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NREL Gearbox Reliability Collaborative Experimental Data Overview and Analysis

Preprint

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Preface

With the growth of wind energy in the energy market, the design and implementation of larger wind turbines has become a common occurrence. Unfortunately wind turbines have been beset by a series of premature gearbox and drivetrain failures well in advance of the expected 20 year design life [1]. With larger wind turbines, the cost of gearbox rebuilds, as well as the down time associated with these failures, has become a significant portion of the cost of wind energy. In an effort to address and mitigate this problem, the National Renewable Energy Laboratory created the Gearbox Reliability Collaborative (GRC)[2]. The main approaches of the GRC are to combine dynamometer testing, field testing, and analysis to better understand the behavior and root cause of the premature gearbox failure. This paper presents a preliminary comparison between the dynamometer test data and analytical results.

Introduction

Most turbines in the market today follow a modular configuration comprised of a main shaft, gearbox, high speed shaft, and generator. The gearbox has the important task of increasing the slow rotor speeds to meet electromechanical requirements. These gearboxes are commonly composed of a planetary stage and several parallel shaft stages. The planetary, or epicyclical, design of the gearbox has many advantages compared to the traditional parallel shaft arrangement. Higher gear ratios can be achieved in a single stage, they are capable to carrying higher loads, and they require less space than the traditional parallel shaft arrangement. For this reason, planetary gearboxes are commonly used in the first stage of wind turbine gearboxes. However, planetary stages are more complex than the typical parallel shaft arrangement and can be affected by deflection in the planet carrier, annulus deformations and bearing clearances. Unanticipated levels of these motions can reduce their life expectancy. This paper gives a brief overview of a subset of the experimental efforts, data, and analysis of the GRC project focusing on the planet carrier deformation.

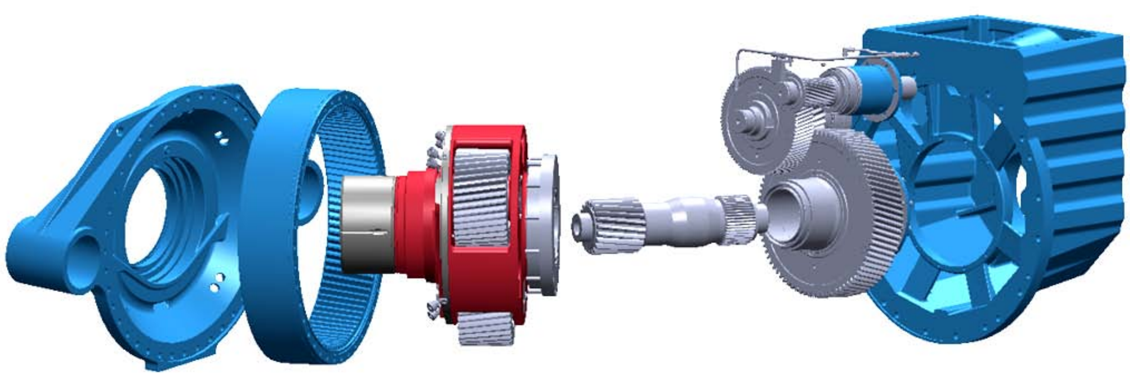

Figure 1: Gearbox Exploded view [2]

The GRC has heavily instrumented two gearboxes in an attempt to better understand the behavior and interaction among the different internal components of wind turbine gearboxes. The instrumentation focused on the planetary stage in an effort to understand its complexity and behavior. The instrumentation included a series of proximity sensors that recorded specially designed targets to better understand motion and misalignments of the different components of the planetary stage. The planet bearings were modified and instrumented to capture and validate the bearing roller load distribution and the ring gear teeth roots were instrumented to capture the tooth load distribution.

To truly understand the data, several post processing techniques and analytical models were implemented. The post-processing included filtering of the data in order to reduce noise and focus on the areas of interest. The modeling included detailed Finite Element Analysis (FEA) models as well as gearing and bearing analysis. The correlation between the experimental data and the existing models allowed better understanding of the data resulting in the generation of validated modes that could be used to better understand the gearbox behavior and estimate bearing life. This document offers an initial glimpse of a subset of the collected data and correlates it to analytical models.

Planet Carrier Deformation

The planet carrier has the important structural task of maintaining the annulus-sun-planets' center distances and alignments as well as transferring the input loads to the different components of the planetary stage. Due to the high and varying loads and the light weight requirements of wind turbine gearboxes, the planet carrier is prone to deflections that could affect the performance and life of the planetary stage. Figure 2 shows an example of the deformed planet carrier. The planet carrier is composed of two parallel circular plates that are connected by structural stiffeners. The purpose of these stiffeners is to increase the rigidity of the planet carrier reducing overall torsional wind up that can result in planet misalignment. The planets are located between these stiffeners and the space required for the planets make it a more compliant region of the planet carrier. The compliance seen in this area is the result of the space required to accommodate the planet as well as the bore that houses the planet pin. Figure 3 shows the planet carrier as well as the area required to accommodate the planet.

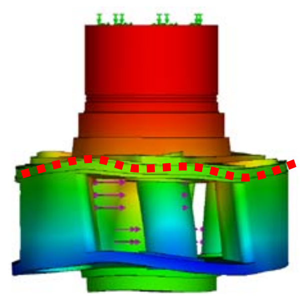

Figure 2: Deformed planet carrier

Experimental Data collection

The deformation is measured by proximity sensors at the upwind edge of the planet carrier. This edge has been machined in order to provide an consistent target for the proximity sensors producing a clean and reliable signal [3]. Figure 2 shows the deformed planet carrier and a dotted line indicates the deformed shape of the target surface. Figure 3 shows the planet carrier and the aforementioned machined surfaces.

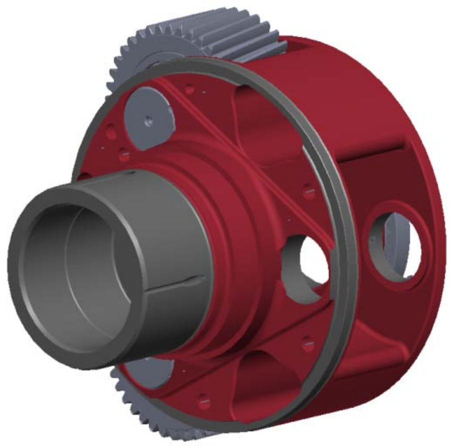

Figure 3: Planet carrier (only two planets are shown)

Data was collected through four proximity sensors that were located on the upwind section of the housing. Sensor locations can be seen in Figure 4. They were intended to capture the rigid body motion of the planet carrier. However, due to the high accuracy of the sensors, the deformation of the planet carrier was captured. It is important to mention that the proximity sensors were located on the housing, therefore, housing deflections could have had an impact on the sensor readings.

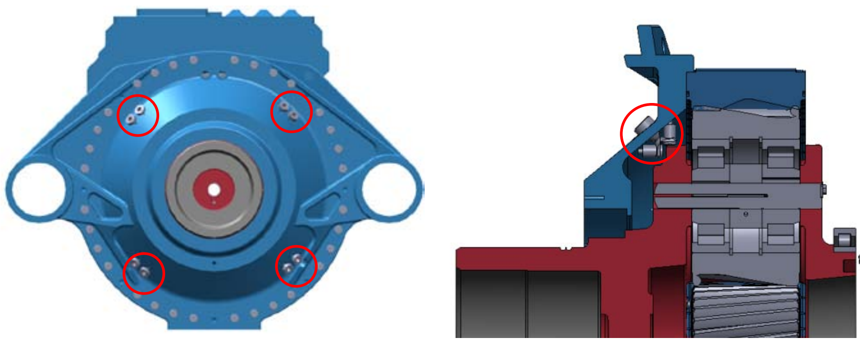

Figure 4: Proximity sensor location and internal side view

Model Description

An FEA analysis was performed on the planet carrier to better understand the deformation and misalignment generated under load. The model was constrained at the input shaft and forces were applied at the planet bearing location of the planet pins in order to simulate the loading conditions the planet carrier experiences under normal operation. The load applied to the pins is equally distributed between both planet bearings. The assumption is that the misalignment of the planet pins does not affect the gear mesh contact patterns and inherently change the load distribution of the planet bearings. The interaction between the planet pins and the planet carrier was modeled under various boundary conditions ranging from a fully rigid connection to the use of contact elements.

The misalignment resulting from the clearance between the planet carrier and the pins was ignored since the focus of this study was to quantify the contribution from the planet carrier deformation.

The coordinate system follows the FAST (Fatigue Aerodynamics Structures and Turbulence design code) [6] convention in which the X axis points along the axis of rotation and is oriented in the positive direction of rotation. The Y axis points to the left when looking in the positive direction of the X axis. The Z axis points upward and is perpendicular to the plane depicted on the X-Y Axis. Figure 5 shows the location of nodes used in the analysis and their deformation in the X direction.

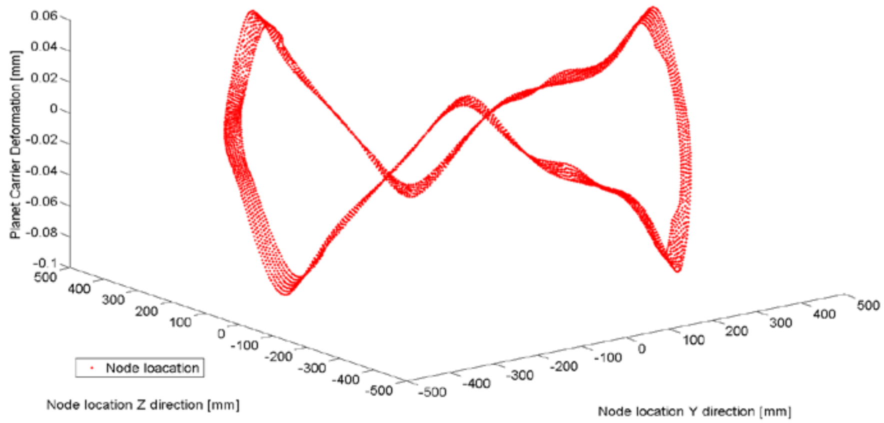

Figure 5: Planet carrier deformation

Analysis Assumptions and Considerations

It is important to consider some of the assumptions included in this work. There is equal load distribution among the planets. The load share between the planet ring and planet sun meshes are assumed to be equal. The planet body is rigid and it does not deform under load. The change in bearing load distribution does not affect the deformation of the planet carrier. The thrust or moment generated by the helix angle does not significantly change the load distribution of the bearings. The designer of the gearbox calculated an estimated design life of twenty years with even load distribution on the planet bearings.

Model & Experimental Data Correlation

It was determined that the simulated results would be processed after the test to establish a good correlation with the experimental data. The simulated axial deflection of planet carrier target swath (18mm wide) is shown in Figure 5. It can be seen that in addition to the general deformation in the x direction, the deflection of the machined surface increases radially outwards from the axis of rotation. The proximity sensors implemented in the test provide an average for the target area. To correlate the experimental and simulated data, an azimuthal average of the FEA results was used. The experimental data was recorded over several revolutions, under constant torque, and at several different load levels to ensure that the true deformation of the planet carrier was captured. The aforementioned experimental data was plotted with respect to the azimuth providing a good means of comparison with the calculated data.

Model Validation

As mentioned in the introduction, the GRC project instrumented two identical gearboxes. The results in this section are a comparison between the experimental measurements and the predictions of the analytical models to validate the models and the assumed boundary conditions within the models. The gearboxes have been arbitrarily named gearbox1 and 2 (GB1 and GB2). The gearboxes are identical except for tolerances and small differences that result from the manufacturing process.

The variation in stiffness described in the planet carrier deformation section results in a characteristic deformation that produces a three per revolution signature in the experimental signal. This type of deformation was observed in GB1.

FEA models were developed to better understand the deformation of the planet carrier. One model implemented contact elements for the boundary conditions that resulted in a model with compliance that is representative of the real system. The compliant joint allows for the planet pin to have more freedom with respect to the planet carrier surface. This results in a deformation that is mainly characterized by the previously mentioned variation in stiffness.

Figure 6 shows the data from GB1 collected at 75% of the rated power overlaid on the results from the analytical model under equivalent loading conditions. It can be seen that there is a good correlation between simulated and experimental data in shape as well as magnitude of the deformation.

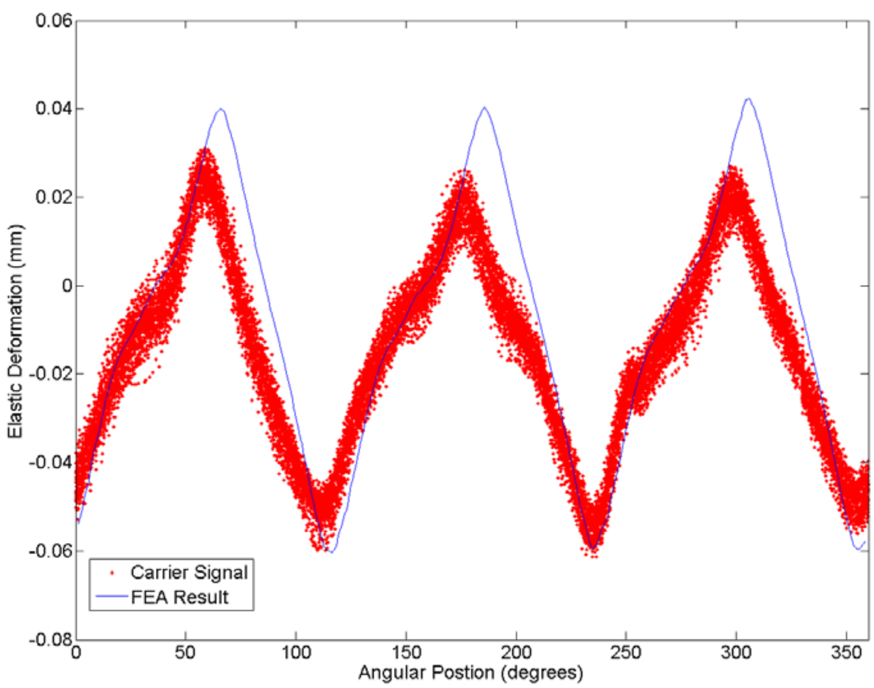

Figure 6: Planet carrier deformation GB1 experimental & simulated vs. azimuth at 75% rated power

The deformation recorded from GB2 testing showed a 6 per revolution (6p) characteristic shape. This type of behavior can be observed in the FEA when the planet pin interaction is stiffer or under a non compliant boundary condition. This behavior is associated with a press fit between the planet pins and the planet carrier.

Figure 7 shows the FEA results with a rigid, or welded, boundary condition. This boundary condition is not representative of the configuration present on the gearbox since it adds excessive rigidity to the system and can be seen as a reduction in the amplitude compared to GB1. However, this change in boundary condition clearly shows the 6p deformation signature.

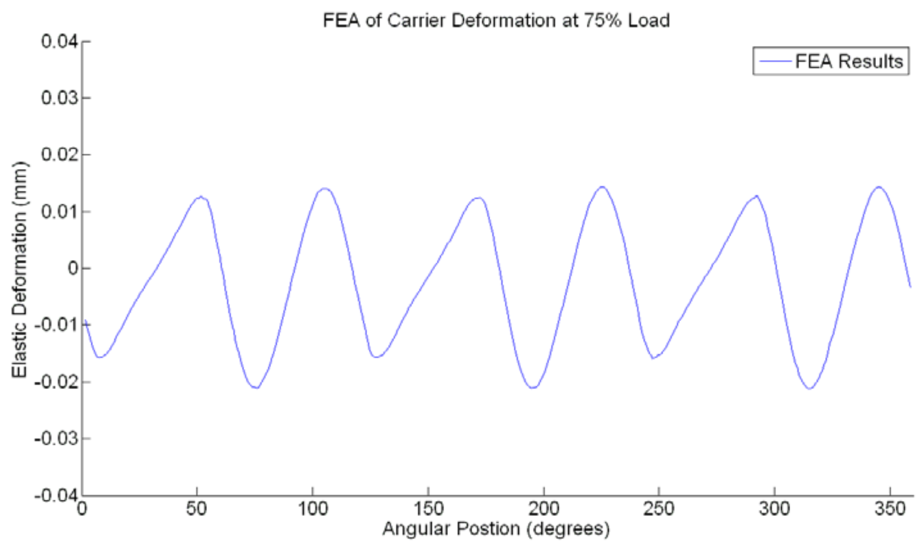

Figure 7: Rigid boundary condition deformation

Recognizing that tighter boundary conditions would change the behavior of the planet carrier, deformation towards the 6 per revolution deformation.The boundary conditions were adjusted to represent a press fit type of interaction to account for the added stiffness and the appropriate deformation magnitude.

Figure 8 shows the simulated results implementing the press fit boundary conditions and the experimental data for GB2 at the same applied torque level . Once again, it is apparent that there is good correlation between simulated and experimental data in shape as well as in magnitude of the deformation.

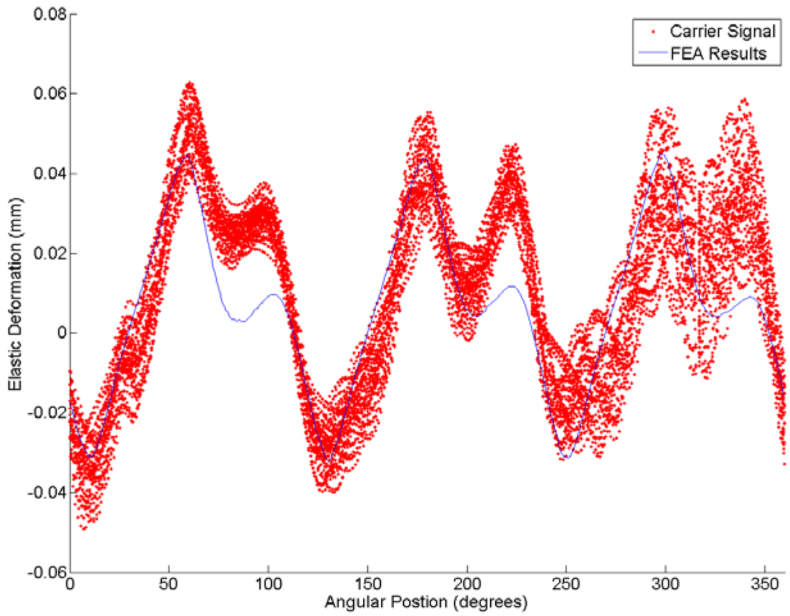

Figure 8: Planet carrier deformation GB2 experimental & simulated vs. azimuth at 75% rated power

Bearing Life Analysis

A problem present on wind turbine gearboxes is planet bearing failure [7]. This can be attributed to many factors such as low lubricant film thickness due to low speeds and high loads, poor roller load distribution due to planet deformation, misalignment, and poor bearing load distribution among others. For this reason, the exploration of bearing life reduction due to pour bearing load distribution resulting from planet carrier pin misalignment was worthy of further exploration.

Taking advantage of the validated models, a load of 100% rated torque was applied and the misalignment of the planet pin was calculated for the GB1 and GB2 models. The effect of the pin misalignment on the gear tooth load distribution throughout the mesh cycle was then simulated for both models [4]. Figure 9 shows the tooth load distribution for both ring gear to planet meshes. Note that the load distribution on GB1 shifted upwind due to the more compliant planet carrier (upwind is the origin of the active face width in Figure 9).

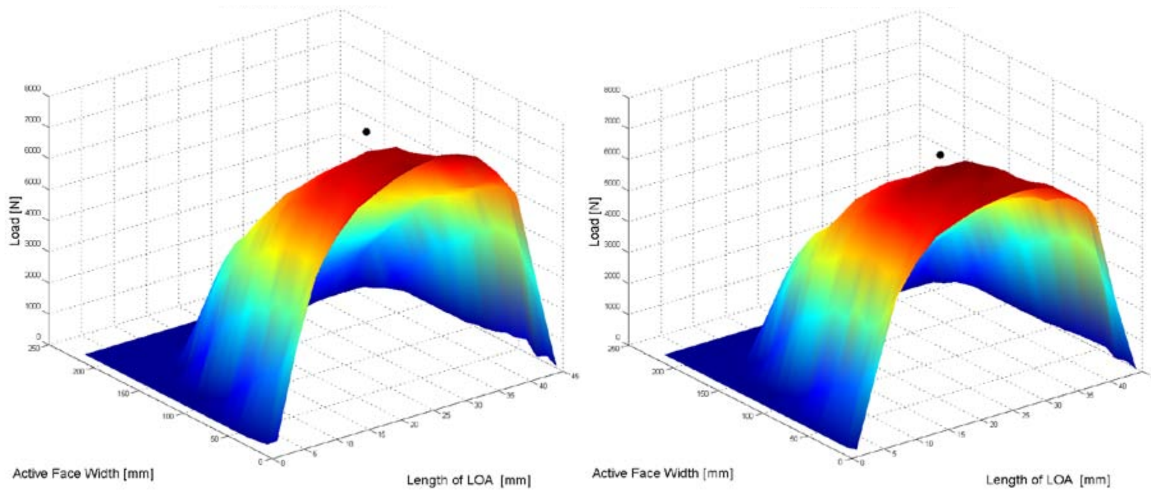

Figure 9: Planet ring tooth load distribution GB1 (left) GB2 (right)

The resultant load along the face width was calculated from the mean location of the load. Figure 10 shows the mean location of the load from a side view and the top view of the calculated mesh cycle. This procedure was performed for both meshes: planet=>ring and planet=>sun and the vectors were added to calculate the magnitude and location of the load that would be transmitted to the bearings.

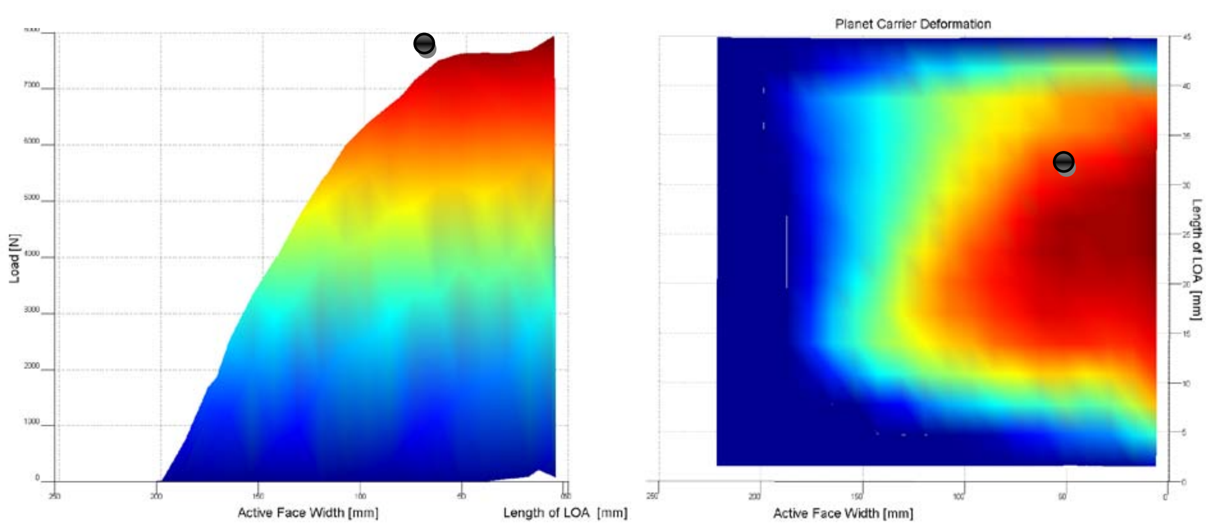

Figure 10: Gear tooth load distribution resultant load

The resultant load was used to calculate the bearing load distribution for both gearboxes. A sample plot of the change in load location and its impact on load distribution for GB1 (blue) and GB2 (red) can be seen in Figure 11.

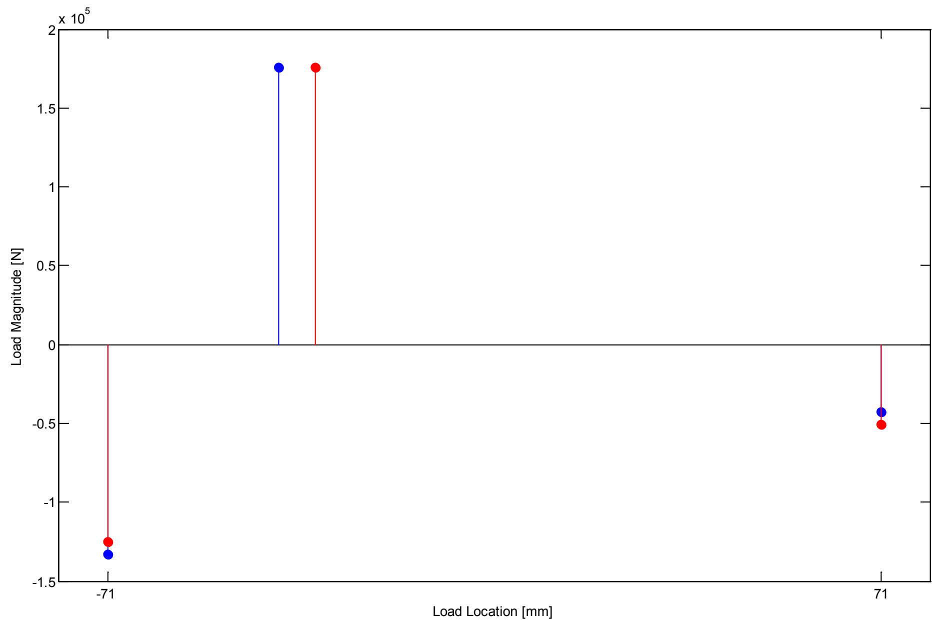

Figure 11: Gear resultant load location and bearing load distribution

Assuming the load was equally distributed between the two bearings within a planet, the gearbox should meet its expected design life of 20 years. Therefore, the variation in misalignment resulting from the planet carrier deformation can be calculated to predict its affect on bearing life. Using the Lundberg Palmgren fatigue life method [5], calculations showed a reduction of 8% in bearing life due to the predicted load distribution.

Conclusion

The fit between the planet pin and the carrier significantly influences carrier deformation. Therefore, FEA models must have the proper boundary conditions at the planet pin to carrier interface to obtain accurate predictions of carrier deformation.

Testing should be integrated into the analysis of wind turbine gearboxes to validate assumptions and enhance the results of analytical models.

Manufacturing variability can change pin-to-planet carrier interactions resulting in uneven bearing load distribution and consequently producing a significant bearing life reduction; in this case, there was a bearing life reduction of 8% out of the expected 20 year design life.

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