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A Prognostic Method for Fault Detection in Wind Turbine Drivetrains

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Abstract

In this paper, a prognostic method is presented for fault detection in gears and bearings in wind turbine drivetrains. This method is based on angular velocity measurements from the gearbox input shaft and the output to the generator, using two additional angular velocity sensors on the intermediate shafts inside the gearbox. An angular velocity error function is defined and compared in the faulty and fault-free conditions in frequency domain. Faults can be detected from the change in the energy level of the frequency spectrum of an error function. The method is demonstrated by detecting bearing faults in three locations: the high-speed shaft stage, the planetary stage and the intermediate-speed shaft stage. Simulations of the faulty and fault-free cases are performed on a gearbox model implemented in multibody dynamic simulation software. The global loads on the gearbox are obtained from a dynamometer test bench and applied to the numerical gearbox model. The method is

exemplified using a 750 kW wind turbine gearbox. The case study results show that defects in the high- and intermediate-speed bearings can be detected using this method. It is shown that this procedure is relatively simple, yet accurate enough for early fault detection in wind turbine gearboxes.

Nomenclature

Abbreviations

INP-A	Main shaft bearing
PLC-A, B	Planet carrier bearings
PL-A, B	Planet bearings
LS-SH-A, B, C	Low speed shaft bearings
IMS-SH-A, B, C	Intermediate speed shaft bearings
HS-SH-A, B, C	High speed shaft bearings
Main SH	Main shaft
IMS SH	Intermediate speed shaft
LS SH	Low speed shaft
NREL	National Renewable Energy Laboratory
GRC	Gearbox Reliability Collaborative
TE	Transmission error
LC	Load case
FFT	Fast Fourier transform
LMD	Local mean decomposition
CM	Condition monitoring
RAMS	Reliability, availability, maintainability and serviceability

Variables

$\mathbf{M}$	Mass/inertia of gearbox components
$\mathbf{C}$	Damping of gearbox components
$\mathbf{K}$	Stiffness of gearbox components
$\mathbf{x}$	Displacement of gearbox components
$\mathbf{F}$	Force/moment applied on gearbox components
ω_{MS}	Angular velocity, main shaft
ω_{LS}	Angular velocity, low speed shaft
ω_{IMS}	Angular velocity, intermediate speed shaft
ω_{HS}	Angular velocity, high speed shaft
e	Angular velocity error
e_{LS}	Angular velocity error, low speed shaft
e_{IMS}	Angular velocity error, intermediate speed shaft
e_{HS}	Angular velocity error, high speed shaft

e_t	Total angular velocity error
α	Inverse gear ratio
Z	Number of gear teeth
ϕ	Angular position
f_m	Gear mesh frequency
f	Frequency range
f_x	Frequency range for fault x
$E(\ )$	Normalised energy function
$\varphi(\ )$	Frequency spectrum

1- Introduction

The trend in wind turbine development is toward applications farther offshore, in deeper waters, using larger, multi-megawatt turbines. Difficult and costly offshore access, maintenance limitations and environmental restrictions demand new design considerations compared to land-based turbines. Moreover, the loadings on drivetrains, in particular on floating wind turbines, are very different than those on onshore turbines [1]. Gearbox solutions for offshore development demand higher reliability, availability, maintainability and serviceability (RAMS) than land-based designs.

Maintenance, in general terms, is classified into corrective and preventive actions [2]. Among preventive actions, condition monitoring (CM) is one approach to improving wind turbine availability. Condition monitoring is based on the fact that an incipient defect can be detected from changes in the system conditions. A condition monitoring system comprises sensors and data acquisition systems that collect vibration, noise, temperature and strain measurements or oil particle data during a predefined period, either online with an integrated measuring system or offline with portable instruments, on a regular basis [3].

Today, there are many condition monitoring systems available in the industry. In gearboxes and drivetrains, noise and vibration signal processing are used for fault detection [4-7]. For example, Elforjani and Mba [8] as well as Ghazali et al. [9] demonstrated the application of acoustic emission technology to detect damage in a single bearing in a test rig. Randall used acoustic measurements and proposed a signal decomposition method to distinguish the signals from gears and bearings in a gearbox [10]. Liu et al. [11] proposed a local mean decomposition (LMD) method that was applied to the gear mesh frequency signal. Wind turbine gearbox fault detection through vibration analysis was also investigated by Feng et al. [12]. Vibration measurement is claimed to have a high hardware cost and to produce false alarms [13]. Gearbox vibration is normally measured through sensors installed on the gearbox housing and supports. Although the overall health of a gearbox can be monitored by measuring the vibration of the housing, it is not possible to find the root cause or to detect the faulty gear or bearing component. One solution is to instrument all the gears and bearings, but this is neither practical nor cost effective in the wind energy sector. Another option is to use a vulnerability map of the gearbox, as introduced by Nejad et al. [14], which can be developed based on analysis during the design phase and used in conjunction with vibration data obtained during an inspection.

In this paper, a prognostic method for the fault detection of gearbox components is introduced. This method is based on angular velocity measurements from the input and output shafts, which are standard measurements already obtained in wind turbine control systems, and two additional angular velocity measurements from the intermediate shaft inside the gearbox. This approach is relatively simple, yet accurate enough to identify defects prior to failure and system damage. The method is exemplified using a case study of bearing defect detection in a 750 kW NREL gearbox model. The main novelty of the proposed method, apart from its simplicity, is that this approach can be implemented using an existing control system and does not require an additional condition monitoring platform.

2- Theory and Methodology

2.1- Multibody Simulation (MBS) Method

Numerical modelling methods, such as multibody simulation (MBS), can be used to calculate the dynamic load responses in a gearbox. A gearbox can be modelled as a system of rigid or flexible bodies interconnected with appropriate joints. In an MBS model, flexible shafts are imported from a finite element model with reduced degrees of freedom. Bearings are modelled as force elements with a linear or non-linear force-deflection relationship (see section 2.4). Gears are modelled as rigid bodies with compliance at the teeth. In general, the gearbox equation of motion in an MBS model leads to a form of [15]:

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F} \qquad (1)$$

where $\mathbf{x}$ represents the displacement vector, $\mathbf{F}$ is the external and internal force vector including torques and moments, and $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are inertia, damping and stiffness vectors of the bodies, respectively.

The Newmark method can be used for the time integration of the above equation [15-17], or MBS software such as SIMPACK [18] can be employed. In SIMPACK, the gear contact is modelled with consideration of the tooth geometry and modifications. Nevertheless, the accuracy of the MBS model results is directly dependent on the precision of the given stiffness, mass and damping values.

2.2- Case Study Gearbox Model

In this study, the drivetrain from the Gearbox Reliability Collaborative (GRC) project at the National Renewable Energy Laboratory (NREL) is used [19,20]. The GRC drivetrain is from a 750 kW turbine with conventional three-point supports (Fig. 1). Table 1 presents the GRC wind turbine specifications.

Insert Fig. 1 here.

Insert Table 1 here.

The case study gearbox has one planetary stage and two parallel stages. Its general specifications are listed in Table 2, and its topography is shown in Fig. 2.

Insert Table 2 here.

Insert Fig. 2 here.

The loads on the gears and bearings are obtained from the decoupled analysis; more information about this approach can be found in Nejad et al. [21,22]. First, the global loads on the drivetrain are measured using a GRC dynamometer test bench. One could also use the numerical results from global response analysis tools such as FAST [23] or HAWC2 [24]. Next, these loads are used as inputs to a multibody drivetrain model in SIMPACK. See Fig. 3 for an illustration of the gearbox model. The torque on the main shaft obtained from the dynamometer test bench is applied at the end of the main shaft where the rotor hub is connected.

Insert Fig. 3 here.

Sixty-second measurements of the torque are collected from the dynamometer test bench in a steady-state, non-transient condition under 100% of the torque at the rated wind speed condition and at 50% of the rated wind speed. The input torque measurement is then applied to the MBS model, and a simulation with a time step of 0.005 second is performed. The first 10 seconds of data are removed to avoid numerical convergence uncertainties. The MBS model used in this paper has been used in many earlier studies and verified with experimental data (e.g., [25-27]).

2.3- Angular Velocity Error

Gears with conjugate involute tooth profiles are expected to transfer motion with a constant angular velocity ratio, provided the input velocity is steady and constant. Theoretically, if ϕ_{in} and ϕ_{out} represent the angular position of a gear pair, it is expected that:

$$\phi_{out} = \frac{Z_{in}}{Z_{out}} \phi_{in} \tag{2}$$

where Z is the number of teeth. However, due to inherent manufacturing variations from a true involute profile, assembly misalignment and varying elastic deflection on the gear teeth, bearings and support structures, equation (2) is only valid in theory. The transmission error (TE) in radians is defined as [28-30]:

$$TE = \phi_{out} - \frac{Z_{in}}{Z_{out}} \phi_{in} \tag{3}$$

The TE represents the difference between the actual position of the output gear and the position the gear would occupy if it were perfectly conjugate [29]. Measuring the TE has been found to be a good indicator of the performance of meshed gears and representative of the geometrical deviations [31]. Normally, it is expected that an external excitation such as an out-of-balance mass will produce vibration in a mechanical system, but in gearing, the relative displacement or TE between mating gears generates the force between the teeth and the subsequent vibrations throughout the system [29]. If the TE is measured under static conditions, i.e., at a low load input, it is called the static transmission error. This static error is the result of manufacturing errors, whereas the dynamic transmission error results from both manufacturing errors and elastic deformations and defects in bearings and gears.

Dynamic transmission error measurement can be implemented in a wind turbine gearbox by measuring the angular velocities. Equation (3) can be written in the form of the angular velocities of the input and output shafts $\omega_{in}, \omega_{out}$ by:

$$e = \frac{d}{dt}(\phi_{out}) - \alpha \frac{d}{dt}(\phi_{in}) = \omega_{out} - \alpha\omega_{in} \tag{4}$$

in which e is the angular velocity error in rad/sec and $\alpha = \frac{Z_{in}}{Z_{out}}$ is the inverse gear ratio.

Fig. 4 shows the angular velocity error for the first stage of the case study gearbox at the rated wind speed.

Insert Fig. 4 here.

In a gear pair, the angular velocity error, similar to TE, is periodic, repeating over periods of both one tooth pitch and one revolution. Therefore, the frequency of TE excitation is often equal to, or near, the gear mesh frequency. The mesh frequency is defined as:

$$f_m = f_{in} \cdot Z_{in} = f_{out} \cdot Z_{out} \tag{5}$$

where:

$$f_{in} = \frac{\omega_{in}}{60},\ f_{out} = \frac{\omega_{out}}{60} \tag{6}$$

The mesh frequency of the bearings follows the same equations, except the number of teeth is replaced by the number of rollers.

A defect in a gear or bearing changes the angular velocity error (e). The idea presented in this paper is to capture gear or bearing defects through signal processing of the angular velocity error in the frequency domain. The reference fault-free angular velocity error is obtained from the gearbox multibody model (MBS) at two wind speeds, the rated wind speed and 50% of the rated wind speed. In practice, the reference values can be constructed from the MBS model for a range of operational wind speeds or can be measured during the gearbox back-to-back test in the gearbox production facility.

2.4- Fault-Free and Faulty Bearing Models

In this paper, the proposed method is applied to the detection of faults rising from the bearings. However, this procedure can also identify gear defects. Bearing damage is modelled by changing the stiffness and clearance parameters of the bearing in the MBS model. Rolling-element bearings are often employed in wind turbine gearboxes. They consist of an inner raceway, outer raceway, cage and rollers. In general, bearing damage is divided into two categories: wear and material fatigue [32]. In both defect categories, there is a loss of material or surface deformation of the bearing internal components, which changes bearing properties such as the internal clearances [33]. Based on the Harris elastic deformation model of rolling bearings [34], the clearance affects the load distribution factor and consequently the force-deflection relationship and the bearing stiffness. More information about bearing load analysis can be found in Jiang et al. [35].

A fault-free bearing is modelled by its initial clearance and linear stiffness in the multibody model. The damaged bearing follows a higher clearance and stiffness curve than the fault-free bearing. Fig. 5 presents the stiffness curve for the intermediate-speed shaft bearing in the local Y and Z directions. The local X direction is parallel to the main shaft axis in the downwind direction.

Insert Fig. 5 here.

The stiffness in the intermediate- and high-speed shaft bearings was changed from 10^9 N/m in the fault-free case to 10^6 N/m in the faulty case. However, the planet bearing was found to be too sensitive to the bearing stiffness. In the gearbox modelled in this case study, the planetary stage uses the floating-sun design concept, which can absorb limited deformations and deflections of the planet gear [27]. Many simulations were compared using various stiffnesses of the planet bearing, and it was found that a stiffness reduction on the order of

10% or higher created a large planet gear deflection and double edge contacts in the gearbox. Finally, the faulty planet bearing was modelled by a reduction in its stiffness to half of the fault-free case.

2.5- Frequency-Based Detection Applied to Angular Velocity Error Signals

Assuming that the frequency spectra of the angular velocity error signals are altered by gearbox faults, the energy content of different frequency ranges can be used to detect the faults.

The concept of using the energy in different frequency ranges of wind turbine sensor signals for fault detection was proposed by Odgaard and Stoustrup [36,37]. As suggested by Odgaard and Stoustrup, the frequency ranges can be implemented as bandpass filters [36], or a Karhunen-Loeve basis method can be used to detect changes in the frequency spectra to detect gearbox faults [37]. In these papers [36,37], a wind turbine benchmark model is used for fault detection and isolation and fault tolerant control. In the Odgaard et al. [38] benchmark, the gearbox is modelled using a linear state-space model with three states.

In the energy method, the frequency spectrum is obtained by applying a fast Fourier transform (FFT) to the angular velocity error data. $\varphi(f)$ is defined as the computed spectrum of the fault-free case, where f is the frequency range on which the spectrum is defined. If f_x represents the frequency range associated with a specific gearbox fault x, then $\varphi(f_x)$ is the frequency spectrum for the faulty case. The energy in the spectrum for the frequency range f_x can be computed by:

$$E(f_x) = \varphi(f_x) * \varphi(f_x) = \int_0^{+\infty} \varphi(\tau)\varphi(f_x - \tau)\,d\tau \tag{7}$$

In the faulty case, the energy function can be normalised by the total energy in the fault-free spectrum from:

$$E^*(f_x) = \frac{\varphi(f_x) * \varphi(f_x)}{\varphi(f) * \varphi(f)} \tag{8}$$

where $E^*(f_x)$ is the normalised energy of the spectrum. The energy levels calculated by this method are compared with a threshold value, which is estimated based on values obtained from fault-free cases. In practice, one can find this threshold by inducing faults in the multibody model and comparing with the fault-free cases. In the results section, it is shown that the energy approach can be used to distinguish the faulty and the fault-free cases.

3- Load Cases

Table 3 and Fig. 6 show the load cases considered in this paper. The reference load case, LC0, is the case with no defects in the bearings. The angular velocity error functions from this case are used as the reference values. In load cases 1 to 3, one bearing in the high-speed, low-speed and intermediate-speed stages, respectively, is considered to have a defect. All load cases are simulated at 100% and 50% of the rated wind speed. The velocity measurements and calculated error functions for each load case are also presented in Table 3.

Insert Table 3 here.

Insert Fig. 6 here.

The ratios $\alpha_1, \alpha_2, \alpha_3$ in Table 3 are inverse gear ratios for the 1st, 2nd and 3rd stages, respectively. The gear ratio is defined as the ratio of the input to the output angular velocities [39]. The angular velocities and sensor positions are shown in Fig. 7. The measurement of

ω_{MS} and ω_{HS} is normally performed in a wind turbine control system, whereas ω_{LS} and ω_{IMS} require additional sensors. It should be noted that only one bearing is faulty in each load case.

Insert Fig. 7 here.

4- Results

The frequency spectra of the angular velocity error functions are computed for the fault-free and faulty load cases at both the rated wind speed and 50% of the rated wind speed. To detect the fault, the energy content and relevant frequency ranges are then compared with the reference values obtained from LC0. The results are presented in the following sections.

4.1- Defect in the High-Speed Shaft Bearing (LC1)

In this load case, bearing HS-SH-A in the high-speed shaft is faulty. The fault is modelled by changing the bearing clearance and stiffness as described in section 2.4.

Insert Fig 8 here.

Insert Fig 9 here.

Insert Fig 10 here.

The analysis of the frequency spectra of e_{LC1} and e_{HS} presented in Figs. 8 and 9 indicates that the frequency ranges between 18.5-20.5 Hz and 59-62 Hz are relevant for fault detection at both wind speeds. If total error signals are used (as shown in Fig. 10), the frequency ranges of 41-41.2 Hz and 60-61 Hz are more suitable. Moreover, considerable differences in the FFT outcomes highlight the possibility of fault detection with or without an additional sensor on the intermediate speed shaft. The normalised energy of the spectrum is evaluated in section 4.4 for a range of f_1 selected surrounding the highest peak in the FFT results.

4.2- Defect in the Planetary Stage Planet Bearing (LC2)

Planet bearing PL-A is the faulty component in this load case.

Insert Fig. 11 here.

A comparison of the error functions e_{LC2} and e_{LS} provides no suitable detection frequency range. In Fig. 11, the comparison of the total error functions e_{t_LC2} and e_t shows the possibility of detection in the range of approximately 36.6-36.7 Hz. Therefore, the frequency range is selected as $f_2 = 36.6\text{-}36.7$ Hz for the spectrum energy analysis.

4.3- Defect in the Intermediate-Stage Bearing (LC3)

In this load case, the bearing in the intermediate speed shaft is faulty. The FFT results (Fig. 12-14) indicate the possibility of fault detection between 25-28 Hz and 77-82 Hz. The 25-28 Hz range contains the largest differences between the fault-free and faulty cases. Fig. 14 compares the frequency spectra of e_{t_LC3} and e_t in this range, and Figs. 12 and 13 compare the e_{LC3} and e_{IMS} error functions. The fault creates a clear increase in the energy level in the frequency range of 25-27.5 Hz, which is selected as f_3 for the energy spectrum comparison.

Insert Fig 12 here.

Insert Fig 13 here.

Insert Fig 14 here.

4.4- Normalised Energy Comparison of the Error Spectra (LC1, LC2, LC3)

For each load case, the normalised energy of the spectrum can be calculated in different frequency ranges. Table 4 presents the normalised energy function of the angular velocity error in three frequency ranges at 100% torque, i.e., at the rated wind speed. Table 5 shows the results for 50% of the rated wind speed.

Insert Table 4 here.

Insert Table 5 here.

A threshold equal to 3 is used to determine whether a fault is present. The threshold value to separate faulty from fault-free cases is found by a trial-and-error approach based on the values from the fault-free cases. Using the threshold of 3, from the results presented in Table 4, it is evident that the LC1 and LC3 fault cases can be detected using either the additional signals or the existing rotor and generator shaft signals without false positive detections. The energy values for LC2 are lower than the threshold, and thus this fault case is not detectable with or without the additional sensors.

5- Concluding Remarks

In this paper, a method for the fault detection of components in wind turbine gearboxes is introduced. The proposed procedure is based on angular velocity measurements from the rotor and generator shafts, with additional angular velocity sensors in the intermediate- and high-speed stages. The error function is then calculated from the velocity measurements. This approach is relatively simple, yet accurate enough to identify defects prior to failure and system damage. The method is exemplified by fault detection in a 750 kW NREL gearbox model.

Faults in gears or changes in bearing properties influence the angular velocity error and can be detected by the analysis of the error function and comparison with the reference values obtained from a reference MBS model. The error functions are then further investigated in the frequency domain. The faults alter the angular velocity error frequency spectra, and the energy content of different frequency ranges are used for detection.

The case study results show that bearing defects in the intermediate- and high-speed stages are relatively easily detected by this method. However, a fault in the planetary stage is difficult to detect prior to any significant failure. This is due to the complexity of the planetary gears and their sensitivity to local errors.

Additional sensors in the intermediate- and high-speed shafts can be used to identify the fault location more accurately. In the case of modular stage design, the faulty components can be replaced without affecting other stages.

It is also emphasised that the gearbox modelled in the case study uses a floating-sun design concept; it is possible that a similar fault in other designs (i.e., flexible pin planets) could be detected. Therefore, it is recommended that future work be devoted to investigating this method for fault detection in different gearbox designs.

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