Experimental Characterization of a Grid-Loss Event on a 2.5-MW Dynamometer Using Advanced Operational Modal Analysis

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Contents

1 Abstract 2
2 Introduction 2
3 Approach 3
4 Test article 3
5 Dynamic event description 4
6 Experimental OMA identification of drivetrain resonance 4
7 HSS tooth strains 7
8 Gearbox high speed shaft (HSS) bending 8
9 Conclusions 8
10 Future work 9
11 Bibliography 9

1 Abstract

This paper experimentally investigates a worst case grid-loss event conducted on the National Renewable Energy Laboratory (NREL) Gearbox Reliability Collaborative (GRC) drivetrain mounted on NREL's 2.5-MW dynamometer test stand that is connected to a controllable grid interface. The GRC drivetrain has a directly grid-coupled, fixed-speed asynchronous generator. The main goal of the study was to assess the dynamic content of the high-speed stage of the GRC gearbox. In addition to external accelerometers, high frequency sampled measurements of strain gauges were used to assess torque fluctuations and bending moments both at the nacelle main shaft and gearbox high-speed shaft (HSS) through the entire duration of the event. Modal analysis was conducted using a polyreference Least Squares Complex Frequency-domain (pLSCF) modal identification estimator. The event driving the torsional resonance was identified. Moreover, the pLSCF estimator identified main drivetrain resonances based on a combination of acceleration and strain measurements. Without external action during the grid-loss event, a mode shape characterized by counter phase rotation of the rotor and generator rotor determined by the drivetrain flexibility and rotor inertias was the main driver of the event. This behavior resulted in significant torque oscillations with large amplitude negative torque periods. Based on tooth strain measurements of the HSS pinion, this work showed that at each zero-crossing, the teeth lost contact and came into contact with the backside flank. In addition, dynamic nontorque loads between the gearbox and generator at the HSS played an important role, as indicated by strain gauge-measurements.

2 Introduction

The wind turbine industry aims to continuously improve turbine designs to reduce overall cost. At the same time, turbine reliability needs to be guarded to secure predictable operating expenses over the designated product lifecycle of 20 years. Loads play an important role in the design of a wind turbine. Field measurements and dynamometer testing can be used to validate simulation models for dynamic load prediction [Hui 06]. However, simulation models as well as dynamometer test validation campaigns require definition of appropriate test cases derived from relevant operational tests. For initial model validity analysis, researchers may use quasi-static operating conditions. However, dynamic events such as start-ups, shutdowns, and emergency events can significantly reduce the overall drivetrain life and should be investigated. The main design load cases, including the dynamic ones, are defined in IEC 61400-1 [IEC 05] and linked to the control system. The control strategy used during a grid-loss event and the approach used for emergency braking has significant influence on the system response. The definition of the control strategy also has significant influence on the overall turbine behavior. One of the drivers in this event was the main torsional drivetrain resonance.

3 Approach

This paper identifies this driving resonance frequency on the NREL 2.5-MW dynamometer for the Gearbox Realiability Collaborative (GRC) drivetrain [Lin 11]. This particular resonance is important for the overall response during several dynamic events. To identify the resonance, a severe grid-loss event was simulated. The emergency stop of the dynamometer was activated to disconnect the generator from the grid after which the dynamometer and drivetrain coast to a stop. In contrast to a grid-loss event on a field turbine, the speed reduction in the dynamometer was realized slowly in order to be able to clearly identify the resonance. During the event, the drivetrain was interpreted as having a torsional spring between the rotor inertia and generator rotor inertia. The mode shape is characterized by the counter-phase rotation of the rotor and generator inertia, as shown in Image 1. The torsional energy stored in the system was released during the dynamic event causing decompression of the drivetrain equivalent spring. The eigenfrequency and corresponding mode shape were identified by means of a polyreference Least Squares Complex Frequency-domain (pLSCF) estimator [Gui 03]. Nonphysical falsely identified poles linked to harmonic excitations were taken out of the set. Moreover, gear contact behavior during the dynamic event was investigated by means of strain gauges at the root of the high-speed shaft (HSS) teeth.

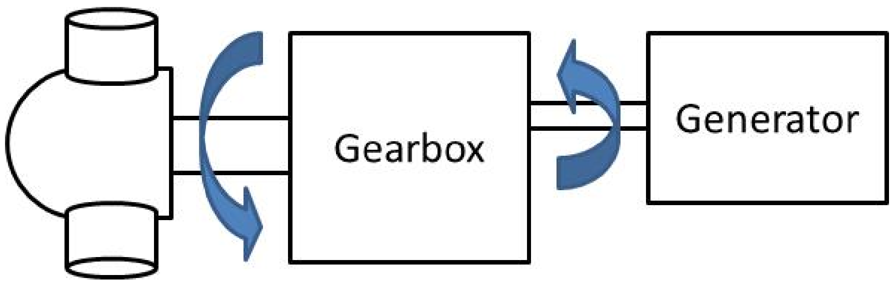

Image 1 Schematic representation of torsional drivetrain resonance

4 Test Article

This work used the GRC drivetrain that was designed for installation in a stall-controlled, three-bladed, upwind turbine with a rated power of 750 kW. The full nacelle is placed on the dynamometer for testing, as is shown in Image 2. The dynamometer consists of a driving electric motor combined with a gearbox to introduce mechanical power at the nacelle hub. During testing, the GRC drivetrain produces electricity, which is fed back into the local grid, as schematically shown in Image 2. The main advantage of this closed loop test configuration is that the nacelle controller stays active and the turbine produces energy as if it were in the field. The energy is fed into an electric micro-grid managed by a control unit that allows the variation of both grid frequency and amplitude. A detailed description of the dynamometer and the instrumentation can be found in [Lin 13].

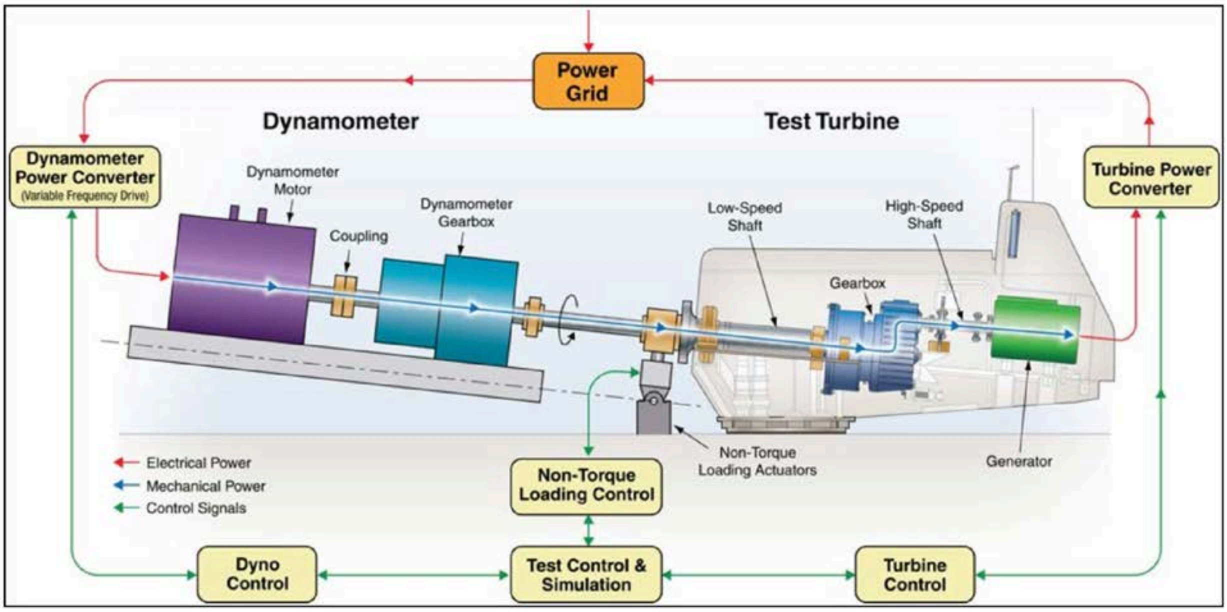

Image 2 Schematic of the NREL 2.5-MW dynamometer test stand [Lin13]

5 Dynamic Event Description

The test case started with normal operating conditions. At 3.2 seconds, the dynamometer emergency stop was activated, resulting in a disconnection of the electricity grid. Image 3 shows that the HSS speed decreased in a controlled manner and a torque reversal was initiated. The torque signal shown in Image 3 originated from the low-speed shaft (LSS) torque sensor. The torque values are scaled between -100 and 100 for easy comparison. The initial phase of the dynamic event shows a transient with torque pulses between the initial value and a negative maximum equal to approximately 80% of the initial value. The torque fluctuations continue until full stop, but are less pronounced after the initial 20 seconds. A fast Fourier transform (FFT) was performed on the torque reversal part of the signal, and it showed excitation in the frequency range below 1 Hz containing the first drivetrain eigenfrequency. An advanced operational modal analysis (OMA) was used to identify this mode shape and its corresponding eigenfrequency and damping value.

6 Experimental OMA Identification of Drivetrain Resonance

Operational modal analysis is used for systems subjected to environmental or mechanically introduced vibrations and is based on output only algorithms for system identification. This paper used the pLSCF approach [Gui 03]. Because the torque signal clearly determines the overall behavior of the system, the first drivetrain torsional resonance plays an important role. The corresponding mode shape, characterized by counter phase rotation of the rotor and generator inertia, is mainly determined by rotor and generator inertia values and drivetrain flexibility. The mode shape is illustrated in Image 1. To perform modal identification, an operational modal analysis was performed on the decay of three torque signals: the torque signal measuring the dynamometer response, the LSS torque signal, and the HSS torque signal.

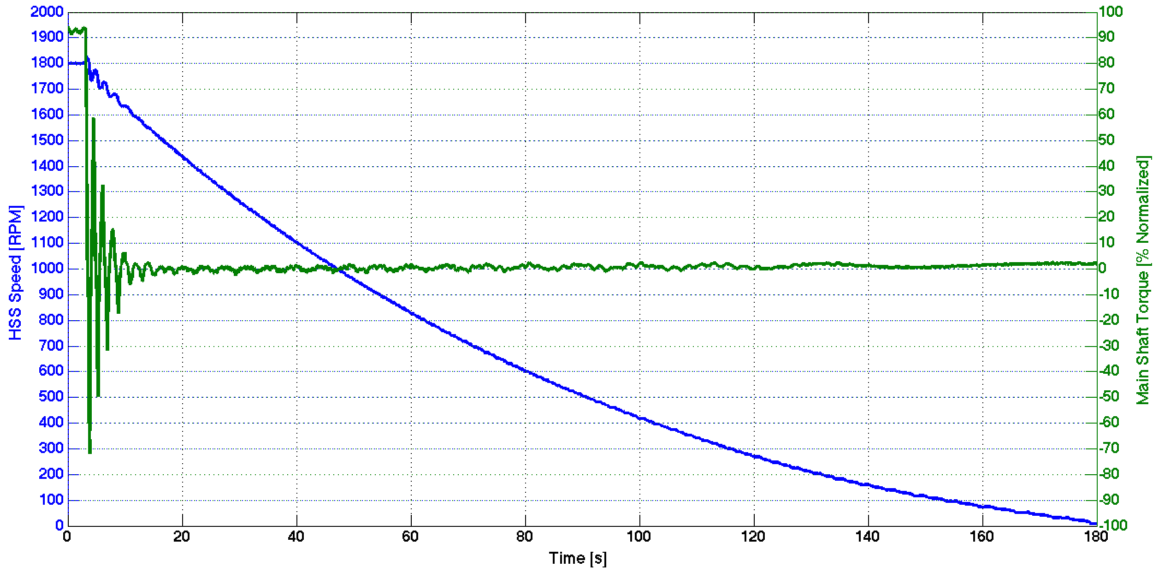

Image 3 Normalized torque and RPM signal during event

Eigenfrequencies, mode shapes, and damping estimates are generated directly by the pLSCF estimator. Image 4 shows the FFTs of the torque signals over a broad frequency range up to 60 Hz. A clear broadband excitation is present between 18 and 30 Hz and between 37 and 60 Hz, which was caused by smearing of the nominal harmonic excitations at 30 Hz and 60 Hz. Harmonic excitations are a challenge for the modal identification algorithms because modal estimators often misinterpret the harmonic excitations, especially when they are smeared, as resonance frequencies. Image 5 shows the pLSCF estimator stabilization diagram for a frequency band up to 5 Hz. The y-axis shows the model order of the fitted modal model. Broader frequency band analyses up to 60 Hz were performed but showed no modes between 30 Hz and 60 Hz indicating that in this frequency range, the modes might be masked by the harmonics. The influence of model order was also investigated. Three poles were found. Two poles, respectively at 0.64 Hz and at 2 Hz, were stable for multiple model complexity levels in both analyses. The first pole at 0.49 Hz only had stable identification for a limited number of model orders. Corresponding damping values are shown in Image 6. The main characteristic of the torsional modes was the counter phase behavior of the rotor inertia and generator rotor inertia. The identified modes showed a phase difference of about π radians, corresponding to the behavior shown in Image 1. Details on the sensitivity analyses are not discussed in this work for brevity reasons, but are detailed in [Hel 14]. Based on [Hel 14], the low number of stable model orders for the pole at 0.49 Hz in the stabilization diagrams and the clear overlap with the harmonic excitation regions, it is concluded that this pole is not a physical resonance. The same reasoning holds for the pole estimated at 2 Hz. Therefore, it can be concluded that the grid-loss event driving torsional mode shape has a corresponding eigenfrequency at 0.64 Hz.

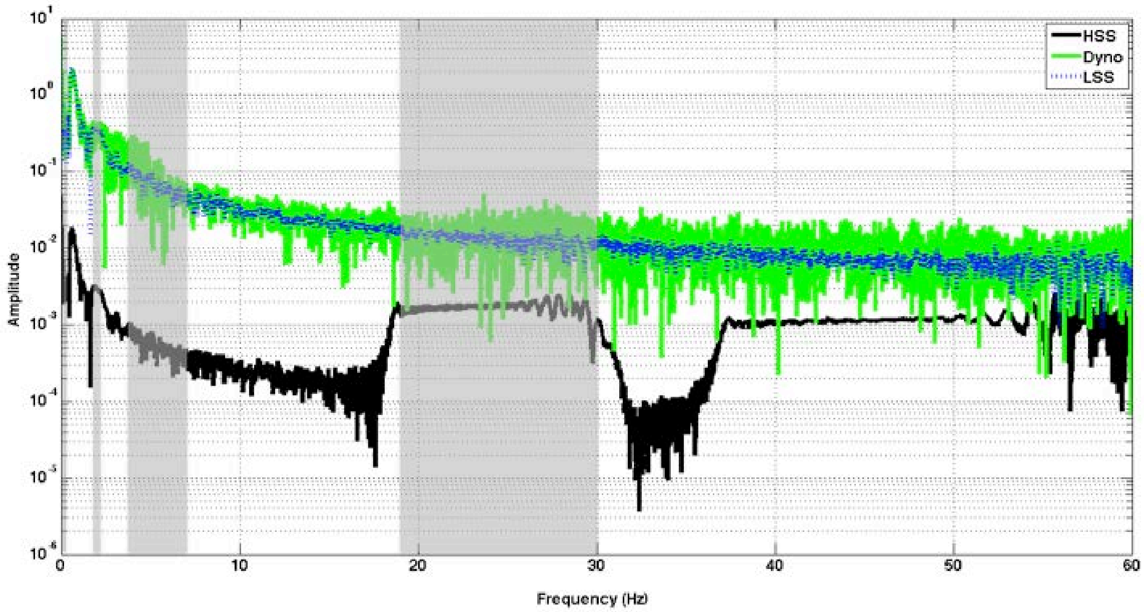

Image 4 FFT spectra of dyno, LSS, and HSS torque for frequency range [0-60 Hz]

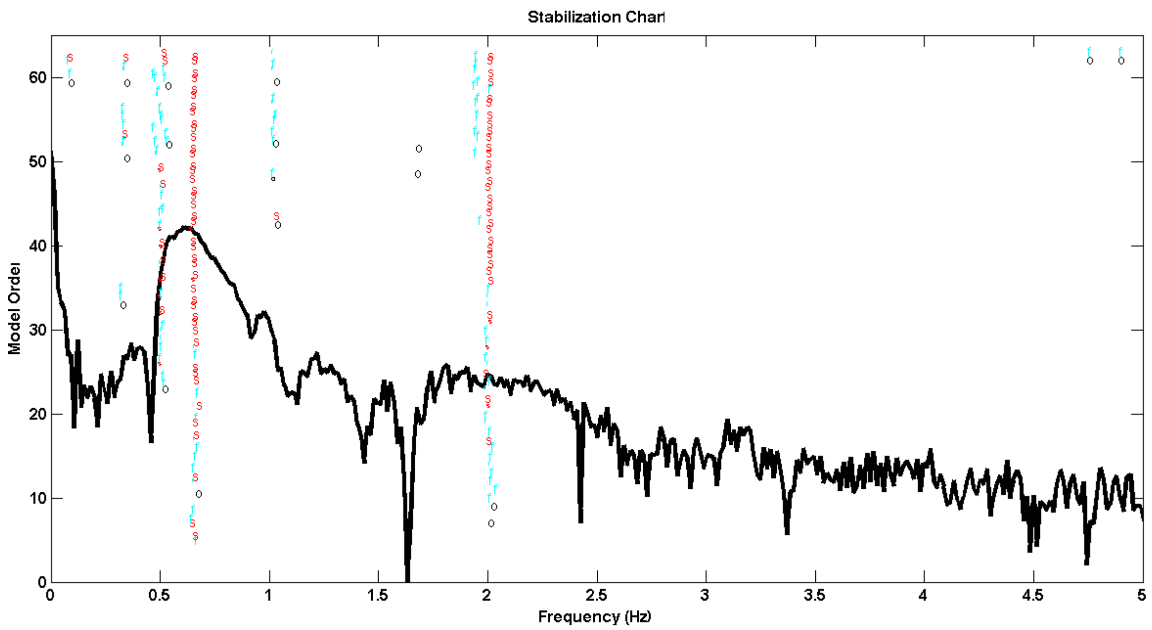

Image 5 pLSCF Stabilization diagram for frequency range for model order 64 [0-5 Hz]

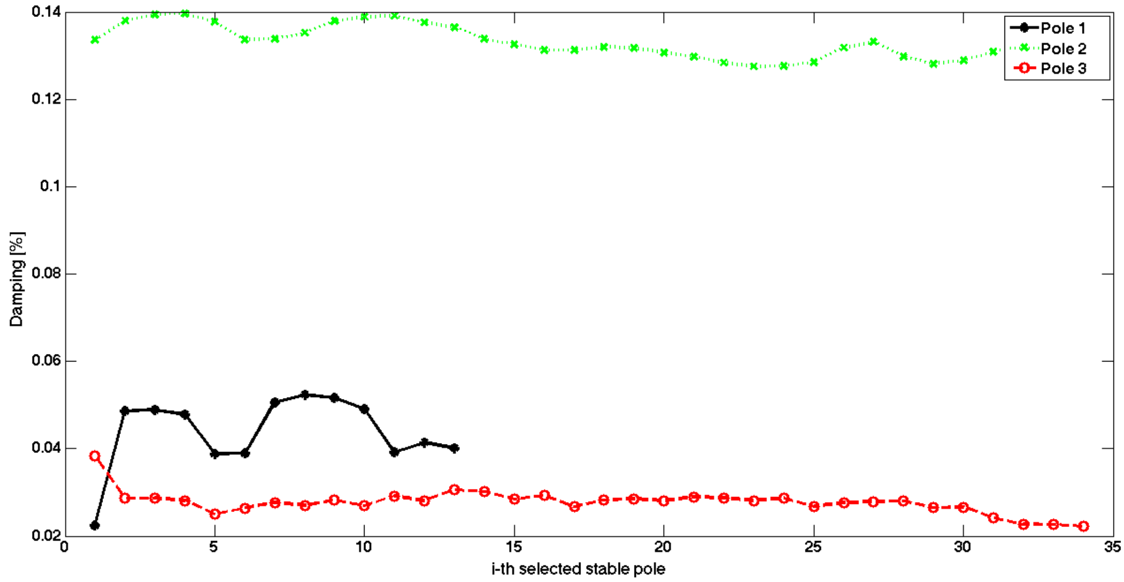

Image 6 Estimated damping values for the first three identified poles

7 HSS Tooth Strains

During the torque reversals, contact between the teeth was lost. The mesh went through the backlash and contact was reestablished between the coast-sides. In the GRC gearbox, strain was measured in the root of four teeth, covering the complete line of action for one mesh cycle. Image 7 shows these strains versus time for the first torque reversal. Each signal has a different color and all quantities are normalized for easy comparison. The first part of the graph shows the behavior during the decreasing positive torque period. Subsequently, the system went through the backlash and is subjected to negative torque. The backlash is clearly visible in the strain gauge signals.

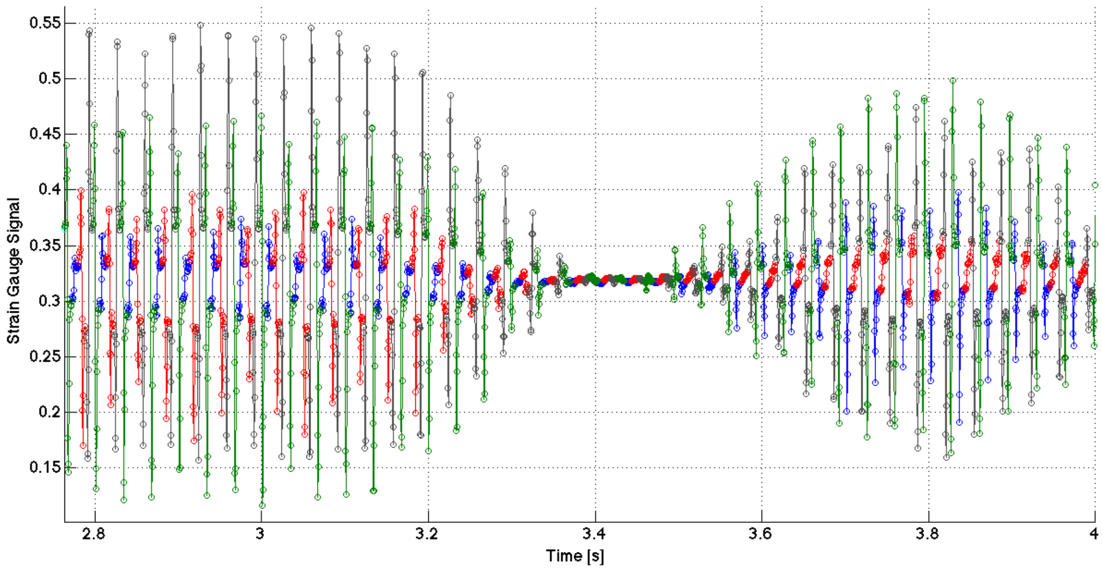

Image 7 HSS tooth strain, rpm, and torque evolution during transient event

Moreover, it can be seen that the strain gauge with dominant amplitude changed from the black one at positive torque to the green one at negative torque. Similar behavior is seen for the red and blue strain gauge signals. This illustrates the changing loading pattern in which the other tooth flank is contacted during torque reversal.

8 Gearbox HSS Bending

The loading induced during the grid-loss event was not strictly torque related. Image 8 shows the main shaft torque and corresponding total HSS bending. At the grid-loss event, the loads on the gearbox were reduced by the significant decrease in torque. Similar behavior can be seen in the HSS bending. After the torque increased, bending was experienced. When the system went through the backlash again, it was released. The subsequent positive torque pulse resulted in a smaller bending. Similar behavior was seen for the following reversals. Based on these measurements, it can be concluded that for this particular drivetrain architecture, the HSS loading during grid loss was not strictly torque driven and had significant bending loading, particularly during the negative torque periods. Further investigation should focus on whether misalignments between gearbox and generator are inducing these bending loads or if they originate from another source.

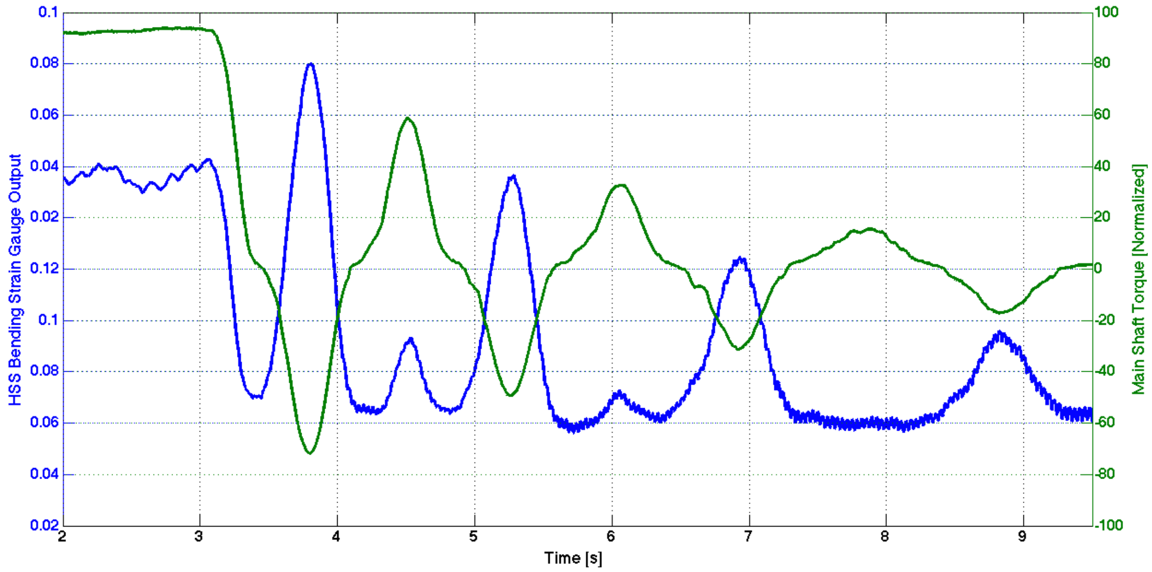

Image 8 Torque and overall bending evolution during transient event

9 Conclusions

This paper examined a grid-loss event conducted on the GRC drivetrain in the 2.5-MW NREL dynamometer. Behavior during this event was characterized by torque reversals inducing nontorque bending at the HSS and resulting in the HSS gear mesh disengaging, going through the backlash, and engaging the coast-side flanks for each torque reversal. Moreover, by means of a pLSCF based operational modal analysis, it was shown that the driving resonance was the first drivetrain mode at 0.64 Hz characterized by counter-phase rotation of the rotor and generator inertia.

10 Future Work

These experimental results will be used for further analysis of HSS bearing behavior. The combination of torque and bending load is expected to load the HSS tapered roller bearings in a way that differs from the steady state loading. Particular focus will be on slip characterization by means of the strain gauges in the bearing outer rings. Moreover, additional dynamometer testing conducted at different load levels will help in understanding the loads induced by these grid-loss type events. However, to have correct loads during the event, extensive field testing should be conducted so that representative event load cases can be applied during dynamometer testing.

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