European Wind Energy Association Conference (EWEA 2011)

Fatigue Analysis of Wind Turbine Gearbox Bearings using SCADA Data and Miner's Rule

Hui Long*, **Jingze Wu, Fiona Matthew and Peter Tavner**

Durham University, UK

*Email: h.long@sheffield.ac.uk

Abstract

This paper presents studies of wind turbine load estimation, based upon SCADA data, to derive bearing load histograms for the high speed shaft (HSS) of a typical, three-stage, wind turbine gearbox. Rain-flow counting has been applied to determine the number of loading occurrences at different load values. Damage ratios for bearings have been predicted using this data by the application of Miner's rule. Results show that the bearings have an overload Cycle Ratio of less than 20% above rated and overloads of Load Ratio of 1.0~1.1 incurring significant damage to the bearings.

Keywords: Wind turbine; Gearbox; SCADA data analysis; Bearing; Fatigue analysis.

1 Introduction

The gearbox is one of the important subassemblies within a wind turbine (WT), in respect of downtime. High bearing failures have been observed in modern WT gearboxes. The premature failure of gearbox bearings causes unplanned WT shutdowns and early bearing replacements, reducing WT availability and increasing the cost of energy (CoE) [1-3]. WT gearbox bearing failure mechanisms have been under investigation in recent years [4-5] and are yet to be fully understood. As the wind industry moves from onshore to offshore, the development of a better understanding of bearing failure mechanisms and new bearing design and analysis methods will be important to ensure low CoE for offshore wind energy.

WT gearboxes are subjected to a wider range of complex and unpredictable load conditions than gearboxes in other, for example marine or automotive, industrial applications. The need for an accurate representation of loads is important to ensure that WT designs can withstand all loading cases and endure for the required 20 year life span. Due to the stochastic nature of wind, a WT gearbox experiences a wide loading variation during its service life, which is difficult to predict. This variation results in premature failures of various forms both at the component and subsystem levels.

Operational control of the WT is intended to maximise wind energy extraction and minimize turbine damage under extreme wind conditions however it complicates the loading condition. Varying wind resources at the WT site and differences in gearbox designs can also contribute to load variation. Currently bearing fatigue life calculation design does not consider real load variations or cyclic loading because field data is unavailable during the WT design stage. Therefore at this stage load variation considerations are mainly based upon computational simulations [6].

2 SCADA Data Analysis

Supervisory Control & Data Acquisition (SCADA) data contain WT operational, performance and maintenance information recorded at wind farms (WF). A variety of parameters are recorded, ranging from wind speed, rotational rotor speed, bearing temperature to power production. The data are collected and averaged over 10 minute periods and could give an indication of possible abnormal conditions. This information could be useful to maximize WT availability and optimise power generation. Recorded SCADA field data could be used to analyse shaft torque and rotational speed time histories to extract load conditions for evaluating gearbox bearing performance and predicting fatigue life.

In this paper the characteristics of the wind speed distribution and its effect on gearbox loading variations are considered in order to derive the loading histograms applied to gearbox shafts. By investigating wind speed, rotor speed and power output histories the relationship between these three parameters can be established. Therefore, cyclic loading conditions, caused by wind speed variation and WT operational control, can be considered for analyzing gearbox bearing performance.

The relationship between rotor and wind speeds can be determined by a weighted average method. The weighted average rotor speed, $\omega_{R,Ave}$, for each wind speed bin or increment value can be calculated using:

$$\omega_{R,Ave} = \frac{\sum \omega_{R,i} M_{ij}}{\sum M_{ij}} \quad (1)$$

$\omega_{R,i}$ is the rotor speed at i^{th} value (rev/min);

M_{ij} is the number of occurrences of the rotor speed at the i^{th} value when wind speed (m/s) is at the j^{th} value.

Using the gearbox configuration design data, such as transmission ratio, gear and bearing parameters, shaft torque values can be determined from output power values. These can then enable the calculation of gear and bearing loads.

To differentiate and break-down the dense and variable torque-time history amplitudes into a more coherent form, the rain-flow counting method [7] has been used to process the SCADA data. The counted numbers of occurrences of load at various values are then used to estimate the cyclic loading applied to bearings.

3 Miner's Rule

Miner's rule assumes that fatigue failure is predicted when the sum of the damages equates to 1 [8]. This can be defined as:

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \cdots \frac{n_i}{N_i} = 1 \quad (2)$$

n_i = No. of cycles at i^{th} load value;

N_i = No. of cycles to failure at i^{th} load value;

$\frac{n_i}{N_i}$ = Damage ratio at the i^{th} load value.

If the fraction of cycles at each loading value is known as percentage rather than actual cycle number, the number of cycles can be expressed as:

$$n_i = \alpha_i N \quad (3)$$

α_i: Cycle Ratio (fraction of cycles) at the i^{th} load value;

N: Resultant fatigue life (total cycles).

Thus the resultant fatigue life can be expressed as:

$$N = \frac{1}{\frac{\alpha_1}{N_1} + \frac{\alpha_2}{N_2} + \frac{\alpha_3}{N_3} + \ldots + \frac{\alpha_i}{N_i}} \quad (4)$$

In this study, the Cycle Ratio was obtained from the loading spectrum by analysing SCADA data. It was calculated by the number of occurrences at each loading value divided by the sum of occurrences. The procedure can be expressed by the following equation:

$$\alpha_i = \frac{n_i'}{\sum n_i'} \quad (5)$$

n_i' = No. of occurrences at the i^{th} load value;

$\sum n_i'$: the total number of occurrences at all loading values.

4 Loading Analysis of HSS Bearings

This study considered a typical three-stage gearbox design consisting of one planetary and two parallel gear stages. An example of the gearbox layout is illustrated in Figure 1 [3]. For this configuration, failures have been observed in the high-speed shaft (HSS) bearings, intermediate-speed shaft (ISS) bearings and the planet bearings [3]. Based on an existing 1.7 MW WT gearbox design, bearing loads can be calculated via gear loads and distances between gears and bearings. The load applied to gears can

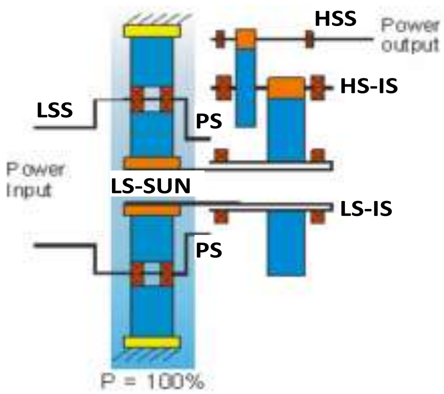


Fig. 1 Typical 3-stage gearbox layout [3]

be calculated from the torque, which can be determined from the power transmitted to the output shaft. To illustrate the method, this paper only presents calculations for bearings mounted on the gearbox HSS. Figure 2 illustrates the arrangement of bearings of the HSS in a typical three-stage gearbox design.

5 Results and Discussion

The SCADA data used in this study were recorded over a period of 29 months on one WT without gearbox problem. Figure 3 shows the WT wind distribution recorded at 10 minute intervals. It is obvious that for the duration of the recorded period the wind speed remained mainly between 3 to 21 m/s with only small numbers of occurrences of wind speed below 3 m/s or above 21 m/s. The distribution roughly followed a Weibull distribution.

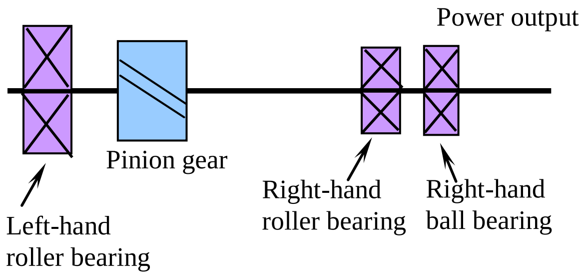


Fig. 2 Arrangement of bearings on HSS

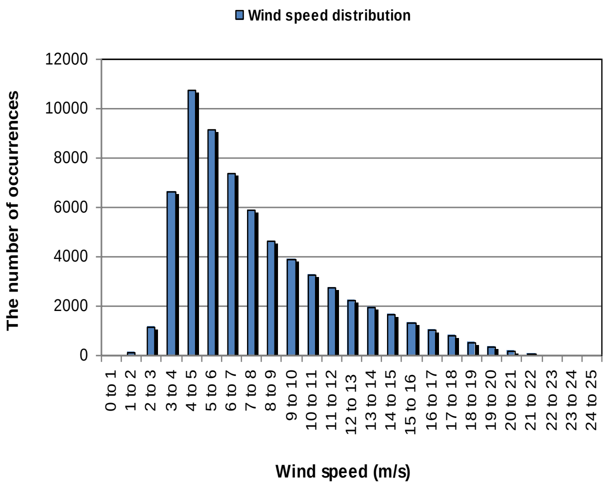


Fig. 3 SCADA wind speed distribution

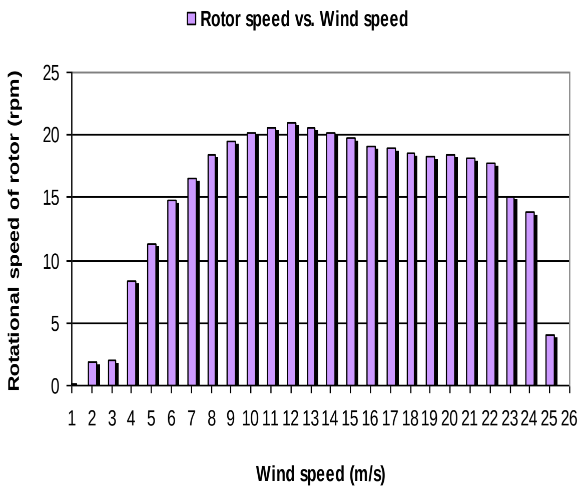


Fig. 4 SCADA rotor speed vs. wind speed

By calculating the weighted average of the rotor speed at different wind speed values, the relationship between rotor and wind speeds can be generated, as shown in Figure 4. The WT cuts in at a wind speed of 2-3 m/s and from cut-in to rated wind speed of 10 m/s the average rotor speed increases with wind speed, under generator control. Above 10 m/s the pitch control starts to operate and the rotor speed remains close to its nominal top speed of 19 rev/min. When the wind speed increases above 24 m/s the turbine cuts-out, causing the rotor speed to reduce rapidly.

By analysing the measured output power and rotor shaft speed data, the variable amplitude torque-time history can be obtained. Using gearbox design parameters, these torque values can then be used to calculate loads exerted on the gears, which in turn are then transmitted to the gearbox bearings.
The rain-flow counting method was used to generate the loading histogram defined as a relationship between the Load Ratio and the Cycle Ratio, calculated from Equation 5. The Load Ratio is defined as the actual load divided by the load at the nominal torque value of the shaft [8]. The Cycle Ratio has been obtained by counting the number of occurrences at each bearing load value and dividing them by the total number of occurrences. By applying the Miner's rule, bearing cyclic loading damage at various Load and Cycle Ratio values have been predicted. The calculation of the basic bearing rating life is based upon a 90% statistical model [9].

Figures 5 and 6 show the load histograms of the gearbox HSS left- and right-hand side roller bearings, respectively. For both left- and right-hand side bearings the majority loading is below the rated loading, however approximately 20% of the Cycle Ratio range is above the rated value. The overloading ratio of the right-hand side (RHS) bearing is much higher than that of the left-hand side (LHS) bearing, although the absolute loading value for the HSS LHS bearing is much higher than that of the RHS bearing, which is compensated for by selecting a bigger size of the LHS bearing.

By analysing the torque-time spectrum and calculations of bearing loads on the HSS, it can be shown that the bearing loading varies significantly with the varying rotational speed of the shaft. To calculate the fatigue rating life of bearings at each load value, the relationship between bearing load and rotational speed has been derived by the trend line of loading distribution. Figure 7 shows an example of the load distribution and the relationship between load and rotational speed on the RHS roller bearing of HSS. Figures 8 and 9 show the damage estimations for the HSS LHS and RHS bearings, respectively. The damage ratio distributions show that overloads around Load Ratio of 1.0~1.1 can be very damaging when they are present over the short cycle time of less than 20%. By comparing damage ratios of the LHS and RHS bearings, it is clear that the overloading around Load Ratio of 1.0~1.1 are more damaging to the HSS RHS bearing than to the LHS bearing.

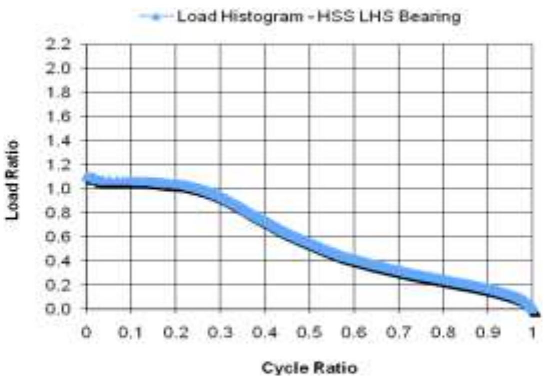


Fig. 5 Load histogram of HSS LHS bearing

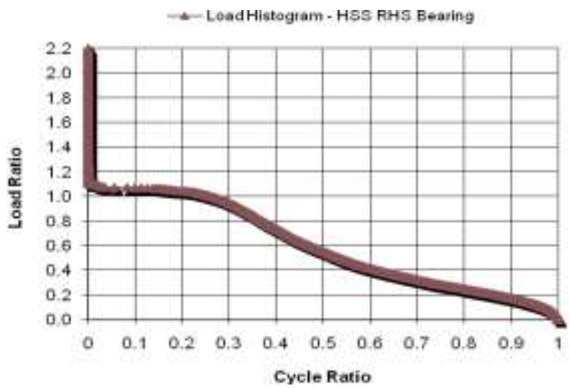


Fig. 6 Load histogram of HSS RHS bearing

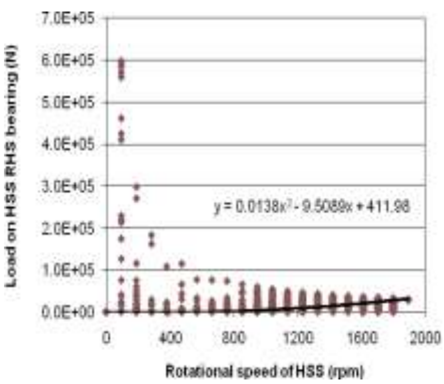
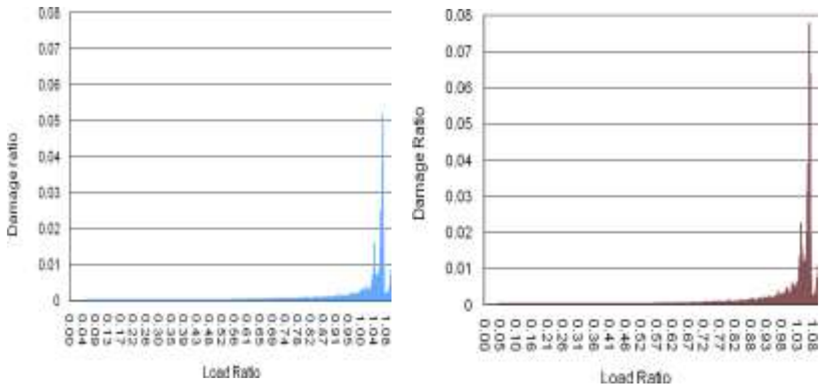


Fig. 7 Load of HSS RHS bearing



Figs. 8 & 9 Damage estimation of HSS LHS & RHS bearings

6. Conclusions

SCADA data analysis enables WT gearbox load estimations to be made by utilising measured field data. The result obtained from the load histogram of gearbox HSS bearings and analysis of bearing fatigue damage ratios draws the following conclusions:

The majority of HSS bearing loading values are below rated load; however there is a noticeable overload Cycle Ratio of less than 20%.Overloads around Load Ratio of 1.0~1.1 cause significantly higher bearing damage when they are present for short periods of time (less than 20% of Cycle Ratio range). The damage distributions show that the overloads are more damaging to the HSS RHS than the LHS bearing, for this specific gearbox bearing arrangement.

Acknowledgement: This work has been funded by the EU FP7 Project RELIAWIND 212966.

References:

[1] W. Musial, S. Butterfield and B. McNiff. "Improving Wind Turbine Gearbox Reliability". 2007 EWEC, Milan, Italy, May 2007.

[2] Wilkinson, M et al (2010), "Methodology and Results of the Reliawind Reliability Field Study", 2010 EWEC Scientific Track, Warsaw, Poland, April 2010.

[3] K. Smolders, H. Long, Y. Feng and P. Tavner. "Reliability Analysis and Prediction for Wind Turbine Gearboxes". EWEC 2010 Scientific Track, Warsaw, Poland, April 2010.

[4] A. Crowther and N. Zaidi. Advanced Analysis Approaches for Wind Turbine Drive train. NREL 2009 Wind Turbine Reliability Workshop, Albuquerque, USA, June 2009.

[5] F. Oyague, C. P. Butterfield and S. Sheng. "Gearbox Reliability Collaborative Analysis Round Robin". The American Wind Energy Association WINDPOWER 2009 Conference, Chicago, USA, May 2009.

[6] P. H. Madsen, K. Pierce and M. Buhl. "Predicting Ultimate Loads for Wind Turbine Design". AIAA/ASME Wind Energy Symposium, USA, Jan 1999.

[7] R.I. Stephens et al., "Metal Fatigue in Engineering", 2nd edition, John Wiley & Sons 2001.

[8] Donald R. McVittie, Robert L. Errichello. "Application of Miner's Rule to Industrial Gear Drives". American Gear Manufacturers Association. Technical paper 88 FTM 9. 1988.

[9] International Standard: BS ISO 281:2007 Rolling Bearings - Dynamic Load Rating and Rating Life.