Identification of wind turbine main shaft torsional loads from high-frequency SCADA measurements using an inverse problem approach

The authors thank the reviewer for the valuable comments that improve the article quality. We have addressed all the reviewer's queries in detail in the following:

Reviewer 2

General comments

1) The first question concerns the simplified mathematical model of the drivetrain (Eqs 1-3). This model is a compact two-disk representation, in which the inertia of the main shaft and that of the gear box and high-speed shaft are collected into the equivalent generator inertia. Therefore, for the sake of consistency of the model, the stiffness term (K) seems to stand for the collective stiffness of the whole drivetrain rather than solely that of the main shaft [1]. However, the manuscript is taking whole drivetrain apart from the main shaft perfectly rigid, so above stiffness is assigned to the main shaft and will then be identified and utilised it to infer the main shaft loads, which may not be right for drivetrains with multi-stage gearbox and shafts. There are several references available (e.g. [2]) that explain the equivalent drivetrain models as the combination of springs in series. The angular displacement of such models are basically the summation of the individual displacements of the drivetrain components. A comparison with simulation results cannot simply validate the above assumption, as the turbine simulation tools are also usually including a simplified drivetrain model, as they aim to capture the global motion and loads of system. In case that the contribution of the other components are known/found to be negligible, this has to be quantitatively demonstrated or at least be discussed adequately. It is not well described whether this work is identifying the main shaft load or the total equivalent torsional load of the drivetrain. Nevertheless, the manuscript's methodology and findings are still interesting. As such, it is recommended to either clarify this point or discuss the probable limitations.

Response:

The two-mass model is widely used to study the torsional oscillations of the mechanical drivetrain of a wind turbine [1-5] because it captures the underlying free-free torsional frequency. This is the main driving frequency for the main shaft torsion. Though the governing equations (i.e., Eqs. (1-3) of the manuscript) remain the same for all the considered work. The underlying assumption regarding the stiffness of the various components in the drivetrain is different for each model and the definition for the 'K' in Eqs. (1-3) in each of these works is given in the following table:

Model	Considered stiffnesses for K
Boukhezzar et al. [1]	Rotor, generator and the main shaft
Shin et al. [2]	Main shaft
Berglind et al. [3]	Collective stiffness with rigid gearbox
Novak et al., [4]	Main shaft
Singh & Santoso, [5]	Main shaft and high-speed shaft, gearbox dynamics dynamic neglected
Girsang et at., [6]	Stiffness of all components were considered.

As seen in the table, the modelling assumptions regarding the gearbox and main shaft are inspired from [1-5]. However, for the inverse-problem based approach, the inputs to the drivetrain model (i.e., the rotor and generator speed) determine the contribution of each component to the 'K' value. For example, if the inputs are from the HAWC2 aeroelastic simulation then the estimated stiffness value has the contributions from the rotor and the main shaft only since the simulation does not account for the gearbox dynamics. If the inputs from the measurements are used, then the estimated stiffness is a collective stiffness that has a contribution from all the components.

To be consistent with all the inputs, the modelling assumptions adopted by Girsang et al.[6] has been referred in the revised manuscript (i.e., by considering the contributions from all the components in the drivetrain). Nevertheless, these modelling assumptions will not affect the methodology proposed herein.

The concerned sentences (i.e., page 3, line 86 and line 88 and Page 4, line 98 and 99) will be removed from the revised manuscript. Accordingly, the discussion about the inputs to the model will be included along with the reference (Girsang et al., [6]) in the revised manuscript. Since the modelling assumption is revised then the concerned figure (Fig. 2 of the old manuscript) does not add any value to the paper and hence it will be removed from the manuscript.

2) The other point is regarding the manuscript's argument that it just needs SCADA data, which sounds compelling due to this data being readily available. The suitable sampling rate of the SCADA data that is required for this methodology has not been discussed within the paper and needs to be closely clarified. This point becomes more important when the manuscript argues that their findings are beneficial to the calculations for life extension of wind turbine system. However, many of the existing turbines that would require life extension are equipped with SCADA systems with very low frequency output data, normally averaged values of order of minutes. It seems that this drawback is going to be removed by artificially increasing the SCADA signal sampling rate without providing any information on the original and the resulted sampling rate. Therefore, it is a far question whether this methodology is really applicable to SCADA data

or condition monitoring system's data is still required. As such, it is again recommended to either clarify this point or discuss the probable limitations.

Response:

We thank the reviewer for pointing out this fact. The proposed approach requires that the sampling frequency of the SCADA measurement be significantly higher than the dominant frequencies of the drivetrain torsional oscillations (i.e., 1p and 3p rotor excitation frequencies and torsional natural frequencies). As a result, the proposed method cannot be used for the turbines that have measurements in terms of 10-min SCADA statistics. The larger the turbine, the lower are the dominant frequencies of the torsional oscillations, hence for large size turbines the sampling frequency of SCADA measurements can be lower. In addition, we have used the phrase high-frequency SCADA in the article title as well. Also, the authors are neither proposing any method for the new data collection nor resampling the measurement data. It is demonstrated that with the existing high-frequency SCADA measurements, the site-specific torsional load can be estimated. This can be used to estimate the yearly damage without historical weather records or condition monitoring data as the wind speed and wind direction measurements are available in the SCADA measurements.

Also, the arguments regarding the estimation of the RUL have also changed to emphasize that those are not the scope of the current work.

We have rewritten the concerned paragraph of the introduction as follows:

The estimated shaft torsional stiffness and displacement are further used to identify the site-specific shaft torsional load. This novel methodology can potentially benefit wind farm owners since both the property of the structure in terms of its stiffness and the structural response and the site-specific load can be determined without requiring additional sensors or information from the wind turbine manufacturer. The significance of estimating the main shaft torsional load is that it affects the fatigue performance of other drivetrain components such as gearbox and planetary bearings (Dong et al., 2012; Gallego-Calderon and Natarajan, 2015; Ding et al., 2018). Hence, the same site-specific torsional load may be used as an input for quantifying the remaining useful life (RUL) (Ziegler et al., 2018) of the main shaft, gearbox, and other drive train components as well. The estimation of RUL/ yearly damage does not require additional historical weather data and condition monitoring data as the wind speed and wind direction measurements are available in the SCADA measurements. However, this is beyond the scope of present work. Further, the proposed approach requires that the sampling frequency of the SCADA measurement be significantly higher than the dominant frequencies of the drivetrain torsional oscillations (i.e., 1p and 3p rotor excitation frequencies and torsional natural frequencies). As a result, the proposed method cannot be used for the turbines that have measurements in terms of 10-min SCADA statistics.

3) Moreover, there are well-established integration techniques that are being used, even for online conversion of acceleration to velocity and displacement signals [3, 4]. In the absence of artificial noise in the simulated speed data, the type of drift shown in Figures 3 and 4 are apparently showing a linear trend due to the accumulation of the error from the initial value. This type of trend can be usually avoidable by the common digital filtering with restriction of frequency range within the pre-processing of the signal, particularly when the signal's mean value (static term) is going to be added later on separately, in which case the initial value of integration doesn't really matter. As such, the authors need to mention that the application of regularisation within a trapezoidal scheme is not the only available method to stably convert velocity into displacement.

Response:

Digital filters and frequency domain integration approach (FDIA) are the widely used techniques in literature to reconstruct displacements from the measured accelerations [7-9]. However, digital filters such as impulse response filters (IIR) and finite response filters (FIR) have several drawbacks when reconstructing the low-frequency displacements, as is the case here [9]. On the other hand, the FIDA methods are sensitive to the time intervals of the measurements [9]. Hence, a least square minimization based regularization technique called Tikhonov regularisation is used in the present study as it is better suited for low-frequency dominant structures as shown by [9].

The above discussion will be added into the introduction of the revised manuscript.

Minor comments:

1) The derivatives for the inverse integration scheme is similarly given in Hong et al 2008, so please refer to this paper in the beginning of the corresponding section.

Response:

The reviewer's suggestion will be incorporated into the revised manuscript.

2) Paragraph 15: "and" missing after strain gauges.

Response:

The reviewer's suggestion will be incorporated into the revised manuscript.

3) Paragraph 75: This is questionable if older turbines really possess SCADA with a desirable sampling rate that suits the manuscript's methodology.

Response:

Please refer to the response to the general comment 2.

4) Paragraph 85: As discussed previously, the assumption of the fully rigid gearbox and higher stage shafts, particularly for a multi-stage systems, needs further clarifications.

Response:

Please refer to the response to the general comment 1.

5) Paragraph 190: computationally "more" expensive.

Response:

The reviewer's suggestion will be incorporated into the revised manuscript.

6) Paragraph 205: Please clarify those three yaw directions.

Response:

The reviewer's suggestion will be incorporated into the revised manuscript.

7) Paragraph 210: where is the noise coming from that needs to be damped. Please clarify.

Response:

There are two sources of the noise: (i) numerical noise and (ii) measurement noise. This clarification will be added in the concerned place.

8) Paragraph 220: "the" modes.

Response:

The reviewer's suggestion will be incorporated into the revised manuscript.

9) Paragraph 225: pds should be expanded.

Response:

The explanation for pdf (i.e, probability density function) will be given in the revised manuscript.

10) Mixed use of symbols has to be avoided throughout the manuscript.

Response:

The reviewer's suggestion will be incorporated into the revised manuscript.

11) Eq 18, it seems that gearbox ratio is missing somewhere.

Response:

The gearbox ratio is used when converting the generator torque from the high-speed side to the low-speed side. T_g in Eq. (19) refers to the generator torque on the low-speed side which already accounts for the gearbox ratio. This explanation of the generator torque will be given in the problem formulation section of the revised manuscript.

12) Fig 8, it seems that some constant or near constant offset exist (observable in both time and frequency plots), apparently theta_static is not appropriately identified. The plotted FFT is too noisy, one can use a better illustration, perhaps, a power spectrum with sufficient averaging to get more clear peaks, as at the moment it is barely possible to get a good picture of the main peaks.

Response:

The constant offset is due to the use of the generator efficiency for all the static displacement calculations. As a result, there is an offset of 2.5 % about the mean value of the rotor torque. However, as an alternative approach, one can use the overall efficiency of the drivetrain as well which accounts for the efficiencies of all components for completeness.

Power spectral density (PSD) plots will be given in the revised manuscript instead of the FFT plots.

13) Paragraph 20: what are the sampling rate details and how the SCADA data's frequency resolution was increased, is it practically legitimate? Please clarify.

Response:

The sampling frequency of the SCADA measurement is 50 Hz. In the considered work, the authors have not increased the sampling rate of the measurement.

14) The results using the actual data does not look to add substantial value to the manuscript, as it is not really supporting the validation of the methodology.

Response:

The authors feel that including the results from the V52 measurements shows that the resulting trend is very similar to the trend observed from the HAWC2 results. Since RUL prediction on wind farms is based on the margins available on the actual structure as compared to prior aeroelastic computations, this result demonstrates the confidence in the proposed method to be utilized in RUL computations of wind farms.

References:

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