Fatigue life prediction of wind turbine main bearing considering internal clearances and pounding forces

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1. Introduction

A three dimensional drive train model has been proposed by Ishihara et al. [1] to predict the dynamic behavior of three point mount wind turbines [2]. The main bearing in this model is assumed as a linear spring. However, the pounding phenomenon was observed at the wind turbine main bearings [2], while this phenomenon could not be explained by the current drive train model. In the bearing industry, empirical load factors were applied to machines exposed to the pounding and large vibration under their operation environments [3]. Takeda et al. [4] applied a pounding model with finite element method (FEM) by using a nonlinear spring to calculate the pounding force on the bridge. The pounding model has not been applied for the wind turbine main bearing. Fatigue life of main bearing can be predicted as the fatigue loads are obtained by the drive train model. For wind turbine main bearings, the formulas shown in ISO 281 [5] are applied to predict the bearing rating life L_{10} with 90% reliability, however, it cannot explain the high failure rate of the wind turbine main bearings in Tomamae Wind Farm. This implies that the new parameters should be included in the formulas in ISO 281 .

In this study, fatigue prediction of wind turbine main bearing is conducted considering internal clearances and pounding forces. Firstly, a pounding model is introduced to simulate the bearing pounding forces and evaluate the load factor. The bearing operating parameters are then determined by the onsite survey. Finally, the main bearing rating life L_{10} is predicted considering the new parameters and is compared with the onsite records.

2. Numerical models

The pounding model of main bearing and theory of bearing fatigue prediction are described in this section.

* December 5, 2019 41stWind Energy Symposium

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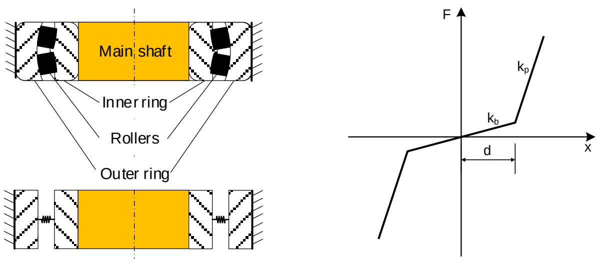
2.1 Pounding model of main bearing

The analysis on pounding phenomenon developed in bridge and structural engineering is considered to apply for wind turbine main bearing in this study. A simplified main bearing model is adopted instead of high fidelity bearing model with detail components, such as rollers and rings, since this research concentrates on investigating the pounding phenomenon of main bearing,

The main bearing is modeled by a pounding spring considering the nonlinearity of internal clearance as shown in Fig. 1 (a). The main shaft and inner ring are connected rigidly and modelled as a lumped mass, while the outer ring is connected to a rigid bearing housing, which is fixed on the nacelle bedplate. Rollers are modeled as one spring for each direction and work in different ways for the cases with and without internal clearances. The pounding spring for rollers are illustrated in Fig. 1 (b), in which k_b , k_p and d, are the initial bearing stiffness, the pounding stiffness, and the internal clearance, respectively.



(a) Main bearing model (b) Pounding spring

Fig. 1 Modelling of main bearing

The governing equation of main bearing model is shown in Equation (1). When the relative motion between inner and outer rings is within the clearance, the pounding force does not occur, while the pounding force occurs when the relative motion exceeds the clearance. Through calculations on restoring forces of the bearing occurred between those two conditions, load factor due to pounding can be estimated.

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}+\{F_p(t)\}=\{F(t)\}-[M]\{\ddot{u}_n\} \quad (1)$$

$$F_p(t)=\begin{cases}0 & -d \le u \le d \\ k_p(u-d) & u>d \\ k_p(u+d) & u<-d\end{cases}$$

where $\{u\}$, $\{\dot{u}\}$ and $\{\ddot{u}\}$ are the displacement, velocity, and acceleration vectors of each node in 6 degree of freedom (DOF), $\{\ddot{u}_n\}$ is the acceleration vector of nacelle bedplate, and $\{F(t)\}$ is the wind loads vector. $[M]$, $[C]$, and $[K]$ are the global mass, damping and stiffness matrixes, $F_p(t)$is the pounding force, u is the relative motion between inner and outer rings.

2.2 Theory of bearing fatigue prediction

In this study, the fatigue life of main bearing is calculated with Equation (2) by employing an additional parameter of life ratio a_ε and the load factor f_w to count the pounding effect into the equivalent dynamic load P'. Those parameters are not considered in the current ISO 281 formula.

$$L_{10}=a_{ISO}a_\varepsilon\left(\frac{C}{P'}\right)^{\frac{10}{3}}, \quad P'=f_wP \quad (2)$$

The load factor is defined as the ratio of bearing load with pounding to without pounding as shown in Equation (3).

$$f_w=F'/F_0 \quad (3)$$

where F_0 and $F'$are the bearing loads without and with pounding, respectively. Considering fatigue evaluation for fluctuating wind speeds, an equivalent load factor f_{w_equ} is introduced with weights of wind speed distribution as shown in Equation (4).

$$f_{w\_equ}=\sqrt[10/3]{\frac{\sum_i q_i(P_i f_{wi})^{10/3}}{\sum_i q_i(P_i)^{10/3}}} \quad (4)$$

where q_i is the probability and f_{wi} is the load factor at the wind speed i.

In addition, the internal clearance causes the load concentration on specific rollers and rings, and subsequently reduces the fatigue life of bearing. A life ratio a_ε is introduced to consider this life reduction, and is defined as the ratio of rating life with to without internal clearance [3]. The relationship between life ratio a_ε and internal clearance is shown in Fig. 2.

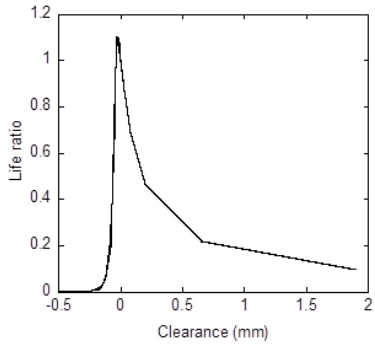


Fig. 2 Relationship between the life ratio and the internal clearance

3. Results and discussions

In this section, the load factor is calculated with the proposed pounding model for the main bearing. The contaminations and internal clearances are selected based on a survey of Tomamae Wind Farm. Finally, the rating life L_{10} of main bearing is predicted and validated by the onsite records.

3.1 Evaluation of load factor

The torque arm behaviors mounted on three point mount wind turbine were analyzed in previous research [2], the horizontal motions in the Y direction at two sides are same, while the motions in other directions are different. In order to clarify the effect of main shaft motions on the main bearing, the pounding effect on the main bearing is evaluated in the horizontal direction Y only. The stiffness of torque arm is considered as linear for simplification of simulation. The linear stiffness of torque arm is determined by the measured displacement and the corresponding stiffness [2]. The initial bearing stiffness k_b is evaluated based on the previous research [2].

At the normal operation with a normal internal clearance, the roller stiffness is converted into the initial bearing stiffness k_b and the pounding stiffness k_p. Nakamura et al. [6] showed that the ratio of those two stiffness is around 0.1 and used in this study. The normal internal clearance at top of main bearing in the vertical direction Z is decided by a clearance as shown in ISO 5753 [7]. In consideration of geometrical relationship of main bearing, clearance in the horizontal direction Y is approximately estimated as half of the vertical clearance. In the vertical direction Z, the normal clearance ranges from 0 to 0.44 mm, a middle value of 0.22 mm is selected to evaluate the load factor, and the horizontal clearance is 0.11 mm in this case. The spring parameters with and without pounding are summarized in Table 1.

Table 1 Parameters for different internal clearances

Cases	k_b	k_p	d
W/O pounding (mm)	150	0	0
With pounding (mm)	15	150	0.11

An example of time series of predicted bearing loads with and without pounding at the wind speed of 8 m/s is shown in Fig. 3. It is found that the pounding forces have large fluctuations due to the collision between rollers and rings. Considering the uncertainty in simulations, 6 turbulence seeds have been used for each wind speed.

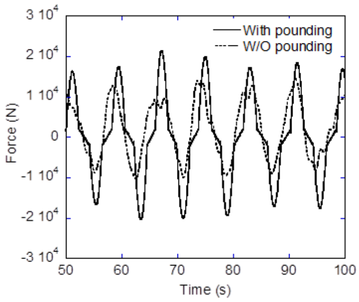


Fig. 3 Comparison of bearing forces with and without pounding.

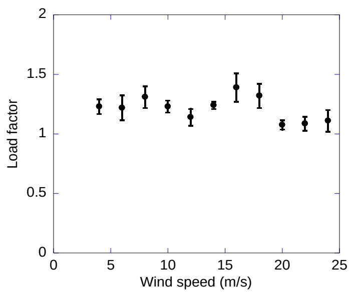


Fig. 4 Variation of load factor with mean wind speeds

The variation of predicted load factors with mean wind speeds is shown in Fig. 4. The plots and bars in Fig.4 mean the average and standard deviation of load factors for 6 seeds respectively. It is found that the predicted load factors range from 1.1 to 1.6, which are comparable to the empirical load factors 1.2 to 1.5 for the slight pounding category used in the current bearing industry [3]. Finally, the equivalent load factor is calculated as 1.3 and used for prediction o f bearing fatigue life.

3.2 Fatigue prediction of main bearing

Besides the load factor of main bearing, the contamination and internal clearance also strongly influence the bearing fatigue life. These parameters are investigated on several wind turbine main bearings in this study, and their relationship is shown in Fig. 5.

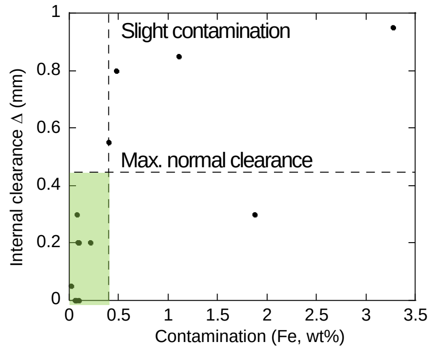


Fig. 5 Relationship of contamination and internal clearance of main bearings

Based on the actual maintenance criteria, the particle concentration (iron) of bearing during the normal operating condition is smaller than 0.4 wt%. Within this criteria, all observed internal clearances are smaller than the maximum clearance of 0.44 mm as shown in ISO 5753 [7]. This means that the bearing operates normally under the slight contamination and small clearance conditions as shown in the green colored area in Fig. 5. Consequently, the maximum value of 0.44 mm is taken as the normal internal clearance to predict the life ratio a_ε. The additional parameters for prediction of main bearing rating life are summarized in Table 2. Finally, the rating life L_{10} is obtained with these parameters.

Table 2. The additional parameters for prediction of main bearing rating life

Case	Case with a_ε	Case with a_ε and f_w
a_ε	0.34	0.34
f_w	1	1.3
a_{ISO}	Typical contamination	

The observed rating life L_{10} for Tomamae Wind Farm is derived from the onsite damage records of main bearing from 1999 to 2017. Duration from the commencement of operation to the first inspection point is regarded as a service life. Because some operations of main bearing were interrupted with unexpected accidents or unscheduled maintenance, a censoring technique is applied for these abnormal fatigue data. It is known that the bearing fatigue life follows a Weibull distribution based on the experimental and numerical studies on the rolling contact fatigue as expressed in Equation (5). In this study, accumulative failure rates are obtained from damage records of main bearing, and the relationship between the reliability and the service life is shown in Fig. 6. Finally, the observed main bearing rating life L_{10} is calculated as 9 years by taking reliability S of 90%.

$$\ln\frac{1}{S} = AL_s^e \tag{5}$$

where S is the reliability, A is a constant as a material factor, L_s is the service life, e is the Weibull slope.

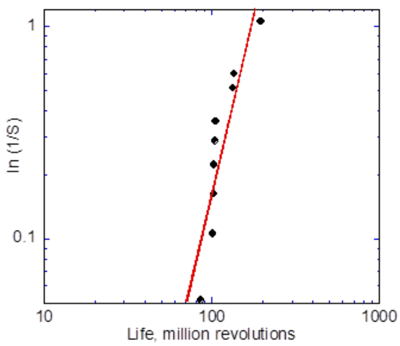


Fig. 6 Weibull distribution of fatigue lifes

The predicted rating life L_{10} are compared with the observed value as shown in Fig. 7 by using different parameters listed in Table 2.

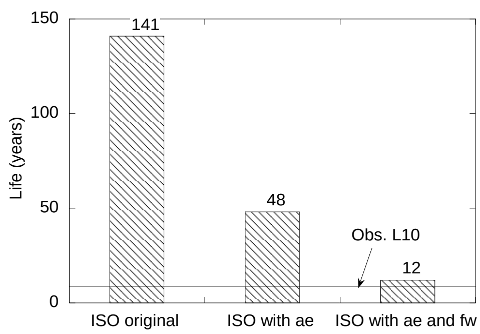


Fig. 7 Predicted rating life L_{10}

It is found that the original formula in ISO 281 overestimates rating life about 18 times. Only considering life ratio a_ε, the predicted rating life is 48 years, which is about 7 times longer than the observed value. When considering both parameters of life ratio a_ε and load factor f_w, the predicted rating life matches well with the observed value.

4. Conclusion

In this study, the fatigue prediction of wind turbine main bearing is performed in consideration of internal clearances and pounding forces. The load factors are evaluated by a pounding model of main bearing. The formula in ISO 281 to predict the bearing rating life L_{10} is modified by introducing new parameters of life ratio and load factor. Following conclusions are obtained:

1) A numerical model for main bearing pounding simulation is conducted. The load factor due to pounding is proposed for the main bearing fatigue life prediction.
2) The parameters of life ratio and load factor are applied to predict main bearing fatigue life. The predicted rating life L_{10} matches well with the onsite records, while that by the formula in ISO 281 is significantly overestimated.

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