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On model-based system approach for health monitoring of drivetrains in floating wind turbines

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Abstract

This paper deals with the health monitoring of a wind turbine drivetrain in a system perspective. A 5-MW reference drivetrain installed on a spar type floating wind turbine is selected. Degradation and damage in the main bearing which carries the axial loads can cause severe damage inside the gearbox as it causes non-torque loads entering the gearbox. Therefore, monitoring of the main bearing is very important. In this paper, different fault cases of the main bearing are considered and responses are obtained by the multi-body simulation. Gearbox failure is then evaluated in two different perspectives: one from the main bearing itself by analysing vibration, and one through studying the remaining fatigue life of another bearing inside the gearbox. The results reveal that not only the local vibration of the main bearing housing is important, but also the consequence of this degradation on other components can be a key factor in the “system” failure. This is a step forward in monitoring the wind turbine drivetrains on a system level.

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Keywords: Condition monitoring; offshore wind turbines; drivetrains; system approach

1. Introduction

The drivetrain is the heart of a wind turbine which converts the kinetic energy of the wind into electrical power. The condition or health monitoring of a drivetrain is an important task in order to increase the production hours and reduce the risk of catastrophic damages, in particular for offshore wind turbines with difficult and costly access. A wind turbine drivetrain typically includes a rotor, gearbox and a generator. Many methods are today available for the health monitoring of gearboxes, e.g based on vibration, temperature and metal particles. Vibration measurement for fault detection has a successful record in rotating machinery [1]. Different methods, either in time or frequency domain, can be used for the vibration analysis. Based on the level of identification, Rytter has classified the relevant methods on four levels [2]:

Level 1: damage identification

Level 2: finding the location of the damage

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Table 1: 5-MW reference gearbox specification [8].

Parameter	Value
Type	2 Planetary + 1 Parallel
Total ratio	1:96.354
Rated input shaft speed (rpm)	12.1
Rated generator shaft speed (rpm)	1165.9
Rated input shaft torque (kN.m)	3946
Service life (year)	20

Level 3: estimating the severity of the damage

Level 4: remaining life prediction

Wind turbine drivetrains' health monitoring systems use both frequency domain and time domain methods. For instance Nejad et al. [3] or Odgaard & Nejad [4] proposed a new prognostic method for wind turbine gearboxes in frequency domain or Ghane et al. [5] demonstrated the application of statistical fault detection methods in the wind turbine condition monitoring. Most of these methods normally cover up to level 3 in the Rytter's classification as the life prediction often is a separate subject with different approaches.

In an earlier work by Nejad et al. [6], it has been shown that reliability level of the components inside the wind turbine gearbox differ. In another work [7], it was also demonstrated that the damage of some components has sever effect on the other components inside the gearbox. A very good example is the main bearing in the drivetrain. In particular, the axial force induced by wind is mainly supported by the main bearing. The main bearing reduces the non-torque load entering the gearbox, thus as this bearing degrades, the dynamic loads on the bearings inside the gearbox increases [7–9]. Therefore, the health of bearings inside is very much dependent on the health of the main bearing.

In this paper, the condition of one bearing inside the gearbox is evaluated while the main bearing is suffering different extents of degradation and damage. The results reveal that not only the local vibration of the main bearing housing is important in the total failure of the system, but also the effect of the main bearing damage on other components can be a key factor in the total "system" failure.

2. Models

A 5 MW reference gearbox [8] mounted on the floating OC3 Hywind spar structure [10,11] is used in this study. This wind turbine is a 3-bladed upwind turbine with the rated wind speed of 12.1 rpm. The details of the spar structure used in this paper can be found in Nejad et al. [7].

The 5-MW reference gearbox used in this study follows the most conventional design types of those used in wind turbines [8]. The gearbox consists of three stages, two planetary and one parallel stage gears. Table 1 shows the general specifications of this gearbox. The gearbox was designed with a 4-point support with two main bearings to reduce non-torque loads entering the gearbox.

The multi-body system (MBS) model of this gearbox as well as its layout is presented in Fig. 1. As it is shown in the figure, the motions are applied on the bed plate and the external loads on the main shaft. The generator torque and speed is then controlled at the generator side [8]. The second main bearing (INP-B) is the one carries the axial load and is considered in this study. From inside the gearbox, PLC-B bearing is chosen - see Fig. 1b.

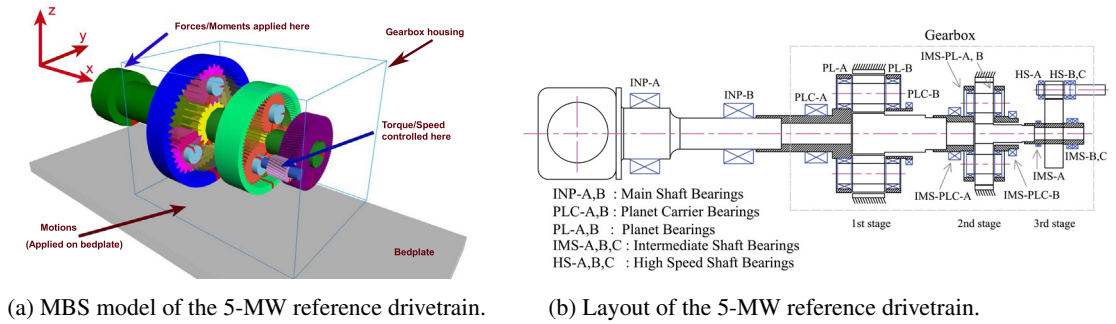


Fig. 1: The 5-MW reference drivetrain [8].

3. Methodology

3.1. De-coupled Approach

A de-coupled analysis approach was employed in this study. First, the forces and moments on the main shaft are obtained from the global response analysis. Second, they are used as inputs to a detailed gearbox model in an MBS model where simulations with a higher fidelity model and smaller time steps are carried out.

The global analysis is conducted by using an aero-hydro-servo-elastic code, SIMO-RIFLEX-AeroDyn [12]. Six 1-hour simulations are carried out at the rated wind speed with wave conditions characterized by significant wave height $H_S = 5$ m and peak period $T_P = 12$ s (modelled by a JONSWAP spectrum). The turbulence intensity factor is taken as 0.15 according to IEC 61400-1 [13]. The environmental data used in this study are taken from the site 3 from the study of Li et. al [14]. Earlier works on wind turbine gearboxes using the de-coupled method include, for instance Xing et al. [15] and Nejad et al. [16,17].

3.2. Main Bearing Model & Fault Cases

The reference gearbox used in this study consists of two main bearings [8]. The second main bearing (INP-B) is the one which carries the axial force induced by wind thrust force. Performance of this bearing is very crucial for the gearbox life [6,7]. As this bearing goes through the degradation and wear, more non-torque loads enter the gearbox and reduces the life of other components, particularly other bearings inside the gearbox. As highlighted by Musial et al. [18], most of the gearbox failures in wind turbines starts with the bearings. It is therefore very important to monitor and evaluate the condition of this main bearing during the operation.

In this study, the bearing is considered as one element which its life follows the Lundberg-Palmgren equation [19] as specified by ISO 281 [20]. The life of the roller bearings is limited to the fatigue life of the material from which they are made and is modified by the lubricant used. From the Lundberg-Palmgren hypothesis, the bearing life is expressed by [19–21]:

$$L = \left(\frac{C}{P}\right)^a \quad (1)$$

in which L is the bearing basic life defined as the number of cycles that 90% of an identical group of bearings achieve, under a certain test conditions, before the fatigue damage appears. C is the basic load rating and is constant for a given bearing. The parameter $a = 3$ for ball bearing and $a = \frac{10}{3}$ for roller bearings. P is the dynamic equivalent radial load calculated from:

$$P = XF_r + YF_a \quad (2)$$

where F_a and F_r are the axial and radial loads on the bearing respectively and X and Y are constant factors obtained from the bearing manufacturer [20]. Equation 1 is one form of an SN curve formulation which is used to estimate the fatigue damage of the bearings [6,7,22,23].

Table 2: Fault cases of INP-B bearing.

Fault case	K_x (N/m)	degradation level %
FC0	4.06×10^8	0
FC1	3.86×10^8	5
FC2	3.45×10^8	15
FC3	2.84×10^8	30
FC4	2.03×10^8	50
FC5	1.22×10^8	70
FC6	4.06×10^7	90

The roller bearing contact is often modelled by Hertzian contact theory [24]. In most of the bearing models, it is assumed that the bearings operate at moderate speeds and therefore the effects of centrifugal, gyroscopic and frictional forces are neglected and the force on rollers is expressed in the form of load-deflection relationship [25–27]. This force is a function of roller material and hardness, geometry and the applied load.

Guo and Parker [26] developed a finite element method to calculate the bearing stiffness. There are also analytical methods for bearing stiffness calculation, for example by Houpert [25]. New bearings have often very large stiffness values in the order of 10^8 and as the bearing wears and the surface hardness reduces, the contact zone increases and consequently bearing stiffness reduces. Such reduction can be seen in the experimental works conducted by Qiu et al. [28].

This property of the bearing provides a practical method for testing the damage detection methods in MBS models. Because of bearing complexities, bearings are modelled as a force element in an MBS model based on the bearing stiffness [8,27]. Bearing wear or damage can then be modelled by varying the stiffness matrix of the bearing and responses can be used for detection. This modelling approach has been used for modelling the bearing fault in land-based wind turbines [3,4].

The remaining life of the worn bearing primarily depends on the load level which gearbox operates. The experimental test conducted by Ocak et al. [29] on roller bearings shows that the time from the initial observation of high vibration until the ultimate failure can be as short as the 10% of total bearing life.

In this paper, the INP-B damage and degradation is studied through six fault cases as shown in Table 2. The damage is considered to be in the axial direction. K_x in Table 2 is the axial stiffness.

4. Results & Discussions

The damage in INP-B bearing results in additional loads transferring to bearings inside the gearbox. Thus, in addition to the INP-B vibration, the remaining life of bearings inside should also be calculated. Based on the vulnerability map described by Nejad et al. [6,8], the planet carrier bearing in the first stage (PLC-B) is selected in this study. Table 3 presents the remaining life of this bearing for different FCs. In this table, the remaining life is presented in normalized term with respect to the life at FC0, and the remaining life less than 50 % is considered to be critical - or in the red zone.

The vibration level of INP-B bearing itself is also evaluated. The root mean square (r.m.s) value of the vibration amplitude is one of the methods used in the industry [30–32]. According to ISO 20816-1 [32] the vibration velocity is often found sufficient for evaluating the severity of the vibration. ISO 20816-1 has classified four evaluations zones:

Zone A: new machines

Zone B: acceptable zone for long-term operation

Zone C: unacceptable for long-term operation

Zone D: can cause severe damage

Table 3: Remaining life of PLC-B.

Fault case	remaining life %
FC0	100
FC1	93
FC2	78
FC3	57
FC4	33
FC5	14
FC6	3

Table 4: r.m.s vibration velocity of INP-B.

Fault case	degradation level %	r.m.s (mm/s)	vibration zone
FC0	0	0.8	A/B
FC1	5	0.9	A/B
FC2	15	0.9	A/B
FC3	30	2.2	B/C
FC4	50	2.4	B/C
FC5	70	8.3	C/D
FC6	90	7.7	C/D

The r.m.s of the velocity for INP-B bearing in different FCs are calculated and evaluated based on the limits suggested by ISO 20816-1 [32] - see Table 4. It should be noted that the limits provided by this standard are based on the experiments from other industries. As of now, there are limited experience to conclude limiting values for large offshore wind turbines, but some field data [30] shows lower limits than ISO values for wind turbine gearboxes. For wind turbine drivetrains, there is only ISO 10816-21 [33] which recommends vibration limits for the onshore wind turbines smaller than 3 MW.

It is interesting to compare these two tables, Table 3 and Table 4. For example, assume a 7 years old turbine where the components have already consumed about 35 % of their fatigue life. If the fault case FC4, with 50 % degradation in INP-B occurs, the vibration signal of INP-B does not indicate necessarily to shut down the turbine as it appears to be still in an acceptable range. However, this fault condition reduces the life of PLC-B to 33 %, and since this is a 7 years old turbine where already 35 % of its life consumed, it is very likely that PLC-B fails in a very short time. Therefore, even though the vibration level of the main bearing is within the acceptable range, the load transferred to the gearbox may damage the bearings inside (PLC-B in this case). It is also interesting to note that the 5 % damage in INP-B can reduce about 10 % of the life of PLC-B.

5. Conclusions

In this paper, the health monitoring of the offshore wind turbine drivetrains is studied in a system perspective. The 5-MW reference drivetrain multi-body model on a floating spar wind turbine was selected and a de-coupled analysis method was employed for the load and load effect analysis. The drivetrain consists of two main bearings in which one carries the radial load and the other one holds radial and axial load induced by wind thrust force. The second main bearing has a crucial role in reducing the axial loads entering the gearbox and as this bearing degrades, the life of bearings inside the gearbox decreases significantly.

A series of damage and degradation levels for the second main bearing was considered and the vibration level of this bearing as well as the life of another bearing inside the gearbox was studied. It was found that in some cases the vibration level of the damaged main bearing is still in an acceptable range while the bearing inside the gearbox experiences a significant life reduction. This fact highlights that the health monitoring of some critical components which have significant effect on other components inside the gearbox, should not be stand alone. The critical component itself may still be in an acceptable condition but it may have already affected other components in the system. Therefore, a comprehensive health monitoring system should account for effects between the components and this task can only be carried out if a reliable model of the system exists.

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