GRC1.5 Project: Joint Industry Megawatt Scale Gearbox Field Tests

Cooperative Research and Development Final Report

CRADA Number: CRD-17-00694

NREL Technical Contact: Jonathan Keller

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Contract No. DE-AC36-08GO28308

Technical Report
NREL/TP-5000-88867
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Cooperative Research and Development Final Report

Report Date: February 12, 2024

In accordance with requirements set forth in the terms of the CRADA agreement, this document is the CRADA final report, including a list of subject inventions, to be forwarded to the DOE Office of Scientific and Technical Information as part of the commitment to the public to demonstrate results of federally funded research.

Parties to the Agreement: Winergy Drive Systems Corporation, a wholly owned subsidiary of Siemens Corporation (also known as Flender Corporation)

CRADA Number: CRD-17-00694

CRADA Title: GRC1.5 Project: Joint Industry Megawatt Scale Gearbox Field Tests

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Sponsoring DOE Program Office(s):

Office of Energy Efficiency and Renewable Energy (EERE), Wind Energy Technologies Office

Joint Work Statement Funding Table showing DOE commitment:

Estimated Costs	NREL Shared Resources a/k/a Government In-Kind
Year 1	$400,000
Year 2	$400,000
Year 3	$400,000
Year 4, Modification #1	$0
Year 5, Modification #2	$0
Year 6, Modification #3	$0
TOTALS	$1,200,000

Executive Summary of CRADA Work:

A new DOE/NREL industry collaboration called the Gearbox Reliability Collaborative (GRC) 1.5 will undertake field testing on current commercial multi-megawatt wind turbine gearboxes to collect loading data from installed turbines to thoroughly characterize gearbox input loads and responses during actual in-field conditions. A chief outcome is to provide operational loading data relative to the most common failure modes. This will provide a greater understanding of steady-state, transient, and fault response for both the input and output of the gearbox, thus facilitating improvements in the gearbox, power converter or turbine controller.

CRADA benefit to DOE, Participant, and US Taxpayer:

- Assists laboratory in achieving programmatic scope
- Uses the laboratory's core competencies, and/or
- Enhances U.S. competitiveness by utilizing DOE developed intellectual property and/or capabilities

Summary of Research Results:

The NREL Gearbox Reliability Collaborative (GRC) was initiated by the U.S. Department of Energy and developed with broad participation from the wind turbine industry to address the fact that wind turbine gearboxes are not achieving the expected 20-year design life. The GRC approach is to document and review the complete design process–from design to loads measurement to design validation – and identify the areas critical to design success that have high sensitivities or uncertainties. To date, not enough data has been collected from installed turbines to thoroughly characterize gearbox loads and responses related to premature failures of gearbox bearings during actual in-field conditions. The primary objective of this CRADA project was to measure the operational conditions on the high-speed-shaft bearings in an installed wind turbine gearbox to characterize the bearing white-etch cracking (WEC) failure mode.

Flender Corporation (formerly Winergy Drive Systems Corporation) and NREL collaborated under this CRADA from October 2017 to October 2023. Jonathan Keller was the NREL Principal Investigator (PI) for the project, while Shawn Doner was the original Flender PI. The objective of this work is directly related to CRADA 16-608 with SKF GmbH, so contributions from SKF also directly supported this CRADA with Flender. As a part of this effort, a Winergy 4410.4 gearbox with specially instrumented SKF cylindrical roller bearings (CRBs) supporting the high-speed-shaft (HSS) was installed in the DOE-owned General Electric 1.5 megawatt (MW) SLE wind turbine at the Flatirons Campus in December 2017. After commissioning, the system was used to collect gearbox and bearing data from January 2018 until the conclusion of this CRADA. The measurements were used to develop and validate NREL, Winergy, and SKF models for gearbox and bearing loads and remaining useful life predictions. The models were then used to evaluate different measures of roller slip losses that are considered as potential measures of the risk of creation of white-etching cracks in wind turbine gearbox bearings. Near the end of the original 3-year Period of Performance (PoP) in October 2020, the Flender PI was changed to Arthur Ohlrich and the PoP was extended for the first time until October 2021. In August 2021, the original gearbox rear upper housing, HSS, and SKF bearings were removed and replaced with a standard gearbox rear upper housing and bearings and a HSS that included the Winergy torque sensor (wTS). The PoP was extended two additional times to October 2023. At the conclusion of this CRADA, the Winergy gearbox with the wTS remain installed in the turbine.

Through the course of the project, multiple tasks were completed and are summarized in this final report for the CRADA. Task descriptions in this report follow the numbering in the Joint Work Statement of the CRADA.

Task 1a. Engineering Support for Instrumentation Development

Winergy Drive Systems responsibilities

Includes the following four subtasks:

1. Participate in collaborative to identify test goals and approach

2. Provide input and guidance to the collaborative on gearbox design and manufacturing issues

3. Provide detailed gearbox information for instrumentation purposes – specification, loads, ratings, drawings, models

4. Provide design analysis, if necessary, including gearbox modifications for instrumentation

NREL responsibilities

Includes the following two subtasks:

1. Lead the testing effort

2. Provide all test planning, integration, and scheduling

This task was divided into two parts. Early on in the project, Winergy provided NREL engineering support in the form of detailed information, including specifications, loads, ratings, drawings, and models, on the selected Winergy 4410.4 gearbox for the purpose of developing the instrumentation for the bearings and the gearbox (Keller 2016; Keller, Vaes, and McNiff 2016; Keller, Gould, and Greco 2019). A schematic of the plan for the instrumentation package is shown in Figure 1. In addition to the CRB roller and cage speed measurements, the HSS speed is measured with an encoder, and HSS torque and bending moments are measured with strain gauges. Additional instrumentation measures the tribological environment of the bearings, including the bearing inner and outer ring temperatures, the lubricant temperature and water content, air temperature and humidity within the gearbox cavity with respect to the wind turbine nacelle, and any stray electrical current across the bearings.

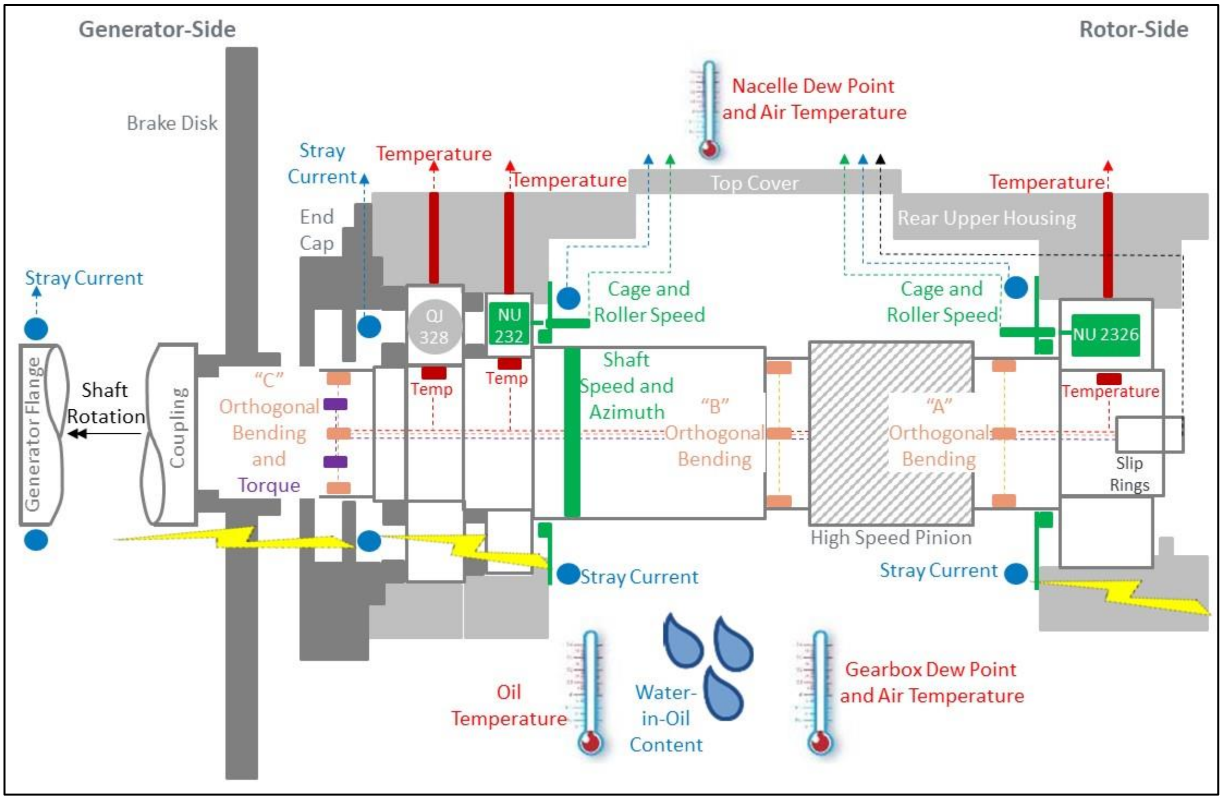

Figure 1. High-speed-shaft instrumentation schematic

Task 2. Gearbox Hardware and Instrumentation

Winergy Drive Systems responsibilities

Includes the following seven subtasks:

1. Provide a gearbox for instrumentation and testing
2. Provide machining of the gearbox high speed shaft and housing
3. Provide shop access the gearbox
4. Provide minor labor to assist in instrumentation installation, such as some bracket and mount fabrications, drilling holes for box mounts, wire passage, etc.
5. Provide and install Winergy gearbox instrumentation
6. Load test the gearbox and provide in situ calibration assistance, if necessary
7. Mount main bearing housing on main shaft, and mate the assembly with the gearbox

NREL responsibilities

Includes the following three subtasks:

1. Provide the test bearings
2. Provide NREL gearbox instrumentation – gauges, sensors, signal conditioning, data acquisition
3. Install and test the gearbox in a turbine

In this task, Winergy provided the gearbox and allowed NREL personnel to install the special SKF-NREL instrumentation package at the Winergy gearbox manufacturing facility in Elgin, Illinois. To accommodate the SKF-NREL instrumentation, the HSS, rear upper housing, bearing end cap, top cover, and oil distribution manifold were all modified by Winergy to NREL's specifications during the Summer of 2017. Most of the modifications were for minor wire routing and sensor attachment, but modifications to the HSS shown in Figure 2 were extensive to accommodate the instrumentation. The shaft diameter was reduced on either side of the gear teeth and to the rear of the locknut. The wiring for the instrumentation on the high-speed shaft was routed into radial holes to an axial through-hole in the shaft.

Figure 2. Machined high-speed shaft. Photo by Jonathan Keller, NREL 49040

The wiring then reached a custom Michigan Scientific SR36M-EL slip ring installed in a counter-bore on the rotor side of the shaft, as shown in Figure 3. The stator of the slip ring was then secured to the gearbox housing with an anti-rotation bracket.

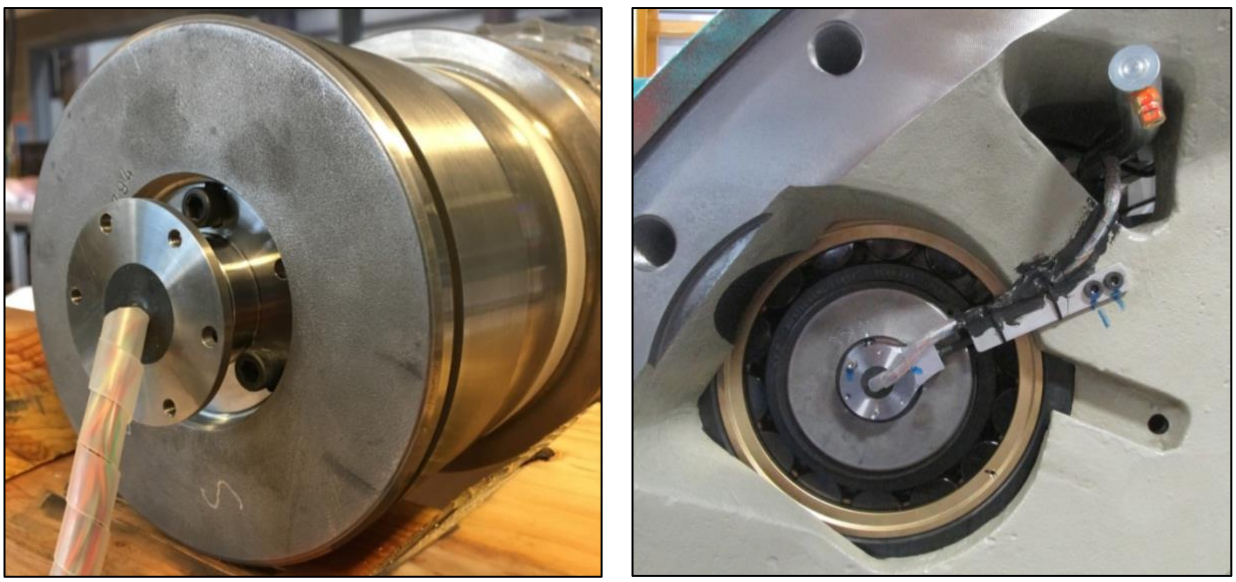

Figure 3. Slip ring (left) and anti-rotation bracket (right). Photos by Jonathan Keller and Shawn Doner, NREL 49043 and 49752

The remaining instrumentation was mounted to the rear upper housing, end cap, and oil distribution manifold. All instrumentation and wire routing were designed to accommodate the housing structural features and internal lubrication system, as shown in Figure 4. The internal wiring bundles were then routed to a set of connectors in a bulkhead on the top cover. NREL and Winergy personnel then assembled the production gearbox with the complete instrumentation package at the Winergy facility in Elgin in September 2017 (Keller 2018, Keller and Lambert 2018, Keller, Gould, and Greco 2019, Keller 2021a).

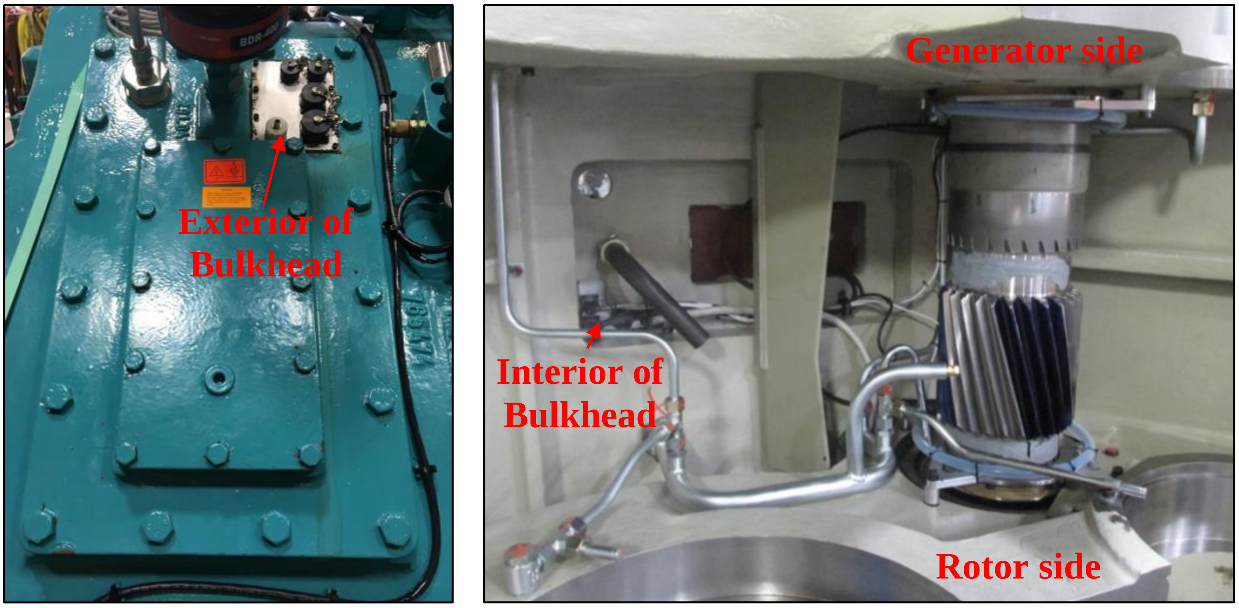

Figure 4. Exterior (left) and interior (right) bulkheads. Photos by Jonathan Keller and Shawn Doner, NREL 49049 and 49750

Once assembled, a standard production acceptance load test was conducted on the gearbox as shown in Figure 5. The load test consisted of an initial flushing of the gearbox, followed by operation at three speed and torque settings up to rated conditions over the course of 3 hours. Data for the instrumentation described in this report were collected during the load test with a system custom-built by NREL. In this manner, the HSS torque and bending gauges were also calibrated and CRB roller and cage speeds were compared to theoretical values (Keller and Lambert 2018).

Figure 5. Production acceptance load test. Photo by Shawn Doner, NREL 50080

Example bearing roller speed measurements are shown in Figure 6. The measurements are plotted against the circumferential location of the roller as it travels with the bearing cage. The orientation of the figure is viewed as though standing at the generator side of the gearbox and looking upwind toward the rotor. In this view, the cages rotate in the counterclockwise direction (the same direction of rotation as the HSS itself). The radial force exerted by the gear mesh on the HSS and hence the bearings, ignoring the effect of the weight of the shaft, brake disk, and generator coupling, is almost exactly to the right of the figure (+*y* or 270°). As seen in the figures, the bearing roller speeds are essentially the same as the theoretical speed near the center of the load zone (+*y* or 270°). As the rollers leave the load zone near the 330° azimuthal location, they slowly decelerate to about ⅔ of the theoretical value by the time they reach the 180° azimuthal location. Upon re-entering the load zone at approximately 210° azimuth, they quickly accelerate again to the theoretical speed.

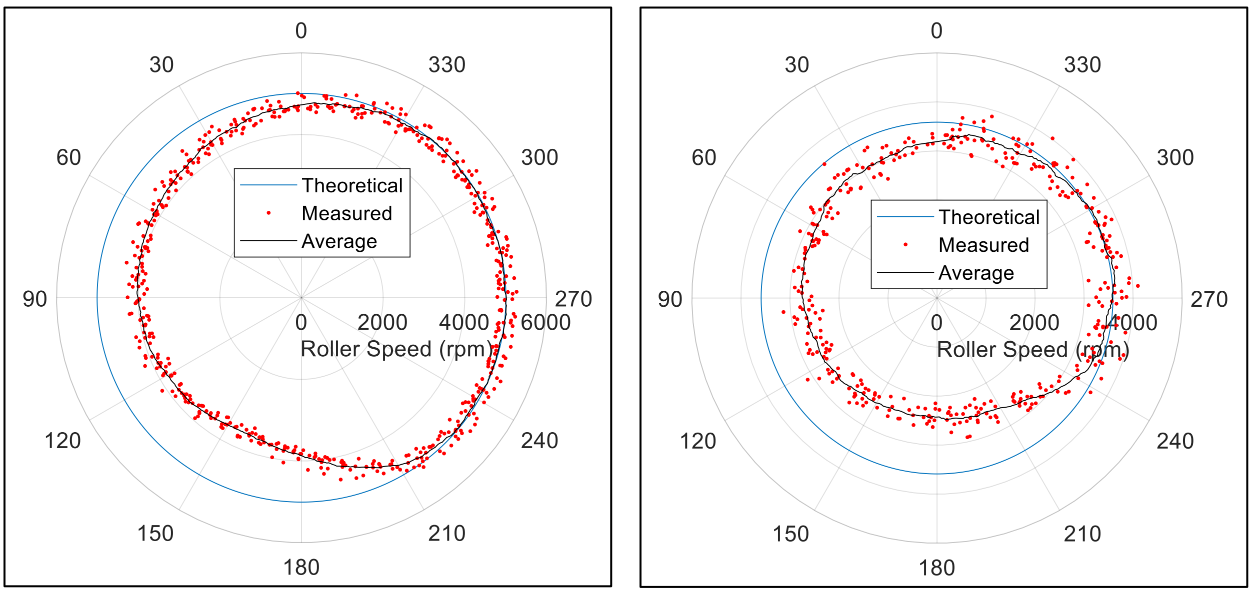

Figure 6. NU 232 (left, viewed from generator side) and 2326 (right, viewed from generator side) bearing roller speeds at rated power

After the production acceptance test and painting, the gearbox was then mated with the main shaft, provided by Winergy through this CRADA, and the main bearing, provided through CRADA 17-702 with SKF (Guo and Keller 2021), and then shipped to NREL as shown in Figure 7. The system was then installed in the GE 1.5-MW SLE wind turbine at the NREL Flatirons campus, as shown in Figure 8, in December of 2017. The gearbox measurements were time-synchronized with measurements from previously existing instrumentation on a meteorological tower in front of the turbine and on the turbine itself, including air temperature, pressure, and humidity; wind speed and direction at several heights, plus nacelle direction, rotor speed, and blade pitch angles; main shaft, tower, and blade loads; turbine power; and several supervisory control and data acquisition (SCADA) channels. The turbine and other instrumentation were recommissioned in January 2018, with full operations beginning in February 2018 (Keller 2018).

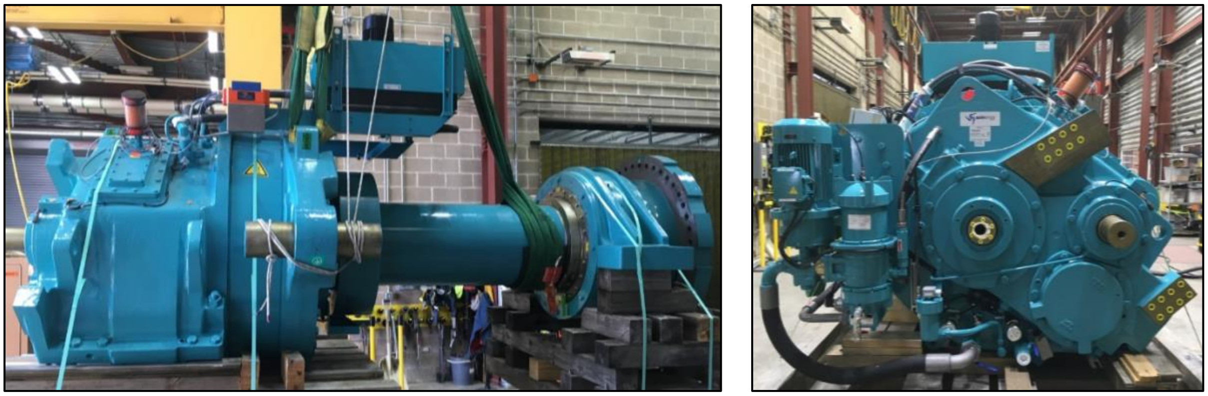

Figure 7. Winergy PEAB 4410.4 gearbox side (left) and rear (right) view. Photos by Jonathan Keller, NREL 49044 and 49045

Figure 8. Gearbox swap (left) and installation in turbine (right). Photos by Dennis Schroeder, NREL 49409 and 49413

Since then, drivetrain and turbine measurements have been acquired over a wide range of operating conditions, including power production in normally occurring winds; parked and idling situations; and intentionally induced transient startup, shutdown, emergency stop, and grid events (Keller, Guo, and Sethuraman 2019). From installation through January 2021 (3 full years), the turbine was operated (i.e., connected to the grid and producing power) for over 4,200 hours. By the conclusion of this CRADA in early October 2023, the turbine was operated for an additional 4,300 hours, for a total of over 8,500 hours of operation on the Winergy 4410.4 gearbox.

Task 1b. Engineering Support for Data Analysis

Winergy Drive Systems responsibilities

Includes the following subtask:

1. Provide data analysis and interpretation supported by gearbox simulations

NREL responsibilities

Includes the following four subtasks:

1. Provide modeling services, such as turbine modeling, drivetrain modeling and finite element analyses
2. Perform data documentation, archiving, verification, post processing and analysis
3. Distribute data within the collaborative group per the CRADA, and eventually distribute to the public
4. Publish NREL reports and journal articles documenting findings

Throughout the operation of the drivetrain and turbine, NREL, SKF, and Winergy collaborated on analysis of the data and interpretation of the results as supported by advanced simulations. Initial analysis focused on developing a thorough understanding of the bearing data and its processing, along with the main shaft and HSS torque and speeds. The NREL bearing model consists of a combination of two previously published analytical bearing models. With these models, the equations of motion are developed for the cage and rollers from the free-body diagrams shown in Figure 9. The CRB roller loads are the most important input to the model. They are derived from the CRB radial loads, which are estimated from the gear and shaft geometry and loads. The roller loads, lubricant temperature and shaft speed are then used to calculate the cage speed, friction between the cage and raceway, and roller speed for the most highly loaded roller. These outputs are then used to calculate the friction between the cage and rollers and, finally, the individual roller speed over its orbit. The analytical model can be used for both steady-state and dynamic operating conditions (Vaes et al 2019a; Vaes et a 2019b; Guo and Keller 2018).

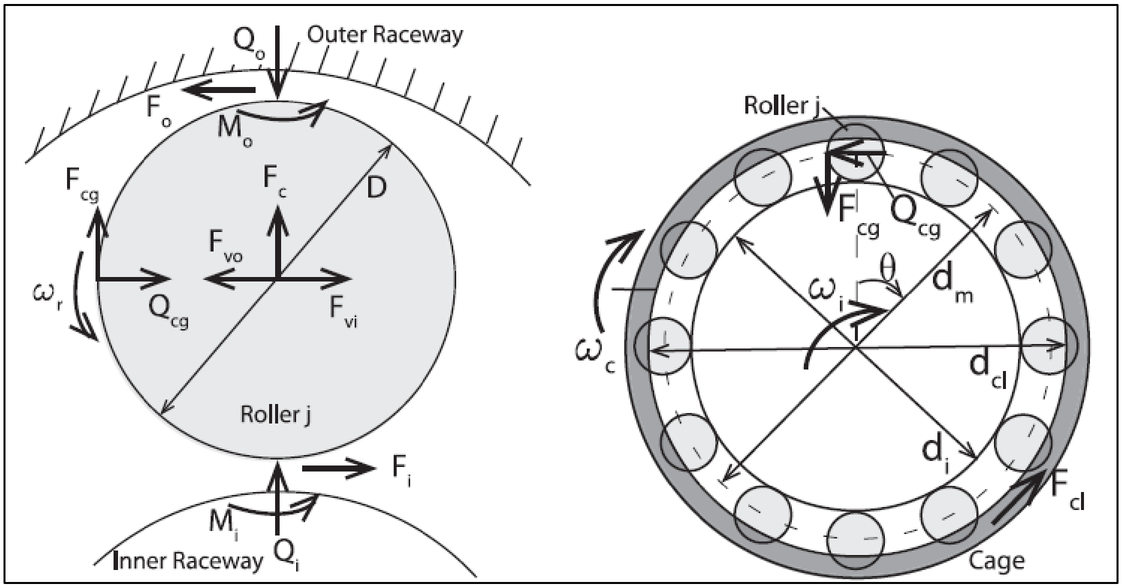

Figure 9. Fee-body diagram of a roller (left) and cage (right) in a CRB

The NREL analytical model was further validated across a wider range of turbine load and speed operating conditions, lubricant temperatures, and some transient events (Guo and Keller 2020). These conditions emphasize the importance of friction between the cage and rollers and the cage and rings (i.e., cage-landing friction). The predicted cage and roller slip compared to the measured results with reasonable accuracy across a wide range of near steady-state operating conditions and transient events. At its best, the analytical model predictions match experimental measurements within 10% for lubricant temperatures above 40° C and wind speeds over 10 meters-per-second (m/s). In steady-state conditions at low wind speeds and low lubricant temperatures, cage and roller slip up to 60% occur in the loaded zone of the bearing, indicating that none of the rollers even come close to pure rolling conditions. In the unloaded zone, up to 80% roller slip occurs. Cage and roller slip then decrease as the lubricant temperature and wind speed increases as shown in Figure 10. Depending on the wind speeds and lubricant temperature, these conditions could occur for extended periods in both normal operations and start-up conditions. Bearing slip is exacerbated by low oil temperatures that might more frequently occur in colder climates. This work led to the development of a Software Record (SWR) 20-34 titled “Roller Slip Simulator for Cylindrical Roller Bearings”.

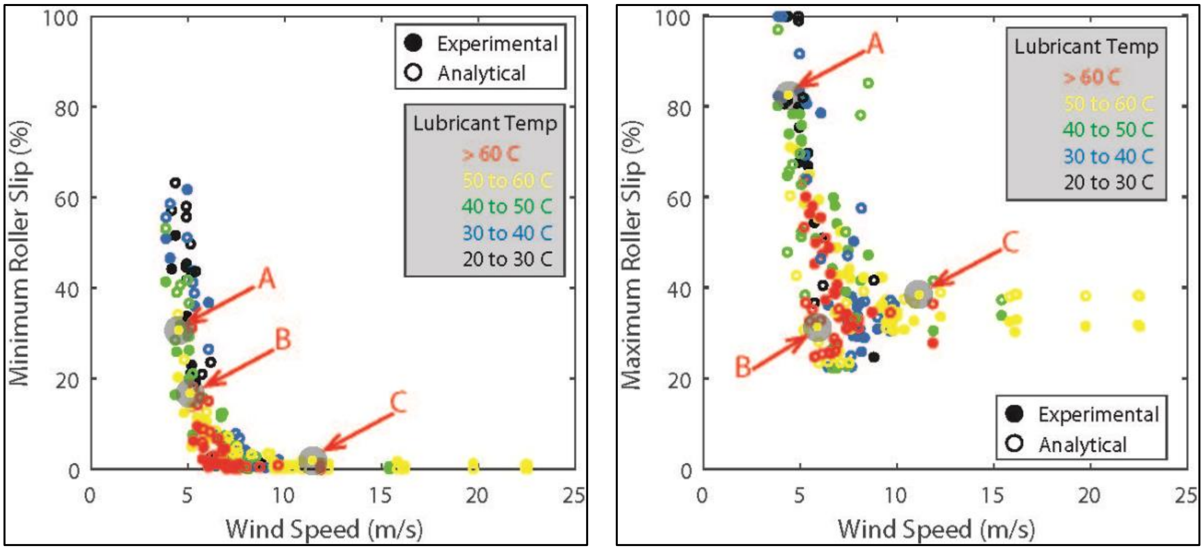

Figure 10. Minimum (left) and maximum (right) roller slip during normal power production

After validation of the roller slip models, they were used to evaluate the roller slip losses or cumulative frictional energy that are considered potential driving factors for WECs in wind turbine gearbox bearings (Vaes et al 2019a; Vaes et al 2019b; Guo, Keller, and Sheng 2019). As a measure to evaluate the risk for roller slip induced failures, the power slip density in the contact (in units of power in watts (W) per unit area in square millimeters (mm^2)), or and cumulative frictional energy (in units of joules (J)) or joules per second (J/s)) between the roller and inner raceway were used. As shown in Figure 11, the larger power slip density peak is always at the entrance of the loaded zone where the acceleration is the largest. The power slip density is generally higher in bearing A (i.e., rotor-side bearing) because it has larger rollers that offer more inertia against the friction force in the contacts. The cumulative frictional energy is a function of friction coefficient, normal load, and difference in velocity between two sliding surfaces over a period of time. Similar to the power slip density, both bearings accumulate the most frictional energy when entering the loaded zone because of the significant roller sliding. A much smaller amount of frictional energy is accumulated when exiting the load zone. The overall amount of energy accumulation in bearing A is more than twice that of bearing B (i.e., generator-side bearing).

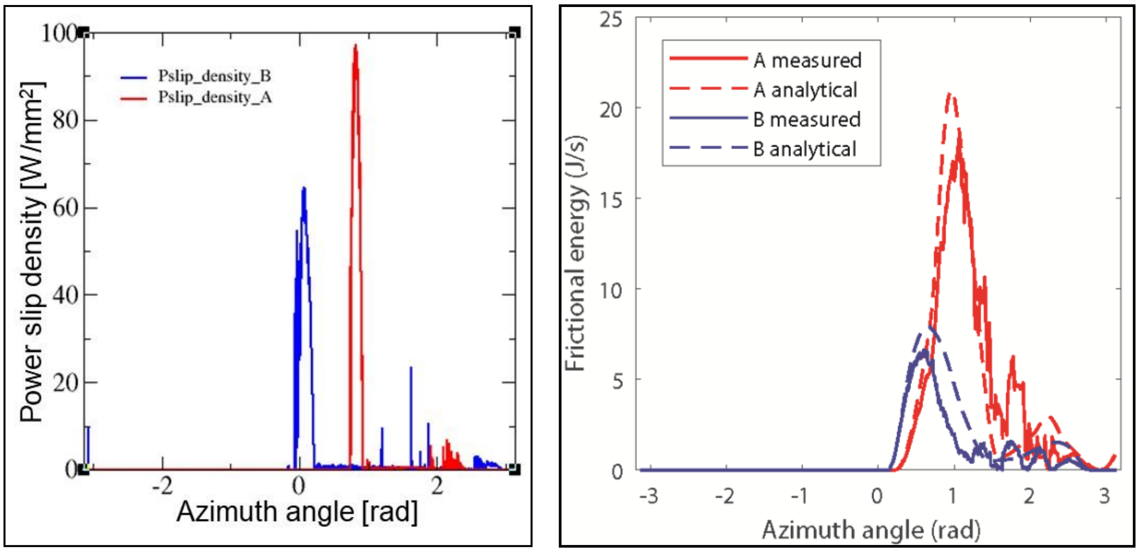

Figure 11. Power slip density (left) and cumulative frictional energy (right) for 80% torque

Because the combination of analytical models quickly estimates the bearing roller and cage speeds and the extent of roller slip, it is suitable for inclusion into models that assess the probability of failure (PoF). An interdisciplinary methodology to calculate the probability of failure of wind turbine gearbox bearings was then developed (Guo, Keller, and Sheng 2019; Guo and Sheng 2018; Sheng and Keller 2018; Veers et al 2018; Guo et al 2020). Assuming that the cumulative frictional energy is the mechanism causing bearing WEC, the methodology was used to calculate the probability of failure of each bearing in each gearbox in each turbine of an operational wind plant and compared to failure records over 10 years of operation of the plant as shown in Figure 12. Through statistical analysis of historical data, the methodology enables reliability assessment of axial cracking in individual wind turbine bearings and connects the reliability forecast with turbine design and operations.

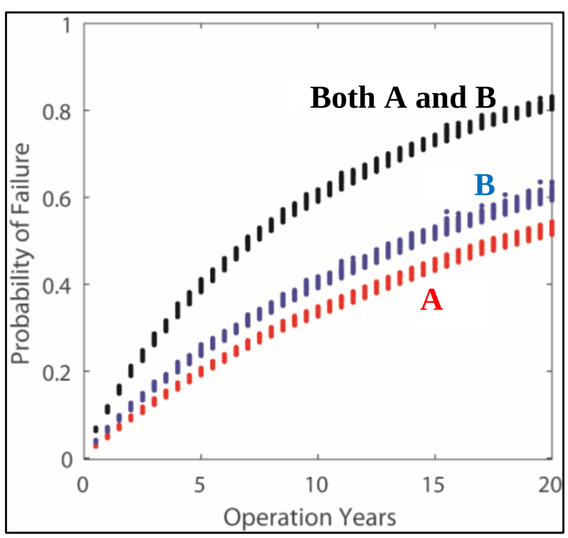

Figure 12. Probability of failure over time for cumulative frictional energy

The PoF model was also used to examine the effect of changes in the lubricant temperature-viscosity characteristics and the bearing mean radial internal design clearance. The study showed that changing the lubricant viscosity from 320-grade to 390-grade did not have an appreciable effect on the probability of failure. However, reducing the amount of time spent in cold lubricant conditions (Case 2) and increasing the average lubricant temperature (Case 3) reduced the PoF up to 4% as shown in Figure 12 (Clark et al 2023). Such a change can be achieved through settings in the turbine control system logic.

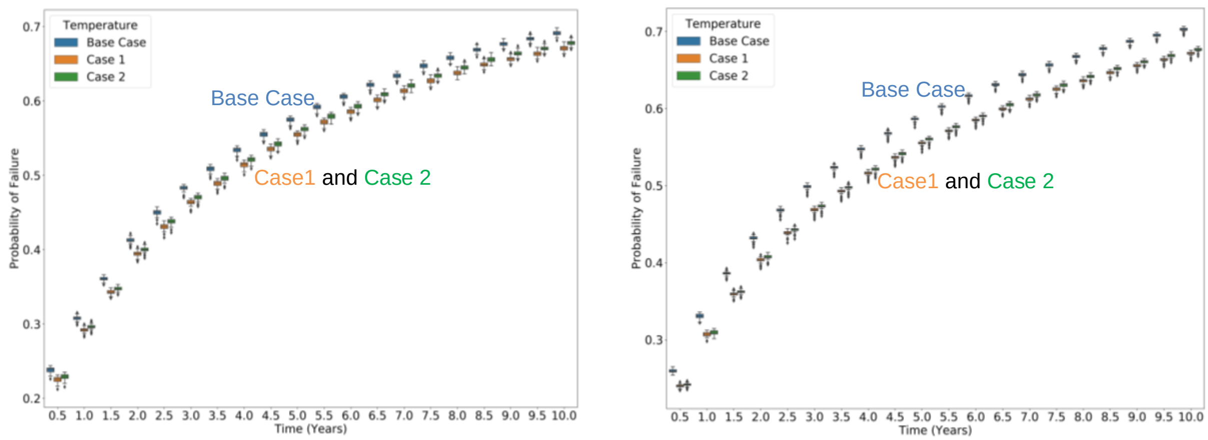

Figure 13. Effect of lubricant temperature on probability of failure for (left) bearing A and (right) B

Finally, digital gearbox technologies under development have the potential to improve O&M and reduce the levelized cost of energy (Reininga 2020; Doner 2020). In August 2021, the original HSS and bearing set was removed from the installed Winergy 4410.4 gearbox and replaced with a HSS that includes the Winergy torque sensor (wTS) and replacement bearing set as shown in Figure 15. The wTS collects high rate (typically 1 to 100 samples-per-second) gearbox HSS torque and speed data for use in advanced remaining useful life analytics.

Figure 14. HSS and wTS (left) and installation in turbine (right). Photos by Jonathan Keller, NREL 82235 (left) and 82234 (right)

An example of a turbine braking procedure is shown in Figure 15. Brake procedures can be mapped very poorly with supervisory control and data acquisition (SCADA) statistics, whereas 1 and 100 sample-per-second wTS data provides more realistic load distributions. The SCADA data only contains torque as a calculated value from electrical current and speed, which in the case of brake events, the current and calculated torque goes quickly to zero. But the actual mechanical torque has high values and fast oscillations. The combination of the wTS and advanced analytics is called the Winergy “Digital Gearbox”, which can potentially enable prediction of the Remaining Useful Lifetime (RUL) of gearboxes in order to safeguard functional performance, optimize maintenance planning and reduce costs (Winergy 2023).

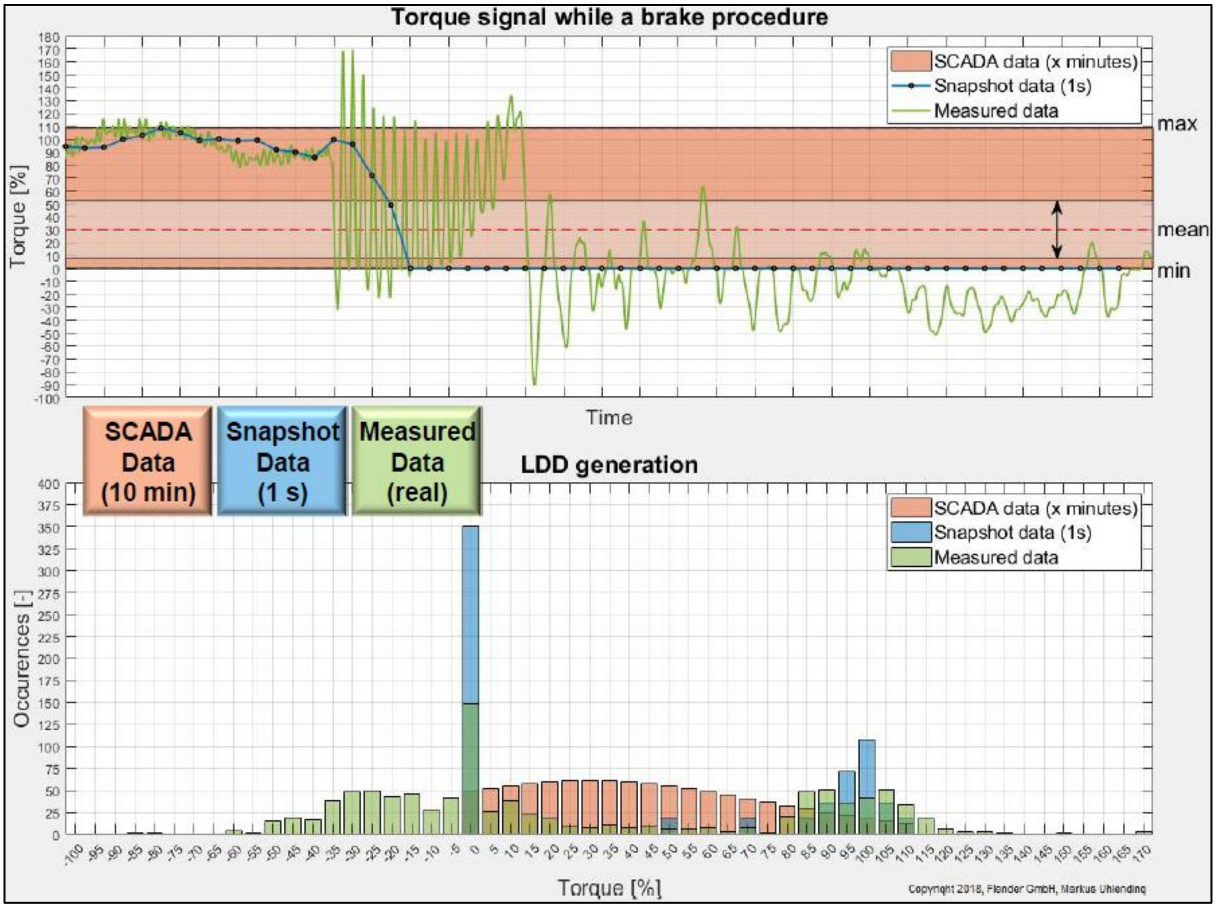

Figure 15. Torque and Load Duration Distribution (LDD) in a braking procedure (Doner 2020)

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