Design Evaluation of Wind Turbine Spline Couplings Using an Analytical Model

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Contents

1 Abstract

Articulated splines are commonly used in the planetary stage of wind turbine gearboxes for transmitting the driving torque and improving load sharing. Direct measurement of spline loads and performance is extremely challenging because of limited accessibility. This paper presents an analytical model for the analysis of articulated spline coupling designs. For a given torque and shaft misalignment, this analytical model quickly yields insights into relationships between the spline design parameters and resulting loads; bending, contact, and shear stresses; and safety factors considering various heat treatment methods. Comparisons of this analytical model against previously published computational approaches are also presented.

2 Introduction

Gearboxes in wind turbines do not always achieve their expected design life [SS13], even though they often meet or exceed the current design criteria and standards in the gear, bearing, and wind turbine industries, as well as third-party certification criteria. The National Renewable Energy Laboratory (NREL) Gearbox Reliability Collaborative (GRC) was established by the U.S. Department of Energy in 2006; its key objective is to understand the root causes of premature gearbox failures and improve gearbox reliability using a combined approach of dynamometer testing, field testing, and modeling [HL11]. Goals of the GRC include facilitating an increase in the accuracy of existing gearbox design and modeling tools, producing these tools if none are available, and making recommendations to improve gearbox design standards.

As part of the GRC program, the work presented in this paper investigates the design of spline couplings, which are often used in modern wind turbine gearboxes to connect the planetary and helical gear stages. In addition to transmitting the driving torque, articulated spline couplings are also used to improve load sharing in the planetary stage by allowing the sun to “float”. A freely floating sun minimizes the negative effects of imperfections, misalignments, and nontorque loads on planetary gear mesh contact patterns. Conversely, without the floating sun, gearbox misalignment and unequally shared loads can occur. As a result, edge loading of the gears and planet-bearing forces increase, leading to reduced gear and bearing life, and increasing the potential for premature failure [YG10]. The amount the sun can float is determined by the spline design and the sun shaft flexibility, subject to the operational loads. A few standards address spline coupling design requirements in varying detail, with the most detailed guidance provided in the American Gear Manufacturers Association’s (AGMA’s) 6123-B06 design manual for single articulation couplings.

This paper presents calculations of spline coupling operational loads and stresses, plus contact, bending, and shear safety factors based on fatigue and yield, for a test gearbox using a reduced-order analytical formulation and high-fidelity finite-element (FE) modeling tools [YG13]. This paper also contributes a new comparison to another model and an examination of the effect of spline design parameters on spline behavior and performance.

3 Test Article and Instrumentation

The test article was originally designed for a two-speed, stall-controlled, three-bladed upwind turbine with a rated power of 750 kilowatts (kW). The gearbox is composed of one low-speed planetary stage with three planet gears and two parallel shaft stages as shown in Image 1. The gears and bearings were redesigned and modified from the original configuration used in the commercial versions of this wind turbine. This redesigned gearbox is hereafter termed "the GRC gearbox."

The radial position of the sun gear relative to the planet carrier is measured using two orthogonal proximity sensors mounted on the planet carrier. These discern the motion of the end of the sun shaft beyond the end of the sun pinion (also shown in Image 1).

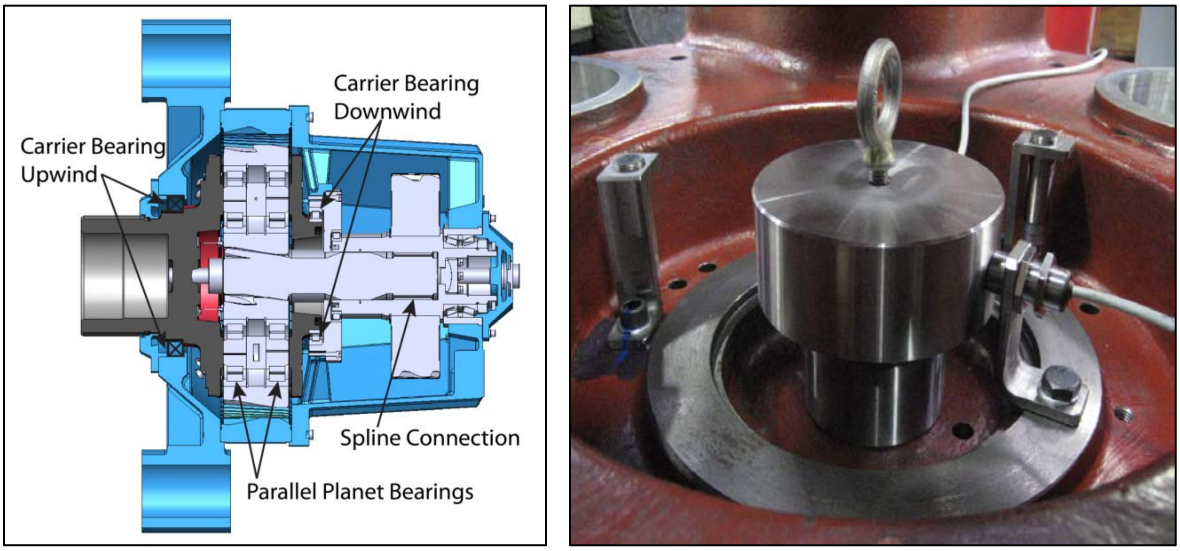

Image 1: A cutaway view of the GRC gearbox configuration (left) and one of two sun proximity sensors (right). *Photo by Edward Overly, NREL 26666.*

4 Modeling Approaches

Four modeling approaches of varying fidelity were explored in this work: 1) an analytical model [YG13], 2) a hybrid, two-dimensional (2D) FE and analytical model, 3) a fully three-dimensional (3D) FE model, and 4) a semianalytical contact analysis model. The hybrid 2D FE model was RomaxWind 14.5.0, the 3D FE model was Calyx, Transmission3D version 2.2700|0.1195, and the semianalytical model was SplineLDP version1.0.0. The FE models of the drivetrain have previously been validated against relevant GRC experimental data for bearing and gear loading [HL11, YG12, JA13, AC11]. Because of the current lack of experimental data, results of the analytical model formulation were compared to the outputs from these higher fidelity models.

The analytical formulation described herein provides much of the same information as modern gearbox design software, plus estimation of spline safety factors. A full list of assumptions of the formulation are detailed in [YG13], including equal tooth spacing and zero pitch error. By its nature, the analytical formulation provides greater insight into the effect of the spline coupling design parameters upon the spline performance and resulting safety factors than the other two approaches. Solutions can be calculated two orders of magnitude faster than higher fidelity models, making it very useful for early design stage parametric studies.

5 Results and Discussion

When an ideally manufactured spline and sleeve are perfectly aligned, all of the teeth are in contact and the torque is transmitted evenly. The loads are centered at the tooth midpoint and the maximum load on a single tooth equals the average load per tooth; however, when the sun shaft is misaligned, the number of teeth in contact is reduced, the loads migrate toward the edges, and the maximum tooth load increases. These effects are explored in the following sections.

5.1 Effect of Shaft Misalignment

Image 2 shows the tooth load distribution at rated torque and selected misalignment angles including no misalignment, the maximum operational misalignment (0.03°), half the jam angle (0.1°) and just before the spline jams (0.19°). When perfectly aligned, the teeth shared loads equally, the tooth loads were centered at midtooth, and the load profile was parabolic. For small misalignment angles, the center of the contact area deviated from the tooth's geometric center in a sinusoidal pattern around the circumference of the spline. The further the center of the contact was from the tooth midpoint (center-to-center distance), the larger the tooth load was. As the misalignment angle grew, the center-to-center distance increased and so did the maximum tooth load. As misalignment approached half the jam angle, some teeth became entirely unloaded. The remaining teeth in contact carried the load and their contact area migrated even closer to the tooth edge. At the jam angle, only about one-third of the teeth were carrying any load at all. In this situation, the maximum tooth load was approximately quadruple the nominal load, and the load was very close to the edge of the teeth. These edge-loaded teeth are at risk for pitting.

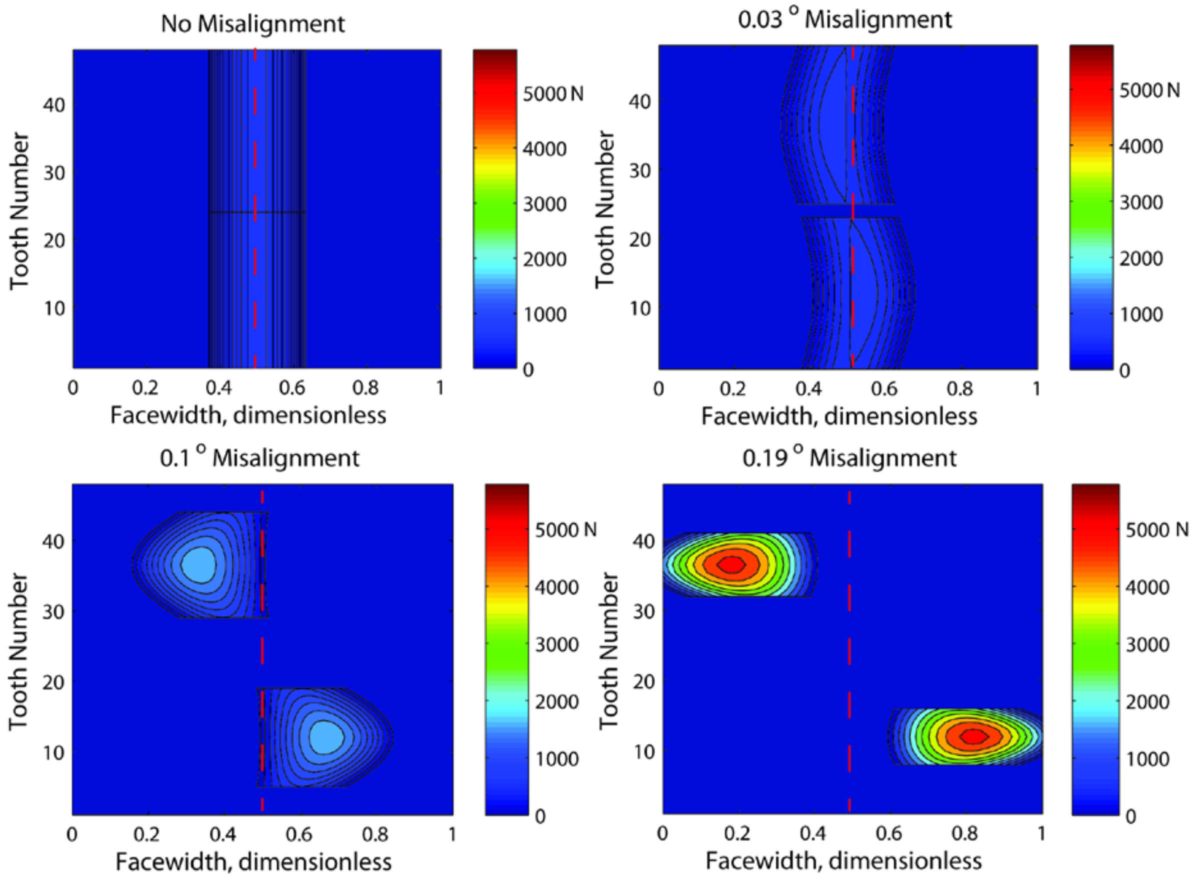

Image 2: Spline tooth load distributions calculated by the analytical model

5.2 Effect of Transmitted Torque

The measured torque spectrum from field testing the GRC gearbox in one turbine ranged from negative torque to two times the rated torque. The effect of torque on the maximum tooth load is shown in Image 3. In the figure, the spline shaft was misaligned by 0.1° (half the jam angle). Deviations existed among the modeling results, and were largely caused by the differences between the modeling approaches. For instance, the Transmission3D model considered the spline tooth and rim elasticities in the three-dimensional space. The RomaxWind model considered gear tooth/rim bending and shaft flexibility, but excluded the buttressing effects between tooth slices. The Gear-ScouP model only considered gear tooth bending. These different approaches of addressing elasticity were likely the main contributors to the differences in results. Experimental validation is clearly needed and will be crucial to accurately characterizing the spline loads and safety factors.

The results of high-fidelity models showed similar trends with varying torque. With a rigid shaft model, Transmission3D predicted the highest loads of all. Of all the other models, SplineLDP gave the lowest single tooth load. The analytical model predicted lower maximum tooth loads than the FE models. This might have been because of the simplification of the contact stiffness calculation of the proposed model. The proposed approach did not include the influence of torque on contact stiffness, whereas the RomaxWind and Transmission3D models did. A safety factor of 1.5 is recommended when using the analytical model for applications where the torque is less than 75% rated or greater than 125% rated. When torque increased from 50% to 150% of rated, the maximum tooth load for the analytical model approximately doubled. Therefore, spline loading and contact conditions were clearly torque-dependent.

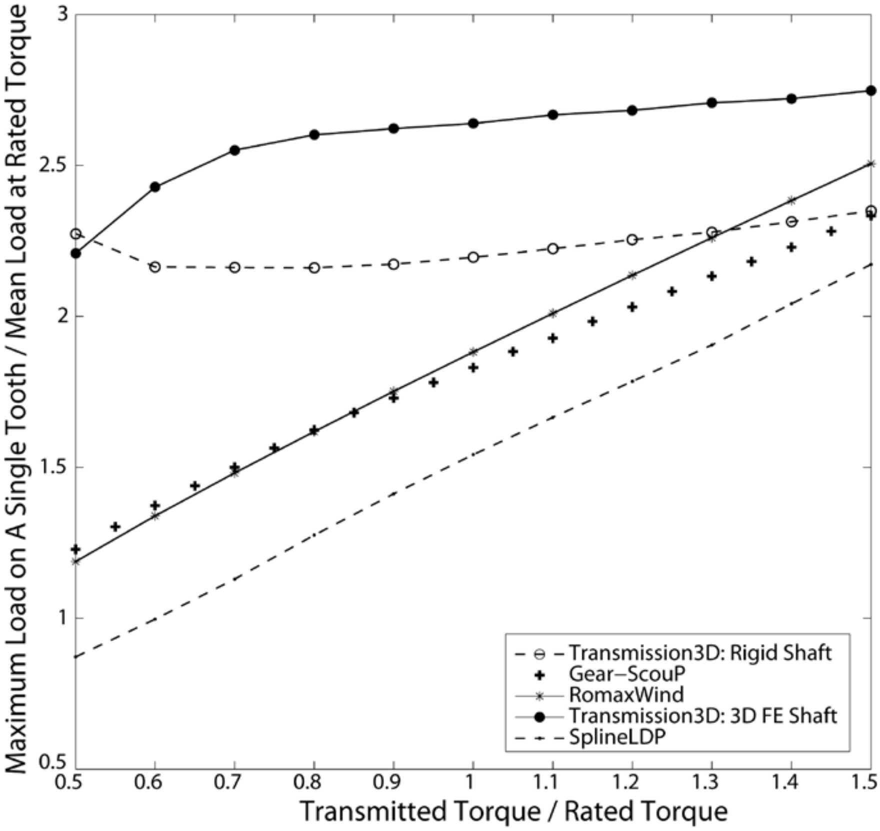

Image 3: Effect of torque on the maximum load of a single tooth

5.3 Effect of Spline Crowning

Crowning is a modification of the spline teeth for avoiding high edge loading. Using the analytical method, the effect of the lead crown radius on the number of teeth in contact and the maximum tooth load are examined in Image 4. Results were calculated in perfect alignment, at the maximum operational misalignment (0.03°), and at half the jam angle (0.1°) at rated torque. At zero misalignment, the load share was perfect and every tooth transmitted the same amount of torque. Crowning had no effect in this situation. With misalignment, reducing the crown radius (i.e., increasing crowning) decreased the maximum tooth load. For operating misalignment conditions, the design crown radius yielded a high load share factor and a maximum tooth load close to the nominal load without misalignment. In this situation, decreasing the crown radius would not have reduced the loads on the GRC sun spline and thus would not have been worthwhile.

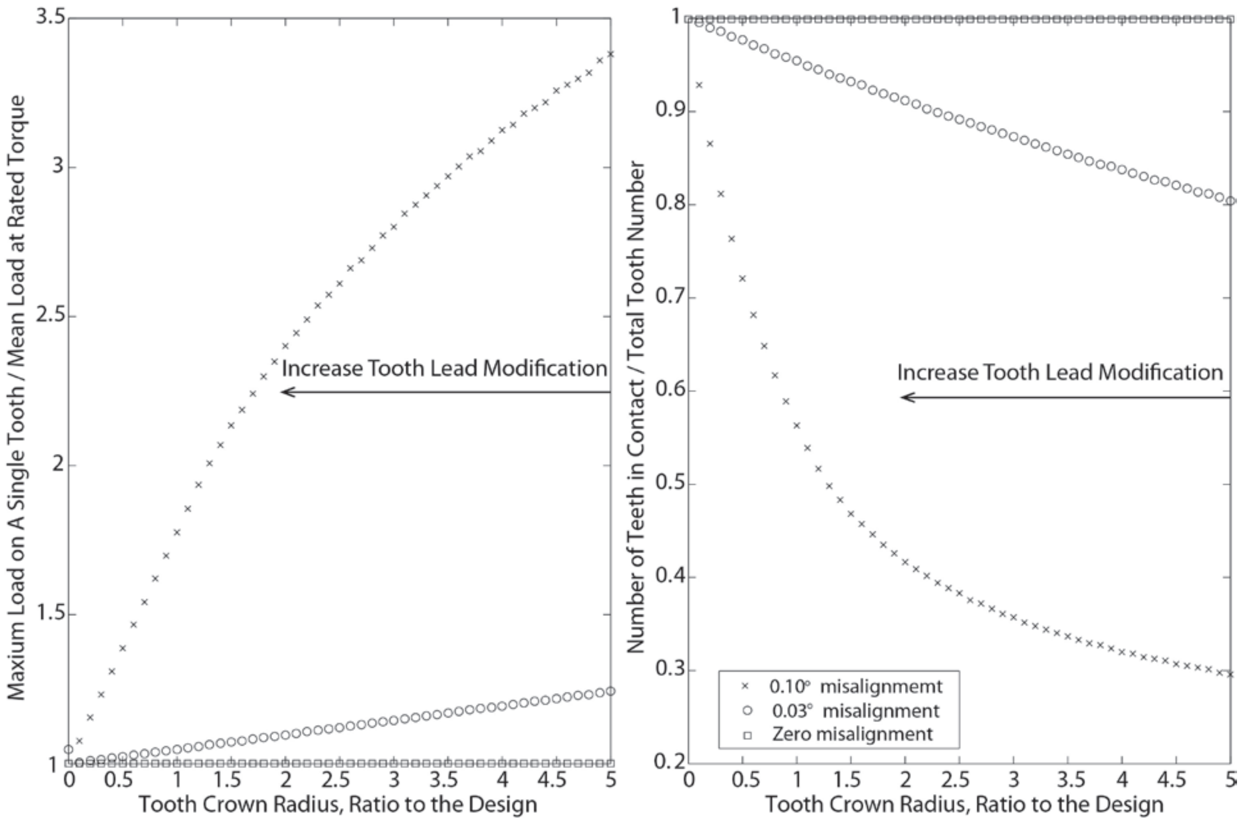

Image 4: Effect of crown radius on the spline tooth loads

5.4 Effect of Heat Treatment Methods

The effect of different heat treatment approaches on the spline fatigue and yielding safety factors are compared in Image 5. Among all heat treatment methods, induction hardening only the tooth flanks (method B) led to the lowest bending and shear safety factors. Through-hardening the teeth caused the lowest contact safety factor. Carburizing the teeth or induction hardening both the tooth flanks and roots (method A) were the best practices to enhance the strength of the spline teeth. AGMA 6006-A03 recommends case hardening of the spline teeth, preferably by nitriding, although this study shows that carburizing teeth or induction hardening both the tooth flanks and roots produced higher safety factors.

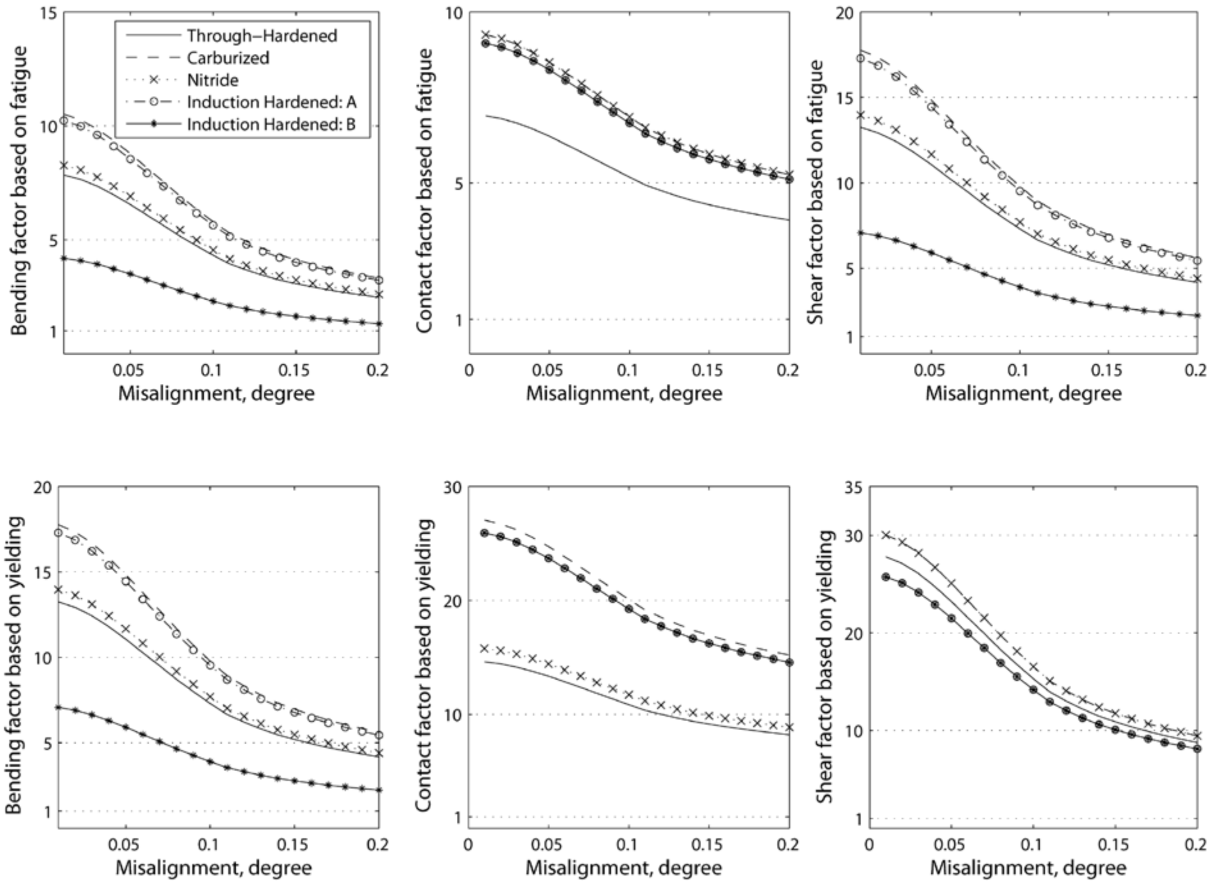

Image 5: Effect of heat treatment on fatigue safety factors

6 Experimental Validation of Numerical Models

There were significant differences among the studied models, particularly when modeling large misalignments. Therefore, it is crucial to perform experimental validation of these numerical models. NREL’s 5-megawatt (MW) dynamometer facility is currently conducting tests on a pair of utility-size couplings that connect the driving gearbox to the nontorque loading system as shown in Image 6. These couplings contribute to the nontorque loads applied to a test specimen during test operations. Validation using the NREL dynamometer is easier than using a gearbox with internal splines, because the splines are external and already instrumented for displacements and loads.

Instrumentation on the coupling shaft includes three sets of full-bridge bending gauges that measure shaft bending moments, two sets of proximity sensors that measure the shaft misalignment and sliding motion, and torque gauges. This instrumentation was installed to verify the coupling-induced loads (global behavior) over a wide range of operating misalignments and load conditions. The next phase of instrumentation will include coupling tooth root strain gauges to characterize the tooth contact (local behavior) of the couplings. These ongoing experiments will provide crucial information to validate the models and characterize coupling performance during operation.

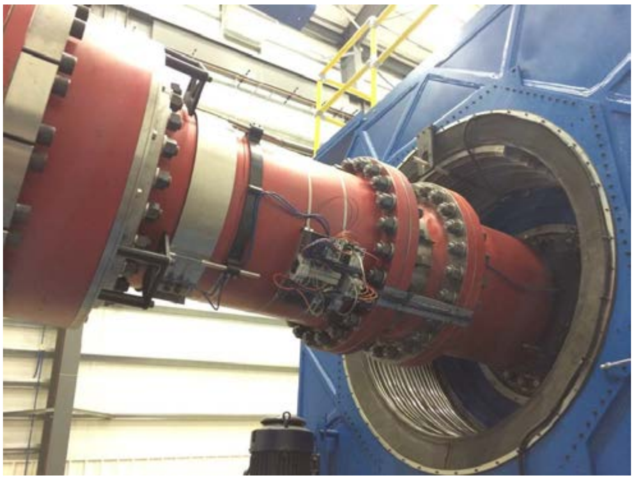

Image 6: Gear tooth coupling installed in NREL's 5-MW dynamometer facility. *Photo by Scott Lambert, Lambert Engineering 32611.*

7 Summary

The analytical formulation described in this report provides much of the same information as modern gearbox design software, plus estimation of spline safety factors. By its nature, the analytical formulation provides greater insight into the effect of the spline coupling design parameters upon the spline performance and resulting safety factors than the other approaches. Solutions can be calculated two orders of magnitude faster than higher fidelity models, making it very useful for early design stage parametric studies. It has been coded into MATLAB software and is publicly available as a stand-alone executable program called the Gear Spline Coupling Program (Gear SCouP).

When the spline is in perfect alignment, the load is shared equally across all spline teeth and the tooth load distribution has a parabolic shape. When the spline is misaligned, the number of teeth in contact decreases and the maximum tooth load increases sharply. Torque affects the spline load share, maximum tooth load, and safety factors. It is important to evaluate the spline design within the entire torque spectrum. Crowning improves the load share factor and maximum tooth load; however, tooth contact stresses can increase and the associated safety factors can decrease. Carburizing and induction hardening the tooth flanks and roots are the best practices to enhance the spline strength among those studied, though the risks of fretting corrosion and scuffing are not yet addressed in the analytical model. Experimental validation of Gear SCouP is underway at NREL's 5-MW dynamometer drivetrain test facility.

8 Acknowledgements

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