. A value of 0.5 indicates that the load is shared equally between the upwind and downwind bearings. A value greater than 0.5 indicates that the gear tooth load distribution favors the upwind portion of the face width. A value less than 0.5 would indicate that the load distribution is more concentrated on the downwind portion of the face width. The upwind bearing load share was calculated for the various manufacturing errors to determine whether the gear tooth load distribution is affected. Figure 3.20 shows the load share of the Planet A upwind bearing for each of the manufacturing errors. The manufacturing errors do have an impact on the ratio of the total load carried by the upwind bearing. In general, the load share decreases as the pin error increases and it appears that the runout does not impact load distributions on Planet A. The maximum change in load share is caused by the 150 µm pin error and changes peak to peak load share by 3% over the nominal case. This would suggest that the load distribution is improving, that is to say the load distribution is evening out and the resultant force is located more toward the center of the face width. Figure 3.21 shows the Planet B upwind bearing load share for each manufacturing error. The Planet B bearing exhibits the opposite behavior as the Planet A bearing. The Planet B upwind bearing load share increases as pin position error increases, implying that the load distribution shifts more toward the upwind portion of the face width of the gear. The maximum change is once again due to the 150 µm pin error which causes an increase of 3% in the load share of this bearing. Figure 3.22 shows the planet C upwind bearing load share for each manufacturing error. The load share, and thus the load distribution, is not as greatly
affected compared to the other two planets. The maximum change in load share is 1% from the nominal case. Figure 3.23 shows the load distribution on Planet A at the ring-planet mesh due to these manufacturing errors. The load distribution is shown for when Planet A is at 60 degrees where the planet bearing load share is worst. The manufacturing errors do not seem to create any visible shift in load distribution at this mesh location. However, in Figure 3.24, which shows the sun-planet load distribution, there is a downwind shift in load distribution as manufacturing errors increase. This is consistent with the planet load share prediction shown in Figure 3.20.

![Figure 3.20: Planet A upwind bearing load share. This is the ratio of the Planet A upwind bearing load to total load carried by the Planet A.](image-url)
Figure 3.21: Planet B upwind bearing load share. This is the ratio of the Planet B upwind bearing to the total load carried by Planet B.

Figure 3.22: Planet C upwind bearing load share. This is the ratio of the upwind bearing load to the total load carried by Planet C.
Figure 3.23: Load distribution on Planet A at planet-ring mesh due to various manufacturing errors. The load distributions shown are for when Planet A is at the 60 degree location, where the load planet bearing load share is the worst.
Figure 3.24: Load distribution on Planet A at planet-sun mesh due to various manufacturing errors. The load distributions shown are for when Planet A is at the 60 degree location, where the load planet bearing load share is the worst.
3.5 Conclusion

A pin position error of 75 micron was shown to induce a peak to peak sun motion of a similar magnitude to what was measured. The floating sun configuration was shown to accommodate errors as large as 150 micron without losing load share capability with respect to the no error condition. The planet load share is most largely affected by the misalignment of the carrier with respect to the other planetary components. This causes the once per revolution load share variation and is unaffected by manufacturing errors. The load share between the planet bearing rows can be used as an indication of gear load distributions. This can also be performed using the experimental data to surmise gear load distributions.
CHAPTER 4

SPLINE MODELING SUMMARY AND EFFECT OF SPLINE LEAD CROWN ON PLANET LOAD SHARE

As mentioned in Chapter 3 the sun spline is used for two purposes, to transmit torque from the sun gear to the hollow shaft and to allow the sun gear to float. The spline is located on the downwind side of the sun gear shaft, as shown in Figure 4.1. This external spline mates with the internal spline of the hollow shaft. The spline is crowned heavily, relative to the amount of expected tooth deflection. The reason for lead crown is typically to avoid edge loading, however there is a second purpose in this spline. As mentioned previously, the sun gear must be permitted to float in order to equalize planet load share. The float of the sun is provided by the motion at the spline, specifically, the angular misalignment of the spline permits the radial motion of the sun gear. Radial motion of the spline will contribute a negligible amount to the float of the sun. The bending stiffness of the spline is the key parameter when evaluating the efficacy of the sun gear to float. There is a tradeoff in this design. As lead crown increases, the stiffness
of the spline connection decreases, which allows more float in the sun and better planet load sharing. However, greater lead crown also increases spline contact pressure as the load is carried on a smaller portion of the spline tooth. The effect of spline lead crown on sun float, spline stiffness, and planet load share are evaluated in this chapter.

![Spline Sun Gear]

**Figure 4.1: Transmission3D sun gear and spline.**

### 4.1 Helical3D and Transmission3D Comparison

Two models were used in the spline analysis. Helical3D allows for the detailed modeling of gears and splines, however, it is limited to one gear mesh or contact pair.
Transmission3D allows for multiple gear meshes, however, the spline model is simplified. Helical3D allows the user to model full involute splines-and exactly match the specified geometry of the component, while Transmission3D only allows for straight sided splines. Splines are very computationally taxing due to the number of teeth in contact. The T3D model completes a simulation in less than ten minutes, compared to approximately three hours for the H3D model. The reason for this is that the T3D model makes an assumption of non-conformal line contact at the spline teeth while this is not the case for involute splines. H3D does not make this assumption and models the contact of two conformal surfaces present in involute splines. The effects of this will be seen in the profile load distributions and will be discussed in subsequent sections.

There are limitations on available modifications and manufacturing errors in T3D as well. H3D allows for full control over the topology of the spline tooth. The user can specify any profile or lead modification desired. Spacing errors and runout may also be included in the H3D model while T3D splines are limited to lead modifications. The H3D model will give the most accurate analysis; however, the T3D model can still be used for purposes such as stiffness evaluation and load intensity evaluation. Both are able to model contact on the back side of the tooth. This was not included in any of the following simulations. The misalignments involved were not great enough to suggest back contact was needed and would only have increased computation time by introducing more potential contact points.

A close up of the Transmission3D spline model is shown in Figure 4.2 and the Helical3D model can be seen in Figure 4.3. As mentioned previously, the two models
differ in their geometries. Transmission3D splines require a width, height, and pressure angle to define the overall geometry of the spline tooth and Helical3D uses more exact parameters to define the involute profile. Figure 4.4 and Figure 4.5 show the tooth geometries for the Transmission3D spline and Helical3D spline, respectively. One goal when generating the model is to match the overall geometry of the two models as closely as possible, that is to say, keep the tooth height and tooth thickness comparable.

Figure 4.2: T3D external spline model and mesh.
Figure 4.3: Helical3D external spline model and mesh.

Figure 4.4: Transmission3D spline mesh, internal spline (top) and external spline (bottom).
4.2 Model Comparison

The following section shows a comparison of the T3D and the H3D spline models. Three parameters are compared; total spline tooth load, spline tooth load intensity, and bending stiffness. The comparison was performed for three crown magnitudes; 180µm, 90µm and no crown. The purpose of this comparison was to validate the Transmission3D models performance in predicting these parameters. The T3D model is more preferred from the standpoint of computational efficiency. It also is possible to integrate this spline model into the entire gearbox. It is important to be aware of the capabilities and limitations of each model and this section aims to identify them. It should be pointed out that the detailed involute spline model is not available as an option in Transmission3D.

Figure 4.5: Helical3D spline mesh, internal spline (top) and external spline (bottom).
4.2.1 Modeling Approach

As mentioned previously, the full Transmission3D gearbox model utilizes a simplified straight sided spline model. One objective was to quantify the stiffness of the GRC sun spline. This stiffness can be used for general design purposes and can be used in other models such as lumped parameter models for dynamic analyses. This section will describe the method used for evaluating the spline stiffness.

First, a simulation was completed using the full Transmission3D model gearbox, as shown in Figure 4.6. A pin position error of 150um was included on Planet A. This was done to provide a basis for which to evaluate the effect of crown magnitude on planet load share and sun motion. The relative displacement of the external spline with respect to the internal spline was extracted. The nodal displacements were extracted at every node around the circumference of the shaft and then averaged. This was done at two axial locations as shown in Figure 4.7. This provided the displacement of the center of the shaft at these two locations. The displacement of this segment in the four degrees of freedom of interest can then be calculated using the following equations.
\begin{align*}
  dX &= \frac{X_1 + X_2}{2} \tag{4.1a} \\
  dY &= \frac{Y_2 + Y_2}{2} \tag{4.1b} \\
  d\Theta_X &= \sin\left(\frac{Y_2 - Y_2}{\text{Face Width}}\right) \tag{4.1c} \\
  d\Theta_Y &= \sin\left(\frac{X_2 - X_1}{\text{Face Width}}\right) \tag{4.1d}
\end{align*}

Figure 4.6: Transmission3D model of GRC gearbox, with the housing shown (left) and removed (right)
This process was completed for both the internal and external spline segment. The difference of these two gives the relative motion of the external spline with respect to the internal spline.

A second model shown in Figure 4.8 that isolates the sun shaft and the hollow shaft was then created. The hollow shaft was constrained and the relative displacement values were used as boundary conditions for this new model. The resulting forces and moments were then extracted. In this way the stiffness could be evaluated.
4.2.2 Analysis of Spline Model with Full Crown

The magnitude of the circular lead crown of the original design is 178 microns. Figure 4.10 and Figure 4.11 show the spline tooth load intensity for the Transmission3D and Helical3D models, respectively. The x-axis is the face width of the spline, which varies from -1, most upwind, to 1, most downwind. The y-axis is the tooth number as described by Figure 4.9. This plot can be viewed as if the spline was unwrapped and laid flat on the page. The load intensity is the load divided by the length of the contact patch. The two models compare well in this regard, matching very well in shape and phase. The maximum load intensity is 679.2 N/mm in the Transmission3D model and 596.2 N/mm in the Helical3D model, a difference of 13.9%. The Helical3D spline model carries load over a larger portion of the face width which lowers load intensity. The Helical3D model
utilizes 36.6% of the face width, whereas the Transmission3D model utilizes 29% of the total face width. This could be caused by differences in the stiffness of the models. As Figure 4.12 shows, the Helical3D spline is slightly more compliant than the Transmission3D model. This would indicate that the Helical3D model would deflect more than the Transmission3D model, which could result in this larger contact patch and as a result a lower load intensity.

As stated previously there is a slight difference in bending stiffness between the two models as shown in Figure 4.12. The Helical3D model is more compliant by approximately 5%. The difference could arise from geometric discrepancies. Also, the load is only carried at one profile location in the Transmission3D model whereas the Helical3D model allows for the load to be carried over the entire profile. As the load moves more toward the tip of the tooth, the moment will increase and so would tooth deflection. This potential for tip loading could also contribute to the differences between the two model stiffnesses.

It must be stated that all of these figures are generated from one position or time step; however, being that the sun gear rotates all spline teeth will see the peak load. Figure 4.13 shows a comparison of the total tooth load between the two spline models. Once again the shape and magnitude compare quite well between the two models. Each predict tooth 35 to carry the most load. It can also be seen that the Helical3D model carries more total load. The torque is identical in each model but the pitch diameter may be slightly different which could cause this effect. Figure 4.14 Figure 4.15 shows the differences between the two models regarding how tooth load is carried. As seen in
Figure 4.15 the Helical3D model allows for contact along the entire tooth profile. This particular spline includes no profile modification, which causes the load to shift toward the root and tip of the tooth, a type of contact that is often referred to as corner contact in the gear terminology. There are 108 potential contact points on one spline tooth in Helical3D, 18 along the face width and 6 along the profile. Figure 4.14 shows the load distribution of the Transmission3D model spline tooth. It can be seen that the load is carried only at the middle of the profile. In the Transmission3D model there were 25 potential contact points per tooth, all in the axial direction. The contours in the model pictures show Von Mises stresses and the raised bars show contact pressures. The Transmission3D contact pressure is not a valid quantification. Helical3D must be used to gather this information. Transmission3D assumes line contact of two non-conformal bodies; this is an issue because involute splines are conformal bodies. Helical3D does not make this assumption and models conformal contact.
Figure 4.9: Spline numbering scheme viewed from downwind.

Figure 4.10: Transmission3D spline tooth load intensity for 178 µm lead crown.
Figure 4.11: Helical3D spline tooth load intensity for 178 μm lead crown.

Figure 4.12: Bending stiffness comparison of H3D and T3D spline for 178 μm lead crown.
Figure 4.13: Spline tooth load comparison for 178 µm lead crown.
Figure 4.14: Transmission3D spline model with load distribution shown on spline teeth for 178 µm lead crown.
Figure 4.15: Helical3D contact pressure distribution for external spline tooth with 178 µm lead crown, blue dots are all potential contact points.

4.2.3 Analysis of Model with Half Crown

Half-crown refers to a lead crown magnitude of 90 µm, which is about 50% of the original design that is referred to as full crown. Figure 4.16 and Figure 4.17 show the spline tooth load intensity for the Transmission3D and Helical3D spline models, respectively. The two compare well in shape, however, the Transmission3D model’s maximum load intensity is 17.7% higher than the Helical3D model. The Helical3D model utilizes 50% of the face width, whereas Transmission3D utilizes 37.5% of the face width, which would contribute to this higher magnitude. Figure 4.21 shows the bending
stiffness comparison of the two models for this crown magnitude with the Transmission3D model being 10% stiffer than the Helical3D model. This could also contribute to the differences in load intensity for the same reasons described previously. The deviation in stiffness is larger for this crown magnitude but the two models still compare quite well. Figure 4.18 and Figure 4.19 show the contact pressure distributions for the Transmission3D and Helical3D splines, respectively. Once again, it must be stated that Helical3D contact pressure is a valid quantity, but the Transmission3D contact pressure is not. The contact pattern in the Helical3D model is shifted to the root and tip of the profile due to the lack of profile modifications. Figure 4.20 shows the comparison of the total load on each spline tooth of the two models. The results compare well in shape and magnitude. The Helical3D spline carries slightly more load than the Transmission3D spline due to differences in pitch diameters.
Figure 4.16: Transmission3D spline tooth load intensity for 90 µm lead crown.

Figure 4.17: Helical3D spline tooth load intensity for 90 µm lead crown.
Figure 4.18: Transmission3D spline model with load distribution shown on spline teeth, for 90 µm lead crown.
Figure 4.19: Helical3D contact pressure distributions for external spline tooth for 90 µm lead crown, blue points are potential contact points.
Figure 4.20: Spline tooth load comparison for 90 µm lead crown

Figure 4.21: Bending stiffness comparison of H3D and T3D for 90 µm lead crown.
4.2.4 Analysis of Spline with No Crown

This section details the comparison of the two models with no lead crown modification. Figure 4.22 and Figure 4.23 show the spline tooth load intensity for the Transmission3D and Helical3D spline models, respectively. The shape and magnitude compare well between the two models. There is only a 3% difference between the maximum load intensity of each model. Figure 4.27 shows there is a larger deviation of the bending stiffness of the two models than was previously seen, with Helical3D being more compliant by about 15%. As seen in the load intensity plots, there is a considerable amount of edge loading under these conditions. Figure 4.24 and Figure 4.25 show the contact pressure distributions for the Transmission3D and Helical3D splines, respectively. This edge loading can be seen in these conditions as well. Figure 4.26 shows the comparison of spline tooth load and the shape of the loading compares well, however, there is some deviation in magnitude. These differences are likely caused by the geometric differences of the two spline teeth models. Also, the fact that the Transmission3D spline carries load at only one profile position could potentially contribute to this discrepancy.
Figure 4.22: Transmission3D spline tooth load intensity for no lead crown modification.

Figure 4.23: Helical 3D spline tooth load intensity for no lead crown modification.
Figure 4.24: Transmission3D spline model with load distribution shown on spline teeth, no lead crown.
Figure 4.25: Helical3D contact pressure distribution for external spline tooth with no lead crown modification, blue points show potential contact points.
Figure 4.26: Spline tooth load comparison of Transmission3D and Helical3D models for no lead crown modification.

Figure 4.27: Bending stiffness comparison for no lead crown.
4.3 Summary of Spline Modeling Techniques

This section has shown that the simplified Transmission3D spline model compares very well, in load intensity and stiffness, to the full involute model in Helical3D, especially with large lead crown modifications. Therefore, it would be beneficial to use the Transmission3D model for evaluating certain design parameters. The user must determine what parameters are of interest and choose the appropriate modeling technique. For this particular scenario, the Transmission3D model is sufficient for evaluating the spline stiffness. This can then be used in lumped parameter models and also for evaluating the ability of the system to equalize the load. If the designer desires more detail, it would be advisable to first run T3D from which misalignments are transferred to Helical3D for tooth stress analysis. Contact pressures should only be gathered from Helical3D simulations. It was also shown that decreasing the lead crown modification will decrease the maximum contact pressure; however, the spline stiffness is increased. The maximum contact pressure decreased by 10.7% when the crown was decreased from 178 micron to 90 micron, but resulted in a 90% increase in spline stiffness. The maximum contact pressure increases when there is no crown on the spline due to edge loading. The effect of spline stiffness on load sharing is evaluated in the next section. As seen in the Helical3D profile load distribution, the load is carried primarily at the root and tip of the tooth. The middle of the profile is largely unused. It would be advisable to consider profile modifications, such as tip relief or profile crown. This would reduce contact pressures at the tooth tips.
4.4 Impact of Spline Lead Crown on Sun Motion and Planet Load Share

This section evaluates the effect of lead crown magnitude and spline stiffness on sun motion and planet load share. The stiffness evaluation method was outlined in the previous section and it was shown that the amount of lead crown greatly affects the bending stiffness of the spline. As stated previously, the full gearbox model is built within Transmission3D and utilizes the straight-sided spline model. The previous section demonstrated that the Transmission3D spline model compared well to the more detailed Helical3D model. This was an important validation because only the straight sided spline model can be integrated into the full GRC gearbox model.

Quadrant one of Figure 4.28 shows the bending stiffness of the sun spline with full lead crown. This bending stiffness is evaluated using the procedure presented in Section 4.2.1. Quadrant two shows the spline moment required to induce the spline misalignment which is shown in quadrant four. Quadrant three shows the sun motion that causes this spline misalignment. All of these values are measured in the fixed frame and plotted against carrier rotation. As the carrier rotates, so does the pin position error and causes the sun to float and this causes the spline misalignment. The spline stiffness varies by 5% over one carrier revolution which would suggest that a constant value could be used in lumped parameter models. Figure 4.29 shows the bending stiffness for multiple lead crown magnitudes and it can be seen that for all crown magnitudes there is nearly no variation in stiffness as the carrier rotates. The bending stiffness for the no lead crown spline is nine times stiffer than the full design crown magnitude (178um). Reducing the crown by 50%, to 90 micron, increases stiffness by 90%. Figure 4.30
shows the average bending stiffness plotted against lead crown magnitude and it can be seen that the stiffness nearly doubles as lead crown is reduced by half. The pin position error creates an unequal load between the planets, where one planet is loaded more than the others. This uneven loading at the sun-planet mesh causes a moment at the spline.

Figure 4.28: Spline bending stiffness, Y-axis sun motion, spline misalignment and spline moment. Transmission3D model, full lead crown.
Figure 4.29: Bending stiffness for multiple lead crown magnitudes, Transmission3D model.
As stated previously, the full Transmission3D GRC gearbox model was used for these analyses. There was a 150 micron pin position error placed on Planet A in order to assess the effect of lead crown on sun motion and planet load share. Figure 4.31 shows the sun motion along the x-axis for multiple lead crown magnitudes. The mean value decreases as crown magnitude decreases, as does the peak to peak value. The mean value decreases by 33.7% when going from full to no crown and the peak to peak value decreases by 19%. Figure 4.32 shows the sun motion in the y-axis for multiple spline lead crown magnitudes. The mean sun y-motion decreased by 51.1% when going from full to no crown, and the peak to peak value decreased by 31.4%. Figure 4.33 shows the
sun orbit for multiple crown magnitudes which is the y-motion plotted against the x-motion and the stiffening effect of crown magnitude can be seen by reducing the orbit diameter and moving the center towards zero. The spline stiffness increases by 900% when going from 178 µm crown to no crown, whereas the peak to peak sun motion only changes by a maximum of 34%. There is still a substantial amount of sun motion, even with no lead crown.

Figure 4.31: Sun motion in the x-axis for multiple crown magnitudes.
Figure 4.32: Sun motion in the y-axis for multiple spline crown magnitudes.

Figure 4.33: Sun orbit, Y motion vs. X motion for multiple spline crown magnitudes.
The planet bearing loads are affected by the spline stiffness as well and are shown in Figures 4.34-4.39. The spline crown affects the peak to peak, mean and phase of the bearing loads. The effect is more pronounced on the upwind bearings and it appears that the largest impact is on the phase of the loading with a 30 degree phase difference caused by removing the lead crown modification. The maximum planet bearing loads are summarized in , which is a design parameter of interest. For the full crown spline, the upwind Planet B bearing is the highest loaded bearing with a load of 239.7 kN. When the lead crown is removed from the spline the bearing carrying the most load is the upwind Planet A bearing with a maximum load of 247.1 kN. By removing the lead crown the highest loaded bearing carries 3% more load. Once again, removing the lead crown increases stiffness by 900%, but results in a 3% increase in load carried by the highest loaded bearing. The peak to peak load on the upwind bearing of Planet A (this is the planet with the pin position error) increases by 22.7% when going from full to no crown and by only 0.7% on the downwind bearing. The mean load increases by 7.4 kN on the upwind bearing (3.4%) and decreases by the same amount on the downwind bearing (4.3%). The Planet B and C bearings exhibit a similar response and the peak to peak change is once again more pronounced on the upwind bearing however, the change on downwind peak to peak loading is larger than was seen in Planet A.
Figure 4.34: Planet A upwind bearing load.

Figure 4.35: Planet A downwind bearing load.
Figure 4.36: Planet B upwind bearing load.

Figure 4.37: Planet B downwind bearing load.
Figure 4.38: Planet C upwind bearing load.

Figure 4.39: Planet C downwind bearing load.
<table>
<thead>
<tr>
<th></th>
<th>100% Crown</th>
<th>50% Crown</th>
<th>25% Crown</th>
<th>No Crown</th>
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<tr>
<td></td>
<td>Max (kN)</td>
<td>Max (kN)</td>
<td>Max (kN)</td>
<td>Max (kN)</td>
</tr>
<tr>
<td><strong>Planet A</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Upwind</td>
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<td>232.9</td>
<td>237.5</td>
<td>247.1</td>
</tr>
<tr>
<td>Downwind</td>
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<td>214.6</td>
<td>212.7</td>
<td>210.2</td>
</tr>
<tr>
<td><strong>Planet B</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
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<td>238.6</td>
<td>237.8</td>
<td>240.8</td>
</tr>
<tr>
<td>Downwind</td>
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<td>197.1</td>
<td>200.0</td>
<td>205.1</td>
</tr>
<tr>
<td><strong>Planet C</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Upwind</td>
<td>227.3</td>
<td>226.7</td>
<td>226.3</td>
<td>228.3</td>
</tr>
<tr>
<td>Downwind</td>
<td>206.9</td>
<td>209.4</td>
<td>212.3</td>
<td>216.0</td>
</tr>
</tbody>
</table>

Table 4.6: Summary of maximum planet bearing loads for multiple lead crown magnitudes.
The restriction of sun motion results in a change in planet load sharing. Figures 40-42 show the planet load sharing for multiple crown magnitudes and Table 4.7 summarizes the planet load sharing for these crown magnitudes. It can be seen that the dominant effect of lead crown on load sharing is on the peak to peak value whereas the mean value is unaffected by the change in lead crown. The peak to peak load share increases by 65.1% for Planet A, 73.3% for Planet B, and 62% for Planet C. This increase in peak to peak load share is due to the restriction of sun gear motion as previously discussed. Recall, that the phase, or location, of the peak bearing load changed due to lead crown magnitude. This was predominantly seen on the upwind planet bearing and the location of maximum load shifted towards the location of maximum downwind bearing load, thus increasing the total load on the planet. The maximum bearing load increased by 3% when going from full crown to no crown, once again showing that the crown effects the location of the maximum more than the maximum itself. Therefore, maximum tooth stresses will increase but maximum bearing stresses will remain nearly constant.
Figure 4.40: Planet A load share for multiple crown magnitudes.

Figure 4.41: Planet B load share for multiple crown magnitudes.
Figure 4.42: Planet C load share for multiple crown magnitudes.
<table>
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<th>Planet</th>
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<th>50% Crown</th>
<th>25% Crown</th>
<th>No Crown</th>
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<tbody>
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<td>Mean</td>
<td>PTP</td>
<td>Mean</td>
</tr>
<tr>
<td>Planet A</td>
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<td>1.02</td>
<td>0.147</td>
<td>1.02</td>
</tr>
<tr>
<td>Planet B</td>
<td>0.131</td>
<td>0.99</td>
<td>0.156</td>
<td>0.99</td>
</tr>
<tr>
<td>Planet C</td>
<td>0.129</td>
<td>0.99</td>
<td>0.147</td>
<td>0.99</td>
</tr>
</tbody>
</table>

Table 4.7: Summary of planet load share for multiple crown magnitudes.
4.5 Conclusion

In this section two spline models were compared and the Transmission3D model was validated against the Helical3D model for purposes of stiffness and load intensity evaluation. The Transmission3D model was then used to evaluate the stiffness of the spline with various lead crown magnitudes. The effect of lead crown on sun orbit was evaluated and it was shown that as lead crown magnitude decreased, so did the sun motion due to the stiffening of the spline connection. It was also shown that the spline lead crown has an impact on planet load share, specifically the peak to peak planet load share but no effect on the mean value. Although there is an increase in the peak to peak load share, the maximum bearing load only increases by 3%. This could suggest that the initial lead crown magnitude of 178 µm may be more than necessary as there is still a substantial amount of float of the sun even with no lead crown. During the spline model validation, the Helical3D model was able to provide contact pressures for the full, half, and no crown splines. It was shown that the maximum contact pressure decreased from 234 MPa to 209 MPa, a decrease of 10.7%. This change in crown only caused the most highly loaded bearing (Planet A upwind bearing) to carry 1.5% more load. Therefore, if spline failure is a concern, it may be beneficial to reduce the spline crown thereby achieving lower contact pressures. The most critical component must be identified and the design can be optimized accordingly. Another solution would be to address the static misalignment of the sun shaft with respect to the carrier which would address a portion of the peak to peak load share variation. Larger error values can be tested to see if these trends continue or if there is a mean change in planet load sharing.
CHAPTER 5

HIGH SPEED SHAFT, TAPERED ROLLER BEARING GROOVE
GEOMETRY VALIDATION

5.1 Introduction

The high speed stage gears and bearings have been identified as critical components of the gearbox [3] [5]. Previously there had been very little instrumentation of the high speed shaft, gears, and bearings. During the next phase of testing, this portion of the gearbox will be instrumented. In this next phase, tests will be developed to evaluate the loads on the high speed tapered roller bearings (TRBs) [20]. The high speed shaft extends out of the gearbox into a flange that supports the brake disk. Between the brake disk and the generator is a coupling that uses flexible links to allow axial, angular, and radial misalignment of the gearbox [20]. Excessive misalignment of the gearbox is theorized to be a potential cause of high bearing loads; therefore, generator misalignment
will be measured as part of the next test plan. In order to enhance the assessment of the high speed stage, three sets of bending gages will be placed on the shaft to define shaft bending loads on both sides of the high-speed gear mesh. An additional eight gages are to be installed in the root of the teeth of the high speed pinion to measure face width load distribution. Gages are also placed in axial grooves which are machined into the outer race of the TRBs. Each bearing has four axial grooves with two gages per groove.

There was initial concern as to whether the strain levels would be high enough in the bearing raceways to provide an adequate signal to noise ratio. A proposed groove design was provided by NREL and is shown in Figure 5.1. The groove is 15 mm wide and 6 mm at its deepest point. The outer race is tapered to begin with, so the groove geometry was tapered as well, with the intent of maintaining a constant radial thickness of the groove as seen in Figure 5.2. The Transmission3D model was used to model the high speed stage with the bearings fully modeled in order to evaluate strain levels.

![Figure 5.1: View of high speed Transmission3D TRB groove geometry.](image)

15 mm

6mm
Figure 5.2: Transmission3D TRB outer race section cut showing groove geometry.

5.2 Transmission3D Model Build-up

The Transmission3D Model was built with gear contacts at all locations and bearing stiffnesses at all locations with the exception of the high speed stage, which had all three bearings (HS SH A, HS SH B, and HS SH C) fully modeled. The Transmission3D model of the high speed, intermediate speed, and low speed stages can
be seen in Figure 5.3. The fully modeled bearing can also be seen and are shown close-up in Figure 5.4 with bearing names displayed.

Figure 5.3: Transmission3D model showing high speed, intermediate speed and low speed shaft stages.
Figure 5.4: Transmission3D high speed shaft stage with bearings fully modeled.

The modified raceway is not a native option in Transmission3D; instead it must be built and exported in much the same way as the housing and carrier structures. The modified raceway was built and meshed in SolidWorks, then the mesh was exported as a Nastran file and the cvtbd utility was used to convert the files for use in Transmission3D. The modified race was imported as a housing structure in Transmission3D. Due to the workarounds employed to include the modified raceways, the whole housing structure could not be included because there is not an option to mate two separate housing components directly. Instead, the gearbox housing was eliminated and substituted with a simplified “canister” model. In this model the low speed,
intermediate speed, and high speed shaft stages were included. Figure 5.5 shows the entire gearbox and housing model and Figure 5.6 shows the mid-plane section cut of the model. It can be seen that the high speed TRBs are housed in a tapered sleeve within the housing. This was still included in the simplified model, because deformation of the housing could affect how load is carried between the TRBs. Figure 5.7 and Figure 5.8 show the simplified “canister” model. All bearings are fixed to ground with the exception of the TRBs which are housed in the tapered sleeve which represents the sleeve that contains them in the actual housing model. The tapered sleeve is fixed to ground at the most downwind portion of the shaft as shown in Figure 5.7 and Figure 5.8.

Figure 5.5: Entire gearbox model, showing entire housing.
Figure 5.6: Gearbox section cut showing the housing sleeve that contains the high speed TRBs.
Figure 5.7: Simplified "canister" model showing all gear stages and bearings.
An analysis was completed with and without the modified race under 100% torque conditions. The TRBs were preloaded with 70 micron of negative clearance in order to accommodate mounting preload and preload due to thermal expansion of the bearing components. The cylindrical roller bearing HS SH A includes 18 micron of diametric clearance. In this opposing TRB configuration, bearing HS SH C will carry the entire thrust load generated at the mesh, adding to the preload of the bearing and the preload will be relieved from HS SH B. Figure 5.9 shows the magnified deformation of the high speed shaft stage with the modified raceway. Figure 5.10 and Figure 5.11 show
the contact pressure distribution on the HS SH B and HS SH C, respectively. It can be seen that all of the rollers in HS SH C are loaded; this is due to the fact that this bearing carries the thrust load. HS SH B however, has approximately 30% of the rollers out of contact. In each bearing there are little signs of edge loading.

Figure 5.9: Magnified deformation of the high speed shaft.
Figure 5.10: HS SH B roller contact pressure distribution.

Figure 5.11: HS SH C roller contact pressure distribution.
Simulations were completed with and without the modified bearing raceway. The strain levels in the outer race were then compared and the strain amplification provided by the groove geometry can be determined. Figure 5.12 and Figure 5.13 show the rollers for both HS SH B and HS SH C, respectively. The roller numbering convention is also shown in the figures. The first roller in each bearing is located at zero degrees or top dead center (TDC) of the bearing.

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**Figure 5.12: HS SH B TRB rollers with numbering convention.**
Figure 5.13: HS SH C TRB rollers and numbering convention.

The individual roller load for each TRB with and without the modified raceway is shown in Figure 5.14 and Figure 5.15. Once again the difference in load zone can be seen due to the thrust load generated at the high speed gear mesh. The groove, locations are circled, does have an impact on roller load as noted in the figures. The groove causes a decrease in the local stiffness of the raceway, thereby causing the roller in that region to carry less load which results in more load being carried by the other rollers. This does not impact the overall load share between the TRBs and is just a localized effect.
Figure 5.14: HS SH B roller loads for bearing with and without modified raceway, circled rollers are located under groove.
Figure 5.15: HS SH C roller loads for bearing with and without modified raceway, circled rollers are located under groove locations.
The hoop strain for the unmodified raceway was extracted around the circumference of the outer raceways. The results of this are shown in Figure 5.16 and Figure 5.17. The strain was sampled at two axial locations, 25% and 75% of the total length of the raceway. These results are from one position in time and are as if the race was unwrapped and laid flat on the page. Roller one is at TDC, which is at zero degrees. The peaks correspond to bearing locations, a peak indicates that a roller was directly under that location and a valley indicates that two rollers are straddling that point. Differences in load distribution along the rollers themselves results in differences between the two axial locations and will indicate whether there is edge loading or not.
Figure 5.16: Unmodified outer raceway hoop strain for HS SH B. These are for one time step and show the strain around the circumference of the raceway. Strains were sampled at two axial locations (25% and 75%).
Figure 5.17: Unmodified outer raceway hoop strain for HS SH C. These are for one time step and show the strain around the circumference of the raceway. Strains were sampled at two axial locations (25% and 75%).
The hoop strain for the modified raceway was extracted around the circumference of the outer raceways. The results of this are shown in Figure 5.18 and Figure 5.19. The strain was sampled at two axial locations, 25% and 75% of the total length of the raceway. These results are from one position in time and are as if the race was unwrapped and laid flat on the page. Roller one is at TDC which is zero degrees. The peaks correspond to bearing locations, a peak indicates that a roller was directly under that location and a valley indicates that two rollers are straddling that point. Differences in load distribution along the rollers themselves results in differences between the two axial locations and indicates whether there is edge loading or not. The groove greatly increases measurable strain in the raceway. Groove “a”, located at zero degrees increased strain by three times over the unmodified race for HS SH B and HS SH C. It was desired to get strain levels over 100 micro strain through the use of these grooves. The simulated results indicate that this has been achieved and the groove geometry has been successfully designed.
Figure 5.18: Modified outer raceway hoop strain for HS SH B. These are for one time step and show the strain around the circumference of the raceway at two axial locations (25% and 75%). Groove locations and labels are shown.
Figure 5.19: Modified outer raceway hoop strain for HS SH C. These are for one time step and show the strain around the circumference of the raceway at two axial locations (25% and 75%). Groove locations and labels are shown and strains from unmodified raceway are in parentheses.
Figure 5.20 and Figure 5.21 show the time trace for groove “a”. This is the strain in the groove as a roller passes under that location. This is indicative of what the actual measurement will be during testing. The peak to peak strain at the zero degree groove location increased from 30 micro strain to 150 micro strain for HS SH B and increased from 20 micro strain to 90 micro strain for HS SH C with the addition of the raceway groove. The stress field as the roller passes the groove location is also shown in Figure 5.20 where it can be clearly seen that the strain is at a minimum when two rollers straddle the groove and a maximum when there is a roller directly under the groove.
Figure 5.20: Hoop strain in groove "a" of HS SH B bearing. This is a time trace and shows the strain in the groove through time, as a roller passes over the groove.
Figure 5.21: Hoop strain in groove "a" of HS SH C bearing. This is a time trace and shows the strain in the groove through time, as a roller passes over the groove.

Peak to peak strain increased from 20 uStrain to 90 uStrain
5.3 Conclusion

The proposed groove geometry has been shown to provide strain amplification of as much as 300% over the unmodified raceway. This should provide sufficient strain levels for measurements and this geometry will be used in future experiments. The raceway does cause a local decrease in stiffness, which results in a decrease in roller load at groove locations. The groove will create a stress concentration in the bearing raceway; however, since the bearing is in a test gearbox and will only see limited cycles, failure should not be expected.
6.1 Thesis Summary

A full finite element model of the GRC test gearbox was modified and validated against experimental signals provided by NREL. The model performed very well in matching sensor data, which provided confidence in future results. Recommendations for future measurements were also made. The main focus of this study was in the planetary stage of the gearbox. The efficacy of the planetary configuration to equalize load sharing was evaluated and the effect of manufacturing errors on gear load distributions was also evaluated. Two spline modeling techniques were tested and various parameters were compared. The straight sided spline model was used for further simulations and the effect of the magnitude of spline lead crown on spline stiffness, sun motion, and planet load share was evaluated. Additionally, groove geometry, for the high speed bearings, was analyzed for the purposes of strain amplification.
### 6.2 Main Conclusions

- A static analysis proved sufficient for matching measured sensor data in the planetary stage. The carrier rim, bearing strain, and planet rim measurements were all accurately predicted by performing multiple static analyses as the components rotate.

- The carrier rim measurements will produce inaccurate predictions of carrier motion. There are not enough sensors in the gearbox to average out the deformation of the carrier rim. More sensors should be employed if an accurate measurement of the rigid body motion of the carrier is to be gathered.

- This particular floating sun configuration is able to mitigate manufacturing errors as large as 150 micron and perhaps larger (150 micron was the maximum pin error simulated).

- The planet load share is affected by the misalignment of the carrier and the floating sun is not able to fully mitigate this error.

- The straight sided spline model performed well when compared to the full involute spline model. The straight sided spline model saved computation time and was able to be integrated into the full gearbox model.
• The magnitude of lead crown causes a large change in spline stiffness and therefore impacts sun motion and planet load share. The effect of spline lead crown is primarily seen on the peak to peak planet load share.

• The lead crown magnitude also has a large impact on the maximum spline tooth contact pressure. An optimum design will be one that balances planet load share and spline tooth contact pressure.

• The proposed bearing race groove geometry proved to be sufficient for purposes of strain amplification and will be implemented in future testing.

6.3 Recommendations for Future Work

• Continue modeling of GRC gearbox and validate model with new high speed data.

• Build new model that will reflect changes made in the upcoming gearbox redesign. These will include tapered roller bearings at planet bearings and carrier bearings.

• Perform more detailed analysis of spline using the full involute model to develop a better understanding of contact pressures.
- Use the involute spline model to evaluate the effect of spacing errors on the spline stiffness and other parameters

- Analyze the intermediate shaft bearings as these have also had high failure rates. The spline loads will be transferred to these bearings so it would be interesting to see the effect of manufacturing errors on intermediate shaft bearing loads.
REFERENCES


