Validation of a Model of the NREL Gearbox Reliability Collaborative Wind Turbine Gearbox

Manufacturers, engineering consultants, and gear and bearing software providers are part of a program to model, build, simulate, and test gearboxes with a goal to improve reliability and reduce the cost of energy.

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The state with the largest installed capacity, Texas, can sometimes generate over 20% of demand from wind [New York Times]. Wind power has the lowest “life cycle emissions” of all energy production technologies, paying back the energy used in turbine manufacturer, installation, operation, maintenance, and decommissioning in three to six months [European Wind Energy Association].

Most wind turbines contain speed-increasing gearboxes to convert the slow main rotor speed into the 1000-rpm range for convenient generator operation. However, many wind turbine gearboxes have a poor reputation due to poor reliability. There have been many issues: grind temper, material inclusions, axial cracking in bearings, poor load sharing on shaft-bearing arrangements, significant gear misalignment leading to premature wear, bearing ring creep, gear scuffing, and gear or bearing micropitting. These are all common and often serial problems. There has been improvement in the last few years for some products as common failure modes have been addressed, yet it is not uncommon for wind sites built as recently as 2008 to have 20–40% of gearboxes requiring a component replacement (such as a high speed pinion or intermediate shaft bearing) by 2012 and 5–10% complete gearbox failures.

Some OEMs are manufacturing direct-drive wind turbines, where the generator rotor turns at the speed of the main rotor. This can lead to challenging structural engineering in maintaining a small generator air gap at a large diameter, inconvenient assembly and difficult transportation. Capital costs are high for direct drive and long-term reliability is unknown. However, some manufacturers accept these increased risks as the reduction in operating expenditure from removing the gearbox is perceived to be so great. Without an improved understanding of gearbox failure modes and more reliable gearbox products, this important market for the gear industry is in danger of being eroded by growth in the use of direct drive machines.

An important program for the industry, “The Gearbox Reliability Collaborative” (GRC), has been funded by the Department of Energy and run by the National Renewable Energy Laboratory (NREL) for several years. The collaborative has brought together OEMs, component suppliers, academia, national laboratories, engineering consultants, and software providers as part of a program to test gearboxes, validate software models, and verify design assumptions. The overall goal is to improve gearbox reliability and reduce the cost of energy produced by wind turbines.

The NREL GRC team has delivered an extensive instrumentation and testing effort on two 750kW gearboxes, described further below. The contribution of the authors to the project has been as members of the Analysis Team as well as designers of revisions to the next gearbox (being built in 2013). The focus of this paper is a comparison between measurement and simulation of key parameters; gear load distributions, annulus deflection, carrier deformation, and planetary gear load sharing. The simulation results that are robust and those that are sensitive to hard-to-predict parameters such as manufacturing and assembly variations will be outlined. Lessons learned in how best to apply computer-aided engineering tools to improve wind turbine gearbox reliability will be described.

TEST ARTICLES

Tests were carried out on two 750kW wind turbine gearboxes obtained by NREL. They were modified to have typical wind turbine bearing arrangements and using typical design practices. The aim of this original modification was not to solve all problems in the design—rather, it was to create gearboxes that were similar to the fleet of modern wind turbine gearboxes in operation. One of the gearboxes was tested in the NREL dynamometer—the other was operated in the field in a wind turbine.

The gearboxes are speed increasers, with the input shaft attached to the wind turbine rotor and the output...
The NREL measurement campaign was extensive with particular focus on measurement of system deflections and load distributions in planet-ring meshes and planet bearings. The gearboxes were instrumented with over 125 channels, for measurements including planetary tooth load distributions, ring gear hoop strains, planet bearing load distribution, sun gear orbit, and carrier deflection. They were then subjected to a rigorous testing regime, both up-tower and on the NREL 2.5MW dynamometer. The measurements used so far for the validation work are briefly described below, but for more details of these and all the other measurements, the full GRC report is from Phases 1 & 2 is publicly available through the NREL website (www.nrel.gov).

The ring gear strain was measured in the tooth roots at several positions across the face width and at three positions around the circumference to simultaneously capture the loading on each planet. Strain gauges were also placed externally on the ring gear. This allowed the hoop stress to be captured, but also strain gauges were placed across the face width, which allowed the tooth load distribution to be observed externally from the gearbox. Grooves were machined circumferentially in the planet bearing inner rings at various angles relative to the peak part of the load zone in order to capture the extent of the load zone, the load on the upwind bearing relative to the downwind, and the overall planet load sharing.

Proximity probes were attached to the housing at locations detecting the proximity of machined rings on the planet carrier in the radial and axial directions. This allowed detection of the deflection shape of the planet carrier.

**MEASUREMENT CAMPAIGN**

**GEARBOX SIMULATION**

The gearbox was modeled in the RomaxWind software package. The power of the system is that it analyzes all components (gears, bearings, housings, carrier, spline, bushings, main frame) in parallel, including system nonlinearities. The model is solved with an iterative method to calculate static equilibrium. This enables accurate prediction of system deflections and misalignments, which can then be fed into component life calculations using AGMA or ISO methods, included in the same software.

The gearbox mounting arrangement in the wind turbine nacelle is the so-called “three-point-mounted.” The main shaft that connects the main wind turbine rotor to the gearbox is only supported by a single spherical roller bearing, and the gearbox is supported by two elastomer bushings mounted to the main frame. In this arrangement, over-turning and yawing moments from main rotor weight and aerodynamic loading are transmitted to the gearbox, as the spherical roller bearing has no reaction to an applied moment. Therefore the model includes a representation of the main frame, main bearing, and main shaft as shown in Figure 2.

2. Loads are applied upwind of the main bearing flange.

The shafts, housing, torque arm, main frame, ring gear, and planet carrier are modeled with finite elements. Shafts are modeled as Timoshenko beam elements, whilst all other parts are solid meshed and then reduced to super-elements using static condensation.

Gear meshes are modeled with tooth contact analysis. The face width is broken down into to multiple finite-width thin strips. The bending stiffness of each strip is calculated using a finite element mesh based on the tooth and root geometry. Local deformations are calculated using Hertzian contact theory. The number of strips in contact depends on the load, misalignment, tooth micro-geometry modifications, and twist along the face width (important for the sun gear). The approach is similar to the many tooth analysis codes on the market, but here the tooth contact is solved in parallel to the full system deflections. The position of the resultant load on the face width changes the loads on the bearings.
and hence the misalignment of the gears, so iterations are required to achieve the force and moment balance.

The gearbox spline is modeled in a similar way to gears, by breaking down into strips and calculating the deflection of each strip including the effect of crowning and a finite element mesh to get the tooth stiffness.

Load sharing between teeth in a spline is critically affected by pitch errors between the teeth, and typically it is assumed that only half the teeth are carrying load. The effect of pitch errors was not accurately captured in the model, as the calculated spline stiffness was varied to check the sensitivity of the results. Variations in spline stiffness did not change the results of interest in this paper significantly, so focus was not put on improving the model in this area.
The roller bearings are modeled based on a similar approach to gears, with a combined numerical and analytical approach. Bearing manufacturers typically do not share bearing internal details, so roller size, number of rollers, roller profile modification, and so on are estimated based on the published load capacity and past experience. This gives a reasonable estimate of bearing stiffness. The operating radial and/or axial internal clearances are calculated based on the bearing clearance as supplied (i.e., CN, C1, C2 etc.), the shaft and housing fits, and thermal expansions based on estimated temperature differences across each bearing, as suggested by ANSI/AGMA/AWEA 6006-A03. Mesh misalignments predicted by the model were shown to be sensitive to bearing clearance, so this is an area of the model in which it is important to get a good prediction.

The model may be used for investigations of the sensitivity to manufacturing tolerances and other variations. All mechanical dimensions will have a tolerance and there are a multitude of unknowns (for instance, the exact operating temperatures and therefore the operating bearing clearance). Although “virtual” investigations into the effect of these variables are straightforward, the investigator may be overwhelmed with the number of possibilities and studies that can be carried out. The validation exercise guides...
these studies and highlights where tolerances or unknowns are particularly important.

**MODEL VALIDATION**

**Ring Gear Load Distribution**

The measurements of ring-planet load distribution showed clearly that there was a different load distribution at each of the azimuth positions. This suggests a misalignment between the planet carrier and the ring gear. The model (with no manufacturing tolerances or differences between each planet) showed a good match at two of the positions, and a reasonable match at one position (see Figure 3). This occurs in the model only when the correct off-axis load due to the applied weight was applied to the main shaft. The effect of this weight is to deflect the carrier relative to annulus such that the gear misalignment varies with carrier azimuth.

Improved validation was achieved by modifying the connection dimensions between the mesh load point and the ring gear solid finite-element mesh. The annulus is reduced to a stiffness matrix (super element) with nodes at the gear mesh points. The node at the gear mesh point is connected to the ring gear finite elements by RBE2 (fully rigid beam) connections. Judgment is required when making these connections as to how many finite elements to connect and their location. The best result was achieved by connecting the gear mesh node to finite elements all across the face width of the annulus in a narrow band (see Figure 4). Although this has a stiffening effect on the ring gear stiffness matrix, the stiffening is counteracted by the fact that the local deflection is captured in the gear mesh model that is used to derive the force and moment applied to the gear mesh node. The results were also somewhat sensitive to the number of finite elements used across the ring gear so this number was increased until the results converged.

**Planet Carrier Deformation**

Proximity probes measured the position of a machined surface on the carrier relative to the housing, capturing the deformed shape of the carrier under load. The model predicts the carrier deforms quite...
significantly as can be seen in Figure 5. However, the measurement included significant amplitude of variation at the frequency of once per revolution of the carrier, as shown in Figure 6. Given the planet carrier is nominally symmetric to each planet, this must be due to a manufacturing or assembly variation. One item that would generate such a variation was error in the perpendicularity of the machined measurement surface relative to the axis of rotation. The magnitude of this error that would give amplitude of variation in the model that was similar to the measured data was well within the tolerance on the drawing. Alternatively the carrier-to-main-shaft connection may have had some fundamental assembled misalignment, but this was considered to be less likely. A large error of this type would cause an irregularity in the loading between each planet that was not observed.

Components of the measurement at the planet passing frequency and its harmonics were captured by the model. These are the deformation of the carrier due to the gear loads. A comparison between measured and simulated data is shown in Figure 7. A sine wave with the amplitude of the component of the measured data at rotational frequency was added to the simulated data. In future tests it is recommended to measure the proximity as the carrier rotates at zero load, or measure the carrier with a coordinate-measuring machine (CMM) to determine the source of this error at rotational frequency. The fact that all the proximity changes due to deflection were accurately represented in the model verifies the modeling of the planet carrier and the connection between the carrier and the planet pins.

The major design concerns other than the fundamental strength of the carrier itself are in the operation of the gears and bearings. The good match between the deformation measurements at the planet passing frequency and the model give good confidence in the methods used for meshing the planet carrier and how the loads were applied. If the component at the rotational frequency of the carrier is indeed due to the tolerance in the machining of the measurement surface, this does not have an impact on the design process.

**Ring Gear Strain Measurements**

The strain gauges external to the ring gear measured a rapid increase and decrease in hoop strain, followed by a smoother increase and decrease, as each planet passed. The model showed a similar characteristic, shown in Figure 8. However, there was a discrepancy between the magnitude of the “spike” predicted by the model and the measured amplitude. Reasons for this are proposed to be the fidelity of the finite element mesh on the external of the ring gear, the reflecting of the length measured by the strain gauge in the predicted results (the predicted results just show the strain in a single finite element), and the accuracy of the measurement itself in capturing the very small strain. Extensive effort was not applied in capturing this magnitude, as it is not a design-critical parameter.

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**Planet Gear and Bearing Load Share**

Data obtained from the strain gauges placed on the planet gear bearing inner rings were post-processed to obtain a measure of load share on each bearing and on each of the planet positions. The results in Figure 9 show load sharing factors for the upwind bearings, downwind bearings, and overall load. The load sharing factor $K_\gamma$ (using the typical gear rating symbol) is presented here as the multiplier to be applied to the average load between the three planets (so, for example, a $K_\gamma$ of 1.1 for Planet A indicates 110% of the average load over all three planets is transmitted by Planet A). Illustrated in Figure 9, the variation in load between the upwind bearings is small, but for the downwind bearings much larger. These graphs do not illustrate the fact that the overall load on the upwind bearings is much greater than that on the downwind bearings. Load sharing predicted by the model is slightly better than that measured. There was a known variation in clearance between each planet, and including this in the model would improve this correlation. For instance, in the measured data it can be seen that Planet A on average transmits more load than the Planets B and C.

The guidelines for wind turbine design suggest the load sharing factor $K_\gamma$ for gear design can be 1.0 for three planet designs (i.e., it can be assumed that the load sharing is perfect). The measured data showed that the peak load on the planet was approximately 1.12 times the average. Also, the bearing loads varied much more widely. This was captured by the model without considering manufacturing tolerances, so is caused by deflections and wind up of the gearbox components. The gearbox designer should ensure that the bearing selection takes this into account.

**CONCLUSIONS**

The model captured the measured behavior of the wind turbine gearbox. To achieve good correlation, several lessons were learned in use of the model. In particular, it was found important to:

- Include enough degrees of freedom in the analysis of structural parts (in particular elements across the ring gear face width),
- Carefully consider bearing clearance and stiffness effects, and
- Include the effects of non-torque loading and to include the main bearing and main shaft in the model.

The measurements taken by the GRC also feedback useful information into the design process, in particular:

- The planet-ring load distributions varied with azimuth positions around the carrier, due to misalignment between the carrier and the ring gear. This is important, as this cannot easily be corrected with a helix angle modification on the planet gears—they operate with differing misalignment as the carrier rotates. Extreme off-axis load conditions will increase the misalignments. This should be considered at the design stage to ensure that edge loading is not introduced.
• The planet-bearing strain measurements showed that there was significantly more load carried on the upwind bearings compared to the downwind. This was due to the combined loading and deflection conditions and captured by the model. This may be overlooked at the design stage if the combined system effects of carrier wind-up, gear forces, and bearing clearances and stiffness are not considered together.

• The three-point mounted configuration (one main bearing and two bushings on torque arms) is sensitive to off-axis loading. Whilst other configurations may be more isolated from off-axis loading, such as four-point mounting with two separated main bearings, off-axis loading should still be considered. Some manufacturers are moving towards a more integrated, less modular arrangement, where the gearbox and generator are integral to (or bolted to) the mainframe of the wind turbine nacelle. In this case it will be very important to understand the deflection of the main shaft, main bearings and main frame itself in order to determine the misalignment at the gear and bearing contacts.

The successful validation of the model has demonstrated the usefulness of a modeling tool for capturing system deflections, misalignments and load distributions. These are key inputs that contribute towards the reliability of the gearbox, but this is only one aspect of the design process. There would be limited benefit to having a gearbox that maintains good internal alignment under all load conditions, but cannot be easily assembled, is constructed from poor quality material, is not sealed carefully, is not properly maintained, or is not lubricated properly. Romax Technology is working with NREL on a redesign of the gearbox for Phase III of the GRC project, where a further measurement campaign will be undertaken and an assessment of the gearbox improvements will be reported to the industry.

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