Life estimation method for a wind turbine main shaft bearing

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Abstract

Due to the increased size and complex loading conditions of offshore wind turbines, the demand from industry for an accurate life estimation method for wind turbine components is increased. In addition to this demand from industry, ORE Catapult's primary aim for developing a life estimation method for a wind turbine main shaft bearing is to study the effects wind fields have on the dynamic loading. The process starts by evaluating the main shaft loads for a given wind field.
As a result of the complexity of a typical wind turbine drivetrain, each major component is analysed separately and the main shaft bearing is further studied in detail. Using standard industrial formulations, the bearing life is calculated for specific wind conditions and drivetrain events.
Here, the dynamic loading on the bearing components is evaluated through the use of a rigid dynamics model utilising a finite element analysis (FEA) method. As a result, the effects from different wind fields on main shaft bearing life are analysed and identified.

1. Introduction

The primary function of the wind turbine main bearing is to support the low-speed shaft and react non-torque loads generated by the rotor. As such, it plays a vital role in ensuring the reliability of all drivetrain components. Main bearing failures are known to cause secondary failure to the gearbox and other drivetrain components[(1)]. Additionally the process of repairing or replacing main bearings tend to be time consuming and very expensive, particularly for offshore turbines. Consequently, the need to understand effects of main shaft loads on the main bearing is clearly very important.

A popular bearing arrangement for wind turbines in the low megawatt class is the 2 point main shaft support arrangement. Common arrangements of this type make use of spherical roller bearings due to their ability to tolerate misalignment. Such a bearing arrangement is assumed for a 1.5 MW machine. The bearing arrangement utilises an SKF 230/630 CAW/30 spherical roller bearing as a primary bearing and SKF 23188 CAW/33 spherical roller bearing for the secondary position.

Section 2 presents the theoretical background to how bearing life is calculated . A Rigid Dynamics (RD) model is presented in section 3. The section describes how the model is contructed such that the load distribution in the roller bearing can be analysed. The section also includes how a Finite Element (FE) model is used to evaluate stiffness characteristics and comparision of model behaviour to theory. Section four demonstrates how the utilisation of the rigid dynamics model will benefit the process of evaluating bearing life.

2. Bearing Life calculation

Bearing life is typically specified by the probability of failure. Among these, the life at which 90% of bearings are expected to have failed (L_{10}) is the most common.

2.1 Standard life calculation

Current industrial practices advocate the use of ISO 281:2007 to calculate bearing life. The basis of the ISO standard is derived from work done by Lundberg and Palmgren in the 1940s. The life rating stated within ISO 281:2007 is given by

$$L_{nm} = a_1 a_{iso} L_{10} \qquad \text{.....................................(1)}$$

In which a_1 is a modification factor for reliability (taken as 0.37 for 98% reliability) and a_{iso} is an iso modification factor. L_{10} is the bearing life as calculated by Lundberg and Palmgren. For a roller bearing this is given by

$$L_{10} = \left(\frac{C_r}{P_r}\right)^p \qquad \text{..(2)}$$

In the Lundberg Palmgren L_{10} calculation the life of a bearing depends on the basic dynamic load rating (C_r), dynamic equivalent load rating (P_r) and p is the life equation exponent which takes the value of 3 for ball bearings and 10/3 for roller bearings. C_r characterises the strength of the specific bearing and thus remains constant. C_r may be obtained from bearing catalogues or from the equation.

$$C_r = b_m f_c (i\, L_{we} \cos\alpha)^{7/9} Z^{3/4} D_{we}^{29/27} \qquad \text{.............................(3)}$$

where,

b_m	-	rating factor for bearing steels
L_{we}	-	effective roller length
α	-	contact angle
Z	-	number of rolling elements
D_{we}	-	effective roller diameter
i	-	number of rows
f_c	-	factor which depends on bearing geometry. Values for f_c can be read from Table 7 in BS ISO 281:2007

For the analysis undertaken, a basic load rating of 6700 kN is used, as specified by the bearing manufacturer. Alternatively, the life of a particular raceway may be evaluated

$$C_{i,o} = 552\, \lambda \frac{(1 \mp \gamma)^{29/27}}{(1 \pm \gamma)^{1/4}} \gamma^{7/9} Z^{-1/4} D_{we}^{29/27} {L_{we}}^{7/9} \qquad \text{..........................(4)}$$

where,

γ	-	$\frac{D_{we}\ \cos\alpha}{d_m}$
α	-	nominal contact angle
λ	-	factor to account for edge loading

The load on the bearing is characterised by the equivalent dynamic load rating. This depends on the load ($|Q_j|$) on rolling element j and number of rolling elements (Z).

$$P_r = \left(\frac{1}{Z}\sum_{j=1}^{j=Z} Q_j^4\right)^{1/4} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(5)$$

This approach requires that the load distribution across the rolling elements is known. Palmgren suggested a simplified formulation which is commonly used now in many applications, including ISO 281:2007 life calculation.

$$P_r = XF_r + YF_a \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(6)$$

The coefficients X and Y are found from bearing manufacturers. For the SKF 230/630 CAW/30 spherical roller bearing the equivalent dynamic load rating is given by

$$P_r = F_r + Y_1 F_a \qquad F_a/F_r \leq e \ldots\ldots\ldots\ldots\ldots\ldots\ldots(7)$$

$$P_r = 0.67F_r + Y_2 F_a \qquad F_a/F_r \leq e \ldots\ldots\ldots\ldots\ldots\ldots\ldots(8)$$

The general procedure for evaluating bearing life is centred on a constant load and speed of operation. However, the loading on wind turbines is highly dynamic and variable. To calculate the life as specified above, it is necessary to convert the real operational loads and speeds to a representative load and speed which causes the same damage to the bearing. The representative force F_m and rotational speed n_m are evaluated as follows:

$$F_m = \sqrt[p]{\frac{\sum_{i=1}^{n} F_i{}^{p} \cdot t_i \cdot n_i}{100 \cdot n_m}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(9)$$

$$n_m = \frac{\sum_{i=1}^{n} t_i \cdot n_i}{100} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(10)$$

Where,

$$\sum_{i=1}^{n} t_i = 1$$

and

t_i	-	fraction of time when load is F_i
n_i	-	rotational speed when load is F_i
p	-	life equation exponent

To improve the accuracy of life prediction; Equation **5** above is used opposed to the common Equation **6**. This requires the load distribution in the bearing to be analysed.

3. Bearing Load distribution

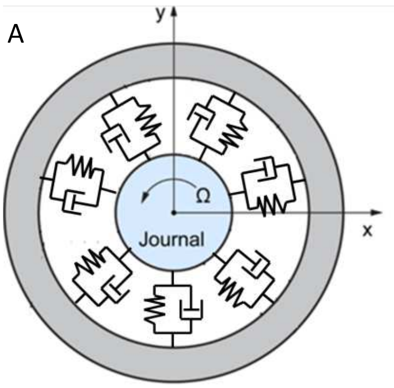
Theoretically, the load carried by rolling element ψ_i in row i in radial location ψ of a static roller bearing is given as a proportion of the maximum possible rolling element load for the applied radial force. The relation is given by Equation 11. Given the dynamic behaviour of the loads on the bearing, a quasi-static approach which utilises Equation 11 may not be fully representative. Consequently, an RD model of the bearing is generated to perform dynamic analysis.

$$Q_{\psi i} = Q_{\max i}\left[1 - \frac{1}{2\epsilon_i}(1-\cos\psi)\right]^{1.11} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots..(11)$$

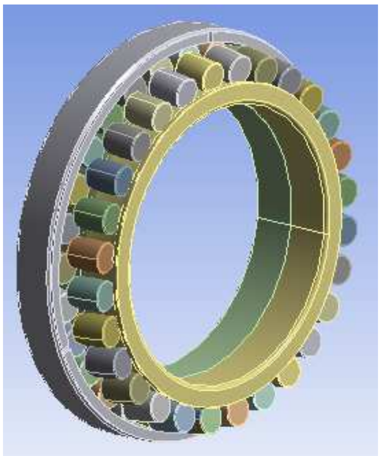
3.1 Dynamic Bearing modeling

A typical approach to evaluating the load distribution is through the use of FE models. These are generally computationally expensive because all the components of the bearing are discretised into small elements, which are then analysed with each body having the relevant degrees of freedoms. To perform a dynamic analysis spanning several seconds or minutes, the use of FE modelling for complex systems requires huge computational resource. As opposed to FE, RD packages represent each component as a rigid body; thus limiting the degree of freedom to a maximum of 6 (3 translational, 3 rotational) per body. The computational requirement is therefore reduced, enabling dynamic analyses over longer periods of time.

A RD bearing model of the SKF 230/630 CAW/30 was created within Ansys Rigid Dynamics.



Bearing Model Schematic



RD model of SK230/630 CAW/30

Figure 0-1 Software representation of bearing

The contact between the inner race and rolling element, and between outer race and rolling element are defined by revolute joints. The rolling elements are defined as nonlinear bushing joints between the contact point on the inner race and contact point on the outer race as shown in Figure 2. The bearing is assumed to have a perfectly rigid and smooth cage, therefore the spacing between rolling element remains constant and friction between cage and rolling element is ignored. Also, given the relatively slow speed of bearing rotation, it is assumed that slipping does not occur between the rolling element and the race way. With this relationship imposed, the cage speed of the rolling element is given by

$$n_m = \frac{1}{2} n_i (1 - \gamma) \quad \text{...(12)}$$

Where,

n_m	-	cage rotational speed in rpm
n_i	-	inner race rotational speed
γ	-	$\frac{D_{we} \cos \alpha}{d_m}$
α	-	nominal contact angle
d_m	-	diameter
D_{we}	-	effective rolling element diameter.

Radial and axial loads are applied at the centre of the bearing as shown in Figure 0-2. To ease the extraction of data for individual rows, the raceways are sliced in two but joined together with a fixed joint .

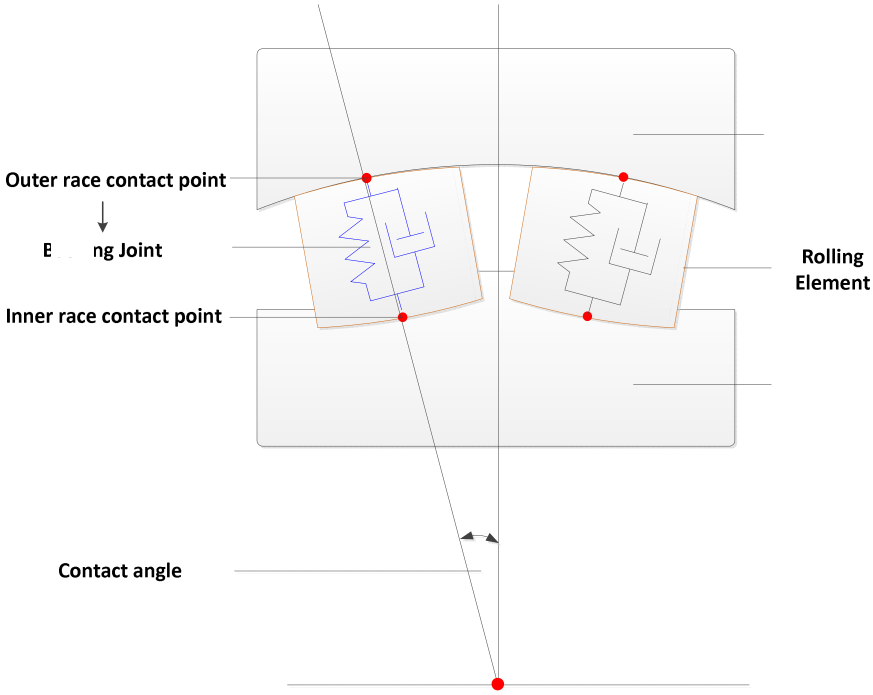


Figure 0-2: Rigid dynamic model representation

As explained above, the rolling element is represented by a nonlinear bushing joint which simulates the rolling element by a spring and damper. The stiffness characteristics of the contact must encapsulate the stiffness characteristics of the actual roller raceway interaction. As postulated by Harris[2], the relationship between the load (Q) carried by the rolling element is related to deflection (δ) by:

$$\delta \sim Q^{9/10}$$

The load-deflection relationship can be characterised by a stiffness which is nonlinear as shown by Equation 13. The total stiffness of the rolling element (k_{total}) is given as the sum of the inner race contact (k_{IR}) and outer race contact (k_{OR}) in series.

$$\frac{1}{k_{total}} = \frac{1}{k_{IR}} + \frac{1}{k_{OR}} \quad \text{...(13)}$$

The nonlinear rolling element stiffness is defined as a compression only spring, therefore it produces no reaction foce in tension. Figure 0-2 shows the stiffness characteristics.

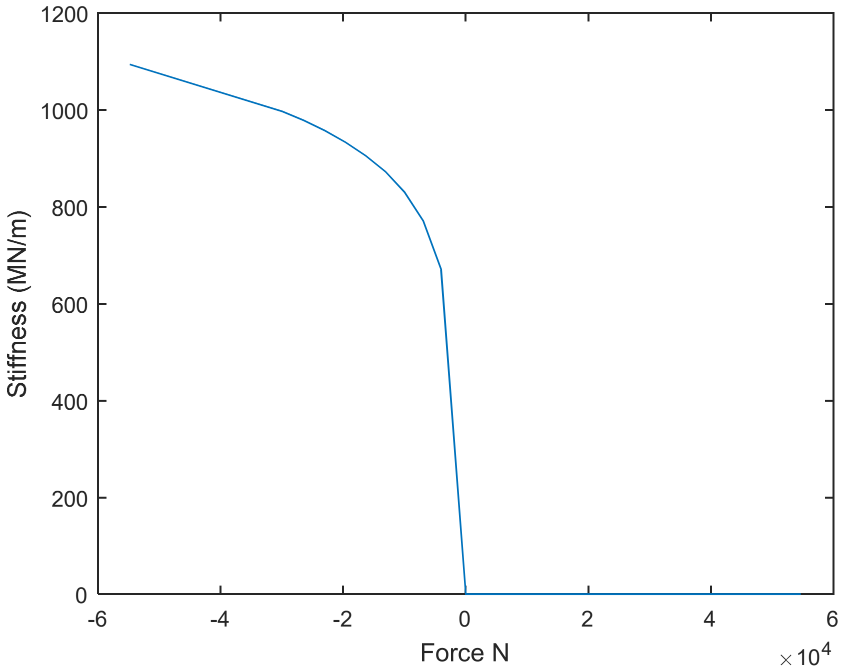


Figure 0-3: Non linear stiffness characterisitics

Hertzian theory is the most popular approach for evaluating the contact characteristics of two elastic bodies. The theory which was initially proposed by Hertz is based on a few assumptions of the contacting bodies. Later Lunderg and Sjovall deduced an equation for the mutual approach of two elastic bodies in line contact[2]:

$$\delta = \frac{2Q(1-\xi^2)}{\pi E l} \ln\left[\frac{\pi E l^2}{Q(1-\xi^2)(1\mp\gamma)}\right] \quad \text{.....................................(14)}$$

In Equation 14, ξ is the Poisson's ratio of the bearing material, E is the elastic modulus of bearing material, l is the effective length of rolling element, Q is the load carried by the rolling element. Equation 14 remains true for an ideal line contact, however most roller bearings are crowned to reduce edge loading. Also, the geometry of the rolling element may lead to deviation from an ideal line contact. To accurately determine the stiffness characteristics of the rolling element, an FE model of the rolling contacts was generated.

3.2 FE model of rolling contacts

The conformal stiffness behaviour of the roller-outer race contact and non-conformal behaviour of the roller-inner race contact were analysed separately to save on computational cost. To further reduce computational cost, a quadrant of the rolling element is modelled with a section of the race way it is in contact with. Figure 0-4 shows the model for the roller-inner race contact while Figure 0-55 shows the roller-outer race contact. As seen from the two FE models, edge loading is present regardless of the crowning of the rolling elements, this is more evident in the contact between the rolling element and the inner raceway (Figure 0-4).

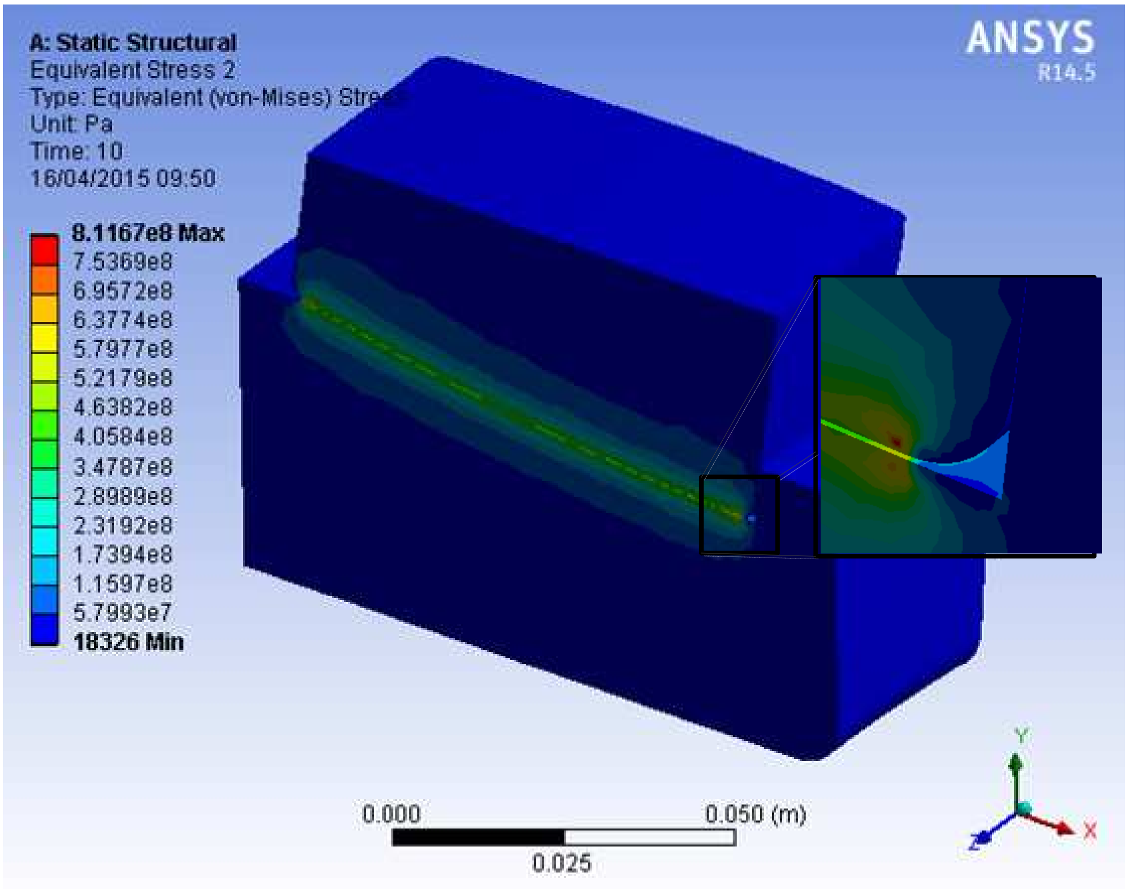


Figure 0-4 FE model of inner race contact

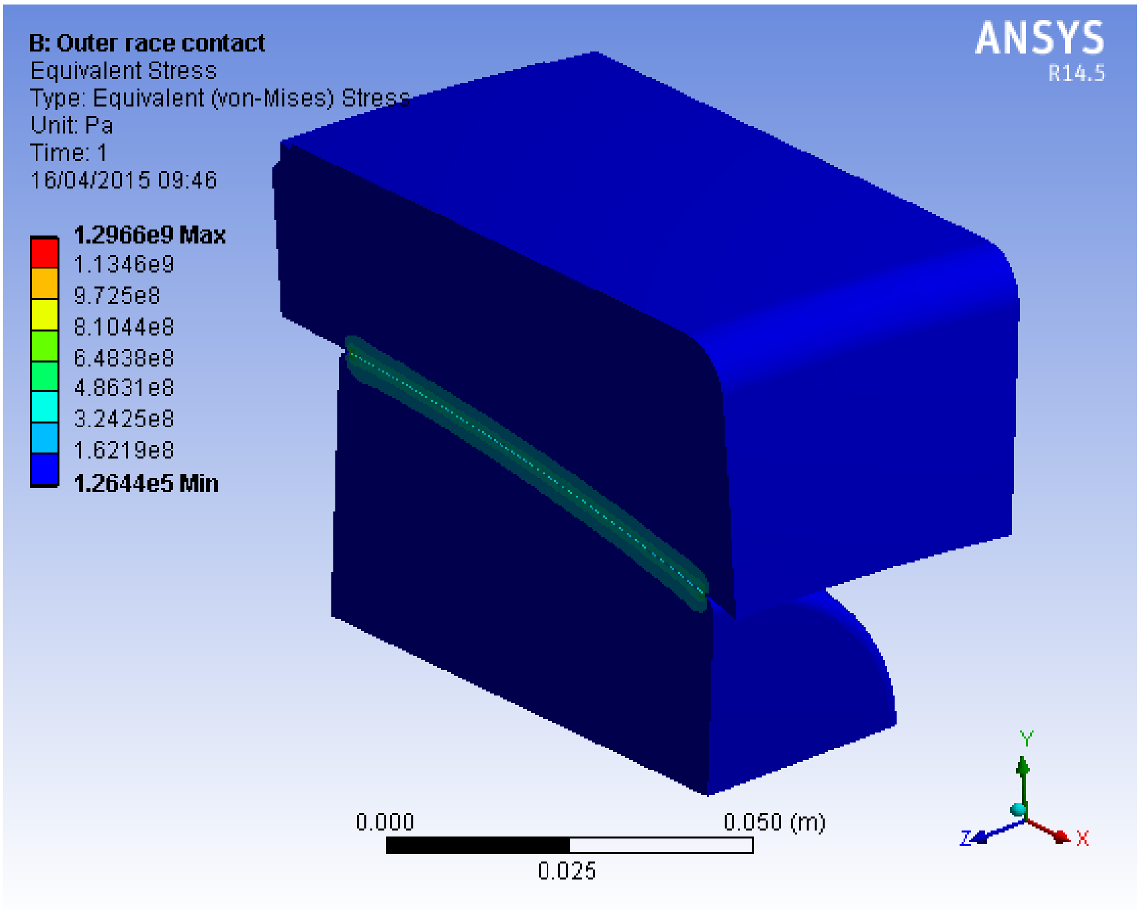


Figure 0-5 FE model of outer race race contact

3.3 Static Validation of Rigid Dynamics Model

Methods for evaluating the load distribution in a static spherical roller bearing have been presented by Harris(2). To validate the RD model, the results are compared to theoretical static scenarios. The first being a static SKF 230/630 CAW/30 loaded by a pure radial load of 100 kN. From Equation 11, $Q_{\max i}$ is calculated as 7.3 kN whereas the RD model evaluates the maximum load as 7.28 kN. Indeed, the reactions within the RD model correlate to within 3.3% of the results from Equation 11. Figure 0-66 shows the loading on individual rolling elements in one row of the bearing from both calculation methods.

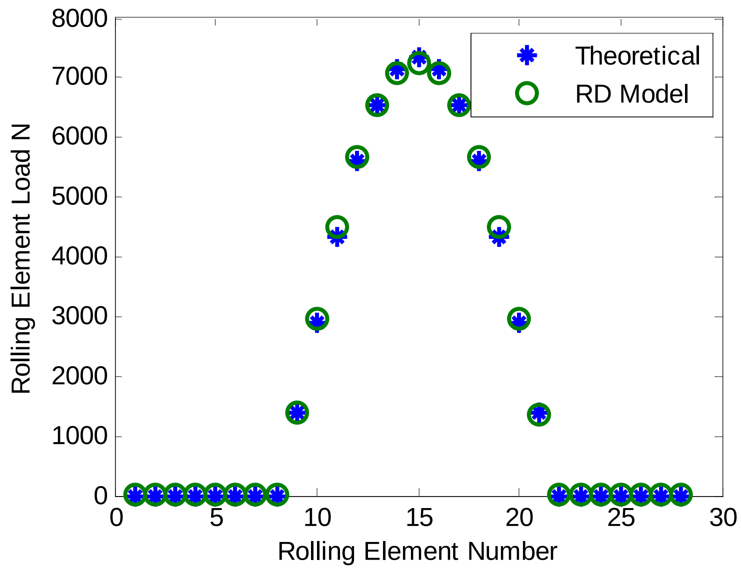


Figure 0-6: Comparision load distribution in static bearing

Similarly, when a 100 kN radial force is applied in conjuction with a 20 kN axial force, Figure 0-7 and Figure 0-88 show the load distributions for the generator side and rotor side of the bearing, respectively.

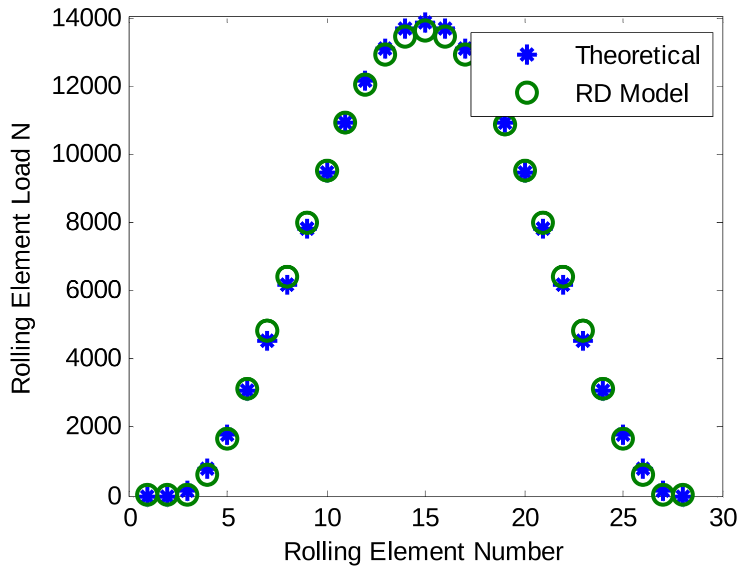


Figure 0-7 Load distribution in generator side

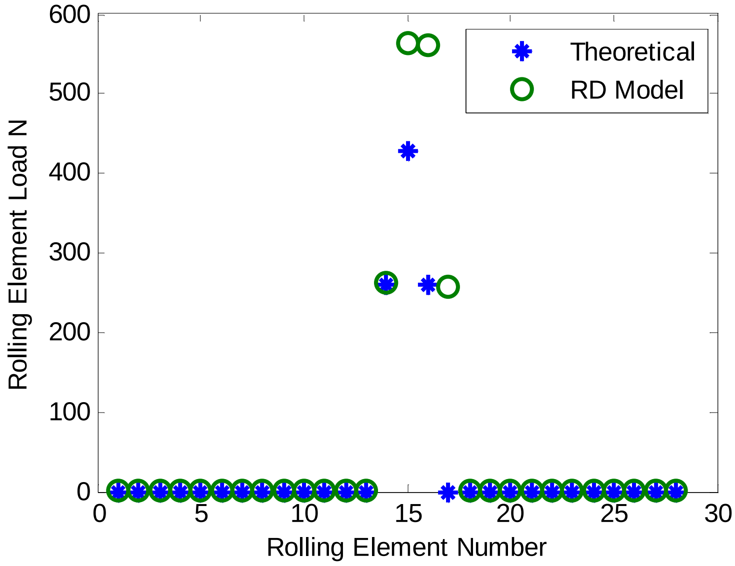


Figure 0-8 Load distribution in rotor side

3.4 Dynamic Validation of Rigid Dynamics Model

In the case of a rotating bearing with a constant load, the load carried by a single rolling element rises to $Q_{\max i}$ when the rolling element location is directly in-line with the effective load. This is observed in the RD model. Figure 0-9 shows the variation of rolling element load in one rotation (60 rpm) if a pure radial load of 100 kN is applied.

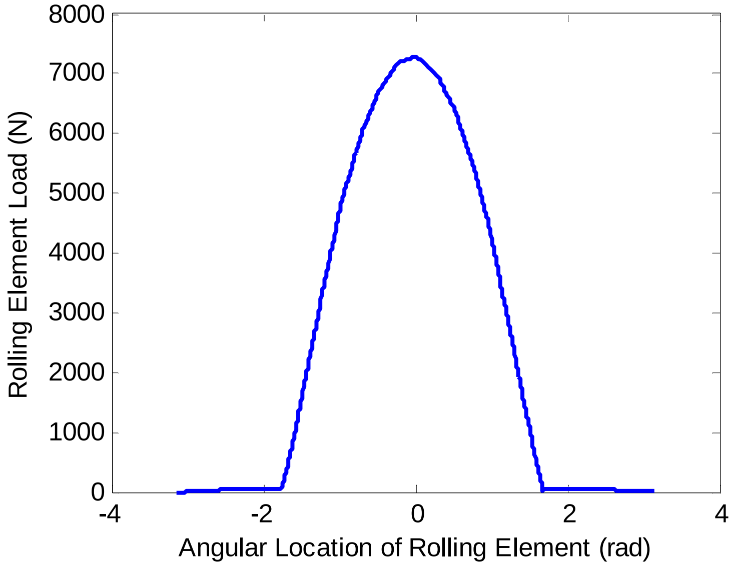


Figure 0-9 Rolling element load in one cage rotation

4. Results/Discussion

To study the effect of the different approaches to calculating the L_{10} life, the bearing was subjected to a 30 second load which may be experienced by a typical 1.5 MW wind turbine operating in a wind field with 12 m/s average wind speed (Figure 0-10). The bearing life is evaluated based on the assumption that this 30 second load is continuously repeated over the entire life of the bearing.

The resultant rolling element loads are presented in Figure 0-11; elements 1 and 14 are at opposite sides of the cage. It is evident that for the variable operating speed and loads applied to the shaft, the resultant bearing loads, as would be expected, vary significantly over time. As the elements in question are at opposite sides of the shaft, it is understandable that peak loads tend to be out of phase. Equally, there is a 6 second cycle apparent on the mean load for each element, which corresponds well to an approximate operating speed of 20 rpm.

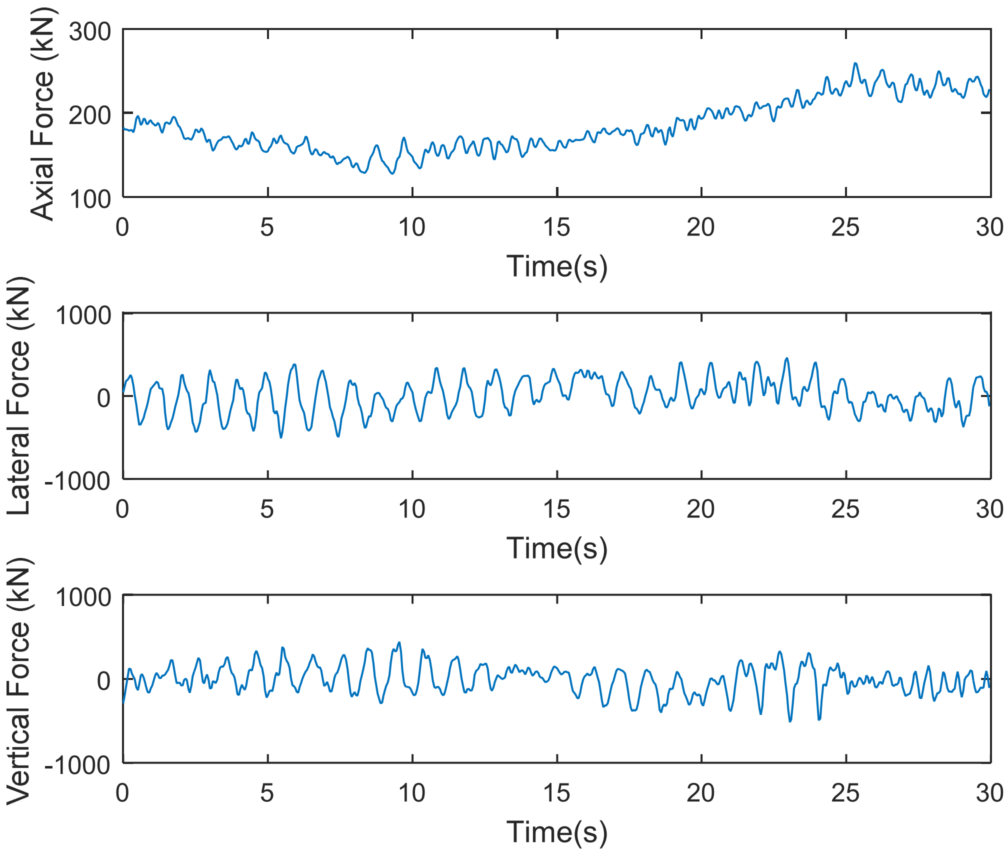


Figure 0-10 Applied shaft loads

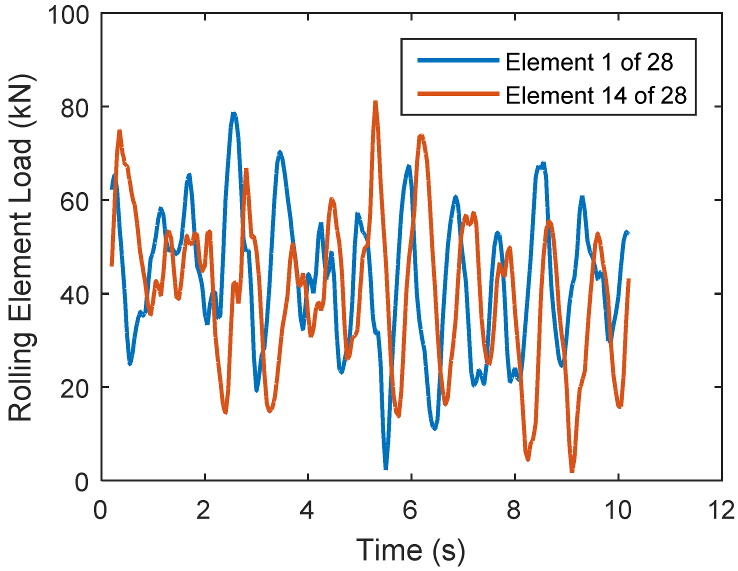


Figure 0-11 Resultant rolling element loads

Using the ISO 281:2007 procedure, the bearing life for the described load case was estimated to be 7.1046 years. In contrast, the more rigorous procedure presented in these findings, where the life of individual raceways are analysed and summed together, predicted a life of 6.31 years. Such a large variation in the L_{10} life calculation (12.6%) can be explained by the difference in the calculation method of the dynamic equivalent load rating. By examining Equation 2 in more detail, assuming an exponent of 10/3 is utilised for a roller bearing, this level of variation can be caused by a 3.5% variation in the calculation of the dynamic equivalent load rating. Due to the method for calculating the expected bearing life, it is clear that greater confidence is required in the determination of operational rolling element loads.

At the initial design stage, Equation 5 offers a simple approximation of the equivalent load rating of the bearing. The procedure for evaluating the equivalent load rating is rooted in empirical formulation. The coefficients of the axial and radial loads which are given in catalogues are generic for the type and size of bearing. Today, advances in computation make it possible to evaluate the dynamic loading history on bearing components with relative ease, thus it may be prudent to integrate more bespoke formulations relevant to the specific bearing designs.

5. Conclusions

A Rigid Dynamics model has been developed and presented which enables dynamic evaluation of loads on the rolling element of a bearing. It has been shown that the standard method for predicting bearing life may deviate from expected life calculated by a more detailed method.

Outputs from the results of the developed main bearing RD model will be utilised as an input to a condition monitoring methodol developed within a further work package. The model will look to replicate some common failure modes of wind turbine main bearings, with possible output data such as load distribution on rolling elements or displacement of raceways as a function of time. These time domain results will be analysed with the proposed algorithms to determine the most appropriate type and location of sensors within the wind turbine, and to set up a baseline for condition monitoring systems to indicate wear, which enhances the ability to identify faults caused by wear from condition monitoring signals and post-processed data.

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