Experimental Dynamic Identification Of Modeshape Driving Wind Turbine Grid Loss Event On Nacelle Testrig

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Abstract

This paper experimentally investigates a grid loss event on the Gearbox Reliability Collaborative drivetrain mounted on the NREL nacelle testrig. It is shown that during the grid loss event the system vibration is driven by a counter phase rotation of the rotor and generator rotor about the drivetrain flexibility. This behavior results in significant torque oscillations with significant negative torque periods. This work shows that at each zero-crossing the teeth disengage contact, go through the backlash and re-engage at the other flank by means of strain gauge measurements at the HSS pinion teeth. The driving torsional resonance is identified by means of a pLSCF modal identification estimator. Challenges for accurate modal parameter estimation related to harmonic excitation are elaborated and tackled. Finally the dominating eigenfrequency, corresponding modeshape and damping value are determined.

Keywords:

Wind Turbine, Emergency Stop, Operational Modal Analysis, System Identification

1. Introduction

The wind turbine industry is continuously improving turbine designs to reduce the overall cost of wind energy. At the same time turbine reliability needs to be guarded in order to guarantee a lifetime of 20 years. Expert insights in loading conditions are one of the main requirements for achieving this goal. The loads to which the turbine is subjected are fluctuating due to the stochastic nature of the wind [1] and cannot be assumed to be quasi-static as in most industrial applications[13]. Wind turbine loading includes aerodynamic loads at variable wind speeds, gravitational loads and corresponding bending moments, inertial loads due to acceleration, centrifugal and gyroscopic effects, operational loads such as generator torque and dynamic loads induced by certain control actions like blade pitching, starting up, emergency braking or yawing[4, 5]. In addition, loads caused by turbulence can result in unbalanced aerodynamic loads on different sides of the rotor. These loads lead to non-torque bending moments that feed into the drive train and induce non-symmetrical loads within the gearbox. The nature and importance of these non-symmetrical loads is significantly determined by the drive train configuration[3, 6, 7].

Field measurements and dynamometer testing are used to validate simulation models for dynamic load prediction[8, 9]. Simulation models as well as test-rig validation campaigns require the definition of appropriate loading conditions as input. For initial model validity analysis it is suggested to use quasi-static operating conditions. However, dynamic events such as start-ups, shut downs and emergency events can significantly reduce the overall drivetrain life and are therefore important to be investigated. For example rapid load variations combined with misaligned gear shafts in the high speed stage can lead to undesired axial load cycles at the bearings [10]. The main design load cases, including the dynamic ones, are defined in standard IEC 61400-1 [11]. In addition standard IEC 61400-4 WTs-Part 4: design requirements for WT gearboxes[12] includes requirements for normal gearbox operation. Examples are starting and stopping

procedures at cut-in, rated and cut-out wind speed linked to the control system concept. Another example is the control strategy for emergency braking procedure after activation of the safety system under grid failure. The definition of the controller strategy has significant influence on the overall turbine behavior. An example of such an event is the grid loss event. It is initiated by the loss of the electricity grid, followed by an emergency stop of the turbine. One of the main drivers in this event is the main torsional drivetrain resonance. This paper identifies this driving resonance frequency on the NREL 2.5MW nacelle dynamometer for the Gearbox Reliability Collaborative (GRC) drivetrain[14]. To do the identification a severe grid loss type event is simulated. The emergency stop of the dynamometer is activated to disconnect the generator from the grid and bring the speed of the drivetrain to zero in a controlled manner. The drivetrain can be interpreted as a torsional spring between the rotor inertia and generator rotor inertia. During this event the torsional energy stored in the system is released by decompression of this spring.

2. Study approach

The main goal of this paper is the identification of the main torsional resonance frequency of the GRC drivetrain on the NREL nacelle dynamometer. The corresponding modeshape is characterized by the counter-phase rotation of the rotor and generator inertias connected by the drivetrain flexibility, as schematically shown in Figure 1.

In the case of the nacelle testrig no full size rotor is present, as is the case in a wind turbine. This lower hub inertia will significantly affect the frequency value corresponding to the first torsional modeshape. However, the main goal of this paper is to show typical behaviour during a grid loss event and more particularly show that it is driven by this drivetrain mode. The change in frequency due to the extra inertia therefore does not influence the core message of this work. Since testrig testing offers significantly more controllability, it is opted to use dynamometer results in this work. The use of a full nacelle test-rig makes that

the relevant boundary conditions, other than the rotor inertia are similar as in the turbine, while keeping the convenience of test-rig testing.

The drivetrain resonance is excited by means of a severe grid loss excitation by disconnecting the generator instantaneously from the electricity grid after which the speed of the system is reduced to zero in a prespecified control mode. In contrast to the grid loss event on a field turbine, the speed reduction is realized slowly in order to be able to clearly identify the resonance. The eigenfrequency and corresponding modeshape are identified by means of a polyreference least-squares complex frequency-domain (pLSCF) estimator [15]. Moreover, gear contact behavior during the dynamic event is investigated by means of strain gauges at the root of the High Speed Shaft teeth. The goal is to identify the first drivetrain resonance frequency, which drives the torque reversals during which the gear contact is lost and the teeth contact the opposing flank. Harmonic excitation poses significant challenges in the modal identification process. These challenges are identified and the first drivetrain resonance determined.

Figure 1: Schematic representation of modeshape corresponding to driving modeshape

3. Test object

3.1. Test article description

The drivetrain used in this investigation was designed in the framework of the GRC and intended to be installed in a stall-controlled, three-bladed upwind turbine with a rated power of 750 kW. It has two rotor speeds, 14.7 rpm and 22.1 rpm. The gearbox converts these speeds to generator speeds of 1,200 rpm and 1,800 rpm achieving a ratio of 81.491. The GRC turbine follows a conventional, three point mounting modular configuration. These nacelle components are the hub, main bearing, main shaft, gearbox, brake, generator shaft, and generator as shown in Figure 2. The GRC gearbox is designed based on current best practices and consists of one planetary gear stage and two helical gear stages. Figures 3 and 4 show the overall gearbox layout. Turbine and gearbox properties are listed in detail in Table 1.

Figure 2: Nacelle of the GRC wind turbine [2]

The gearbox consists of one planetary and two helical gear stages. The planetary stage uses a cage planet carrier and a floating sun for optimal load sharing amongst the planets. The floating sun is connected to the first helical stage shaft by means of an articulated spline connection.

Figure 3: GRC gearbox [2]

Figure 4: GRC gearbox topology[2]

Table 1: GRC Drivetrain properties[2]

General Properties						
Power			750kW			
Control			Stall Control			
Generator			Asynchronous			
Rotor Speeds			14.7RPM and 22.1 RPM			
HSS Speeds			1200RPM and 1800RPM			
Nacelle configuration			Three-Point Mounting			
Gear Properties						
Gear	Teeth	Mate	Root Diameter	Helix	Facewidth	Speed
Element		Teeth	(mm)	Angle	(mm)	Ratio
Planet gear	39	99	372	$7.5^{\circ}L$	227.5	5.71
Ring gear	99	39	1047	$7.5^{\circ}L$	230	5.71
Sun Pinion	21	39	186	$7.5^{\circ}R$	220	5.71
Intermediate gear	82	23	678	$14^{\circ}R$	170	3.57
Intermediate pinion	23	82	174	$14^{\circ}L$	186	3.57
High-speed gear	88	22	440	$14^{\circ}L$	110	4.00
HSS pinion	22	88	100	$14^{\circ}R$	120	4.00
Overall speed ratio						81.49

91 *3.2. Set-up and Instrumentation*

92 The complete nacelle is tested in the dynamometer, shown in Figure 5. It
93 consists of a driving electric motor combined with gearbox to introduce me-
94 chanical power at the nacelle hub. This allows the GRC drivetrain to produce
95 electricity which is fed back into the local grid, as schematically shown in Fig-
96 ure 6. Main advantage of this configuration is that the investigated nacelle
97 controller stays active and the turbine produces energy as if it were in the field.
98 The electric micro-grid is managed by a control unit that allows the variation
99 of both grid frequency and amplitude.

Figure 5: Experimental set-up on 2.5MW nacelle test-rig. Photo credits: NREL 24914, Lee Jay Fingersh

100 The GRC gearbox has extensive instrumentation both on the outside and
101 inside. A variety of sensors is mounted consisting of amongst others strain
102 gauges for low frequency behavior, accelerometers for high frequency analysis,
103 oil particle and flow monitor sensors. Strain gauges are mounted on the rotating
104 and non-rotating components. Since the phenomenon under investigation is
105 related to torsion vibrations it are mainly the strain gauges which are used. To

Figure 6: Schematic of the NREL 2.5MW nacelle test-rig [2]

characterize overall behavior two strain gauge bridges are used to characterize the torque fluctuations at the drivetrain main shaft (LSS), –the input of the gearbox–, and at the High Speed Shaft (HSS) ,–the output of the gearbox–. Figure 7 shows the location of these bridge and their layout.

In addition to the torsional resonance behavior the tooth contact properties are of particular interest. More in particular the unloading of teeth during the dynamic event and the link to the applied torque loads are investigated. Strain gauge bridges in the root of the teeth of the HSS, shown in Figure 8 are used. Two bridges are used at a different center position. Since the HSS pinion teeth are helical the strain bridges are placed according to the helix angle. The advantage of using these two bridges is the ability to asses load distribution in the direction of the tooth width.

4. Dynamic event description

The load case used in this paper is a grid loss event. Initially the drivetrain operates at normal conditions. After 3.2s the dynamometer emergency stop is activated, resulting in a disconnection of the electricity grid. At that instance the generator induced counter torque is lost. The RPM is brought to zero

Figure 7: Low and High speed shaft strain gauge bridge for torque [2]

123 in a controlled slow manner over 3 minutes time. Figure 9 clearly shows the
 124 instantaneous decrease of active and reactive power to zero. Correspondingly
 125 the speed of the drivetrain starts to decrease in a controlled manner by the
 126 controller and a torque reversal is initiated. Figure 10 illustrates the torque
 127 and RPM behavior. RPM signals are measured at the HSS, whereas the torque
 128 signal originates from the testrig torque sensor. All quantities are normalized
 129 for easy comparison. The torque starts at 25% of the drivetrains nominal value,
 130 whereas the speed is at nominal RPM. During the first seconds of the transient
 131 the torque pulses between its initial normal value and a negative maximum equal
 132 to approximately 80% of this initial value. The torque fluctuations continue
 133 until full stop. Figure 11 shows the Fast Fourier Transform of the non-constant
 134 part of the torque signal starting at second 3.2 until second 18. There is clear
 135 excitation in the lower frequency range below 1Hz. The resonance frequencies
 136 of a full turbine in this range are predominantly related to tower motions and
 137 the first drivetrain eigenfrequency. Since no tower is present on the test-rig it
 138 is expected that the first drivetrain resonance frequency is driving the event.

Figure 8: High speed shaft pinion face width tooth load distribution sensors: Bridge One: HSP-KABCD, Bridge Two: HSP-K1234 [2]

Figure 9: Active and reactive power produced during grid loss event

Advanced operational modal analysis will be used to identify this modeshape and its corresponding eigenfrequency and damping value.

5. Gear loading

In Figure 11 it can be seen that the torque is constant during a short period between the different torque reversals. Each of these constant parts corresponds to the gears going through their backlash. This is confirmed by the strain gauge measurements at the HSS teeth. Figure 12 shows the normalized strain of one of the two HSS tooth strain bridges plotted in combination with the normalized torque signal. The tooth strains show similar behavior as the torque signal. Zooming in on the initial dynamics illustrates this, as shown in Figure 13. A clear beating phenomenon is seen in the two HSS tooth strain bridge strains. The fast fluctuating strain signal is related to the meshing of the gears. Teeth engage into contact and disengage. Since the strain gauge bridge is divided over four teeth, as shown in Figure 8 four distinct peaks are present for each revolution of the gear. Figure 13 zooms in on the strain signals. The four distinct peaks related to each revolution are clearly visible. The beating phenomenon is a clearly visible in the envelope of the high frequency strain signal. The

Figure 10: Normalized torque and RPM signal for grid loss event

Figure 11: Torque oscillation and corresponding frequency excitation spectrum

156 envelope is related to the overall fluctuation in the load on the teeth. By means
 157 of demodulation it is possible to extract the envelope signals from the original
 158 signals. These are plotted in Figure 14. The envelope signals that result from
 159 the demodulation process are clearly determined by the same trend as the torque
 160 reversals. Tooth loading is first decreasing towards zero, staying constant and
 161 increasing again to a maximum. Consequently it is decreasing again and so
 162 on. In between each of the beating events the teeth loose contact and are
 163 in backlash, as illustrated in Figure 14. Since at the same time the torque
 164 experienced by the test-rig changes sign the teeth are initiating contact at the
 165 non-driving flank while they are still at high speed. During the first seconds
 166 of the event the speed has only changed with 2.5-5%. The negative counter
 167 torque generates a braking effect, whereas the positive torque results in a speed
 168 increase. The speed fluctuation is limited and shown in Figure 13. At each
 169 torque reversal the teeth are experiencing an impact loading. The amplitude
 170 of these impacts may not be present in the strain measurements, since not all
 171 teeth are instrumented and it is not known whether the instrumented teeth are
 172 the ones participating in the initial engagement. Based on the envelope signals
 173 it can be concluded that the torque fluctuation is mainly determining the event.

Figure 12: Tooth strains for gauge bridge 2 combined with normalized torque signal for grid loss event.

Figure 13: Tooth strains for gauge bridges 1 and 2 combined with normalized torque and normalized RPM signal for grid loss event. Grid loss start indicated with dashed line

Figure 14: Envelopes of tooth strains for gauge bridges 1 and 2 combined with normalized torque signal for grid loss event. Grid loss start indicated with dashed line

Figure 15: Tooth strains for gauge bridges 1 and 2 combined with normalized torque and normalized RPM signal for grid loss event. Zoom on torque reversal

174 6. Identification of driving eigenfrequency and modeshape

175 6.1. Polyreference least-squares complex frequency-domain

176 The pLSCF estimation method is an extension of the LSCF method and
177 was first introduced to find initial values for the iterative maximum likelihood
178 method [16]. The LSCF method estimates a so-called common-denominator
179 transfer function model [17]. Quickly it was found that the LSCF method
180 yielded very clear stabilization diagrams with a very small computational ef-
181 fort [18, 19]. In the pLSCF, the common-denominator transfer function model
182 was replaced by a right matrix-fraction description (RMFD) transfer function
183 model to obtain a truly poly-reference implementation [15]. The most impor-
184 tant advantage of the LSCF estimator is the fact that very clear stabilization
185 diagrams are obtained [20, 21]. A complete background on frequency-domain
186 system identification can be found in [22].

187 6.2. Experimental identification

Figure 16: Signals Used in OMA analysis. Zoom on the HSS torque strain signal

Figure 17: HSS torque strain signal filtered by means of low pass filter with 10Hz filter frequency

188 Since the torque signal is clearly determining the overall behavior of the
 189 system it is expected that a torsional phenomenon is playing a crucial role in
 190 the event. The most likely driver of the behavior is the first drivetrain torsional
 191 resonance. The corresponding modeshape is characterized by counter phase ro-
 192 tation of the rotor and generator inertia and is mainly determined by the rotor
 193 and generator inertia values and the drivetrain flexibility, illustrated in Figure
 194 1. To perform modal identification an operational modal analysis is done on the
 195 torque strain gauge signals. Three signals are used: the torque signal measuring
 196 the dynamometer response, the LSS torque signal and the HSS torque signal.
 197 The discussed grid loss load case generates a broad excitation of the drivetrain
 198 and the decay of the signals is used in the operational modal analysis. Normal-
 199 ized signals used for visual illustration of the signal shape are shown in Figure
 200 16. The high frequency fluctuation on the HSS torque signal in Figure 16 is
 201 shown in zoom on in the small subfigure of Figure 16. The fluctuation is related
 202 to a phenomenon above 10Hz, since it is absent in the 10Hz low pass filtered
 203 HSS torque signal. Figure 17 illustrates this. Therefore, the high frequency
 204 behavior is outside the frequency range in which the first torsional drivetrain

205 mode is expected.

206

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208 An operational modal analysis (OMA) analysis is performed, using the decay
209 of the signal to allow for accurate estimation of the damping value correspond-
210 ing to the modeshape. Eigenfrequencies, modeshapes and damping estimates
211 are generated directly by the pLSCF estimator. Figure 18 shows the FFTs of
212 the used signals over a broad frequency range up to 80Hz. In this Figure a clear
213 broadband excitation is present between 18 and 30Hz and between 37 and 60Hz.
214 These ranges correspond to a smearing of the nominal harmonic excitations at
215 30Hz and 60Hz. Harmonic excitations are a challenge for the modal identifi-
216 cation algorithms, since modal estimators often misinterpret them as resonance
217 frequencies. Figure 22 shows the pLSCF estimator stabilization diagram for a
218 frequency band up to 50Hz. The y-axis shows the model order of the fitted
219 modal model. One stable pole is present at 0.64Hz. No other stable poles were
220 identified indicating that mainly a low frequency torsion based mode is excited.
221 In the excitation ranges at 30Hz and 60Hz no modes were found by the pLSCF
222 estimator. Therefore in this frequency range the modal identification is not
223 disturbed by the harmonic excitation.

Figure 18: FFT spectra of Dyno Torque, LSS Torque and HSS Torque signals for frequency band [0-80Hz]

Figure 19: Stabilization diagram for pLSCF estimator eigenfrequency identification for frequency band [0-50Hz]

224 To enhance the quality of the eigenfrequency estimation the modal identi-
 225 fication focusses in this section on a smaller frequency range. Main goal is to
 226 identify whether the identified pole is a single pole as identified in Figure 22 or
 227 multiple poles are present in this frequency range. Figure 20 shows the FFTs of
 228 the signals in the zoomed frequency range. In grey the gear meshing excitation
 229 zones are shown. In these zones it is expected that the pLSCF could experience
 230 difficulties in performing the modal estimation. To assess estimation quality
 231 two model complexity levels are used. The latter will help to identify whether
 232 poles are expected to be physical or just mathematical noise of the modal fitting
 233 algorithm. The resulting stabilization diagrams for pole picking are shown in
 234 Figures 21 and 22 for a model order respectively of 32 and 64.

Figure 20: FFT spectra of Dyno Torque, LSS Torque and HSS Torque signals

Figure 21: Stabilization diagram for Polymax eigenfrequency identification for frequency band [0-5Hz] for model order 32

Figure 22: Stabilization diagram for pLSCF estimator eigenfrequency identification for frequency band [0-5Hz] for model order 64

235 Three poles are found/ Two poles, respectively at 0.64Hz and at 1.99Hz are
 236 stable for multiple model complexity levels in both analyses. The first pole at
 237 0.49Hz shows some stable estimations for only a limited number of model orders.
 238 Figure 23 shows the eigenfrequencies for the poles estimated as stable for the
 239 different model orders in the stabilization diagram shown in Figure 22 . Stable
 240 values are found for all three frequencies. Nonetheless only two eigenfrequencies
 241 have a significant amount of estimations. Corresponding damping values are
 242 shown in Figure 24 . Two identified modes have low damping values whereas
 243 one modeshape has a higher damping value, which could be related to the
 244 fact that the speed is brought to zero in a controlled manner by the dyno
 245 controller thereby counteracting the system response. The main characteristic
 246 of the torsional modes is the phase behaviour of the rotor inertia and generator
 247 rotor inertia. Figure 25 indicates the phase difference between the modeshape
 248 rotational displacements of those inertias. All modes show a phase difference of
 249 about π radians, corresponding to the behaviour shown in Figure 1.

Figure 23: Estimated eigenfrequencies for selected stable poles in stabilization diagram shown in Figure 22

Figure 24: Damping values for selected stable poles in stabilization diagram shown in Figure 22

Figure 25: Phase difference between rotor and generator inertia for selected stable poles in stabilization diagram shown in Figure 22

250 Since significant gear related harmonic excitation is present in the system
 251 the OMA resonance estimates should be handled with care. In the OMA analy-
 252 sis no input forces are measured. Therefore attention should be paid to identify
 253 and separate the harmonic components from the structural modes and to re-
 254 duce their influence on the extracted modal parameters. Jacobsen [23] showed
 255 that there can be significant bias on the modal parameters obtained from OMA
 256 or even no estimated structural modes. Therefore potentially not all identified
 257 resonances are physical but also mathematical modes can result from the esti-
 258 mator. To get insights in the quality of the estimated modes a synthesis of the
 259 frequency response functions (FRFs) is performed.

Figure 26: Modal synthesis results using estimated eigenfrequencies and modes shapes. Black dashed curve: Measurements Red curve: Synthesis

260 In such an FRF synthesis a limited number of eigenfrequencies and mode-
 261 shapes is used to describe the system response. For proportionally damped
 262 systems this relationship between the synthesised FRF matrix $H(\omega_k)$ and the
 263 modeshapes is expressed by:

$$H(\omega_k) = \sum_{i=1}^N \frac{\{\phi\}_i \{\phi\}_i^T}{(\omega_{n_i}^2 - \omega_k^2) + j.2.\xi_i \omega_{n_i} \omega_k} \quad (1)$$

264 with:

265 N the number of calculated modes

266 $\{\phi\}_i$ the i-th mass normalized modeshape

267 ω_{n_i} the i-th natural frequency

268 ξ_i the i-th modal damping ratio

269 Results are shown in Figure 26. In the zones dominated by harmonic exci-
 270 tation around 0.4Hz and 2Hz, shown in grey in Figure 20 the FRF synthesis is
 271 of low quality. Peaks are present, which were not found in the measured signal
 272 potentially indicating the presence of mathematical poles.

Figure 27: Stabilization diagram for pLSCF estimator eigenfrequency identification for frequency band [0-5Hz]

Based on the synthesis results and the low number of stable model orders for the pole at 0.49Hz in the stabilization diagrams it is concluded that this pole is not a physical resonance. The same is expected for the pole estimated at 2Hz. To strengthen this conclusion Figure 27 shows the stabilization diagram when only the LSS Torque and HSS Torque signals are taken into account in the OMA estimation. In this case the pole is unstable for lower model order and splits in two poles for higher model orders. The latter in combination with the clear presence of harmonic excitation and non-stable FRF synthesis results indicates that the estimated pole is not a resonance frequency but a harmonic excitation. Therefore it can be concluded that the grid loss event driving torsional modeshape has a corresponding eigenfrequency at 0.64Hz.

7. Conclusions

This paper experimentally investigated a grid loss event on the NREL nacelle testrig. The system vibration was driven by a modeshape characterized by a counter phase rotation of the rotor and generator rotor about the drivetrain flexibility. This behavior results in significant torque oscillations with significant negative torque periods. At each zero-crossing the teeth disengage contact, go through the backlash and re-engage at the other flank. Strain gauge measurements at the HSS pinion teeth were used to show this. The driving torsional resonance was identified by means of an pLSCF estimator resulting in eigenfrequency, damping and modeshape. Harmonic excitation originating from the gears had a clear influence on the estimation and resulted in mathematical modes. It was shown that these modes were non-physical. However these harmonic excitations modes were not present in the frequency range of the torsional mode of interest resulting in a clear estimation of the eigenfrequency at 0.64Hz and its corresponding damping value and modeshape. The knowledge of this eigenfrequency and modeshape gives insights in the dominant response of the system during grid loss without external braking.

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