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# Calculation analysis of yaw bearings with a hardened raceway

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ABSTRACT

The yaw bearing is a key support structure of wind turbines and is often exposed to substantial complex loads that cause damage and fatigue failure. Raceway surfaces accommodate high contact stress and require a hardening treatment. The hardened depth has a great influence on both the carrying capacity and fatigue life. We establish a whole finite element model of a yaw bearing and use non-linear springs instead of a ball to obtain the maximum contact load. The results of a strain gauge experiment and an empirical formula are compared to verify the spring model results. A local finite element model of a ball and raceway with different hardened depths is established to analyse the stress distribution and fatigue life. The raceway is divided into a hardened layer, transition layer, and core layer. An indentation experiment verifies the raceway model with different layers. The stress results are compared with Hertz contact theory, and the fatigue life results are compared with yaw bearing fatigue life theory. The influence of different hardened depths on the stress and lifetime of yaw bearings is analysed.

## 1. Introduction

Wind energy is an abundant, renewable green energy. With the recent energy development and utilization occurring worldwide, the wind power industry has become an important component of energy-saving emission reductions and environmental protection. The yaw bearing and pitch bearing are important parts of a wind turbine, as shown in Fig. 1(a), with the former being a key support structure. The yaw bearing is large; contains an inner ring, an outer ring, a ball, a cage and other components, as shown in Fig. 1(b); and is installed in the wind turbine nacelle base to rotate the cabin around the tower and to adjust the fan's windward angle in a timely manner for maximum power output. The substantial complex loads, cabin weight of hundreds of tons, installation location of up to tens of metres, inconvenient disassembly and expensive maintenance demand that yaw bearings have high reliability and long operating lifetimes.

To study yaw bearings, many researchers have used a finite element method, which effectively reduces test costs and shortens development time. Wang et al. [1] established finite element models of single-row and double-row four-point-contact balls to analyse the pitch bearing carrying capacity. The models considered the effects of clearance, raceway curvature and initial contact angle. The degree of influence on the bearing capacity is different when different loads are combined. Guanci et al. [2] considered a four-point contact ball bearing as an example to research the influence of the ball size and a hardened raceway on fatigue life using the stress-life, strain-life, and international standard methods. The results showed that the depth of a hardened raceway cannot be ignored when calculating the fatigue life of rolling bearings. Jon et al. [3] proposed a superelement-based finite element model to research the slewing bearing in wind turbine generators. The ball was replaced with a rigid shell element, rigid beam elements, and traction-only spring elements in the finite element calculation. Although this model cannot simulate the contact between the bearing, blade and rotor, it produces an accurate load distribution while ensuring a low computational cost. Göncz et al. [4] established a finite element model of a three-row roller slewing bearing with a hardened raceway to evaluate the static carrying capacity. The model took into account an arbitrary roller geometry and the raceway material of the slewing bearing, which can help manufacturers improve the static load capacity of the slewing bearing. Gao et al. [5] used non-linear springs instead of a ball to calculate the load distribution of a single-row ball slewing bearing, but this model lacks experimental validation. Aguirrebeitia et al. [6] used theoretical calculations combined with a finite element model considering the preload to analyse the slewing bearing static carrying capacity of wind turbine generators. This model more closely approximated the actual working conditions of slewing bearings in wind turbine generators because theoretical calculations and finite element calculations were combined to validate the model. Göncz et al. [7] assessed the rolling contact fatigue life of an induction hardened raceway. Their experiment yielded 42CrMo high-contact fatigue parameters and verified the fatigue life. Kania et al. [8] used superelements (non-linear truss elements) instead of a ball to calculate the carrying capacity of a slewing bearing while considering the influence of ring flexibility, bolts, and a rolling element. Gao et al. [9] proposed methods to evaluate rolling contact fatigue reliability and combined the methods with Lundberg-Palmgren theory and

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**Nomenclature**

| | |
|---|---|
| $\alpha$ (°) | Contact angle |
| $a_1$ | Life modification factor for reliability (90% reliability) |
| $a_2$ | Life modification factor for bearing steel or material |
| $a_3$ | Life modification factor for lubrication |
| $a_4$ | Life modification factor for a flexible supporting structure |
| $C_a$ | Basic dynamic axial load rating |
| $C_{id}$ | Groove curvature centre of the inner ring, lower raceway |
| $C_{iu}$ | Groove curvature centre of the inner ring, upper raceway |
| $C_{od}$ | Groove curvature centre of the outer ring, lower raceway |
| $C_{ou}$ | Groove curvature centre of the outer ring, upper raceway |
| c | Ductility index |
| $c_1$ | Constant coefficient |
| $c_2$ | Constant coefficient |
| $D_L$ (mm) | Raceway centre diameter |
| $D_i$ (mm) | Outer ring inner diameter |
| $D_O$ (mm) | Outer ring outer diameter |
| $D_w$ (mm) | Ball diameter |
| $d_O$ (mm) | Inner ring, outer diameter |
| $d_i$ (mm) | Inner ring, inner diameter |
| $E_1$ (MPa) | Ball elastic modulus |
| $E_2$ (MPa) | Raceway elastic modulus |
| $f$ | Groove radius coefficient |
| $F_a$ (N) | Axial load |
| $F_r$ (N) | Radial load |
| $H$ (mm) | Outer ring height |
| $H_1$ (mm) | Total height |
| $i$ | Rolling element row |
| $h$ (mm) | Inner ring height |
| $K$ | Elastic deformation index |
| $K'$ (MPa) | Index |
| $k$ | Ball constant |
| $L_{10}$ | Slewing bearing basic rating |
| $L_{nm}$ | Slewing bearing ANSI/ABMA standard-modified rating life |
| $M$(N·mm) | Overturning moment |
| $N_f$ | Cycle number |
| $n$ | Ball number |
| $n'$ | Index |
| $n''$ | Index |
| $r$ (mm) | Groove radius of curvature |
| $P_{ea}$ (N) | Dynamic equivalent axial load rating |
| $Q$ (N) | Contact load |
| $Q_{max}$ (N) | Maximum load |
| $T_1$ | Theoretical number of raceway stress cycles |
| $z$ (mm) | Hardened raceway depth |
| $\alpha$ | Index |
| $\beta$ | Index |
| $\delta$ (mm) | Deformation |
| $\delta_q$ (mm) | Indentation depth |
| $\sigma_{0.2}$ (MPa) | Yield strength |
| $\sigma_a$ (MPa) | Stress amplitude |
| $\sigma'_f$ | Fatigue strength coefficient |
| $\sigma_v$ (MPa) | Equivalent stress |
| $\sigma_{vperm}$ (MPa) | Allowable equivalent stress |
| $\mu_1$ | Ball Poisson ratio |
| $\mu_2$ | Raceway Poisson ratio |
| $\Sigma\rho$ | Sum of curvature |
| $\Delta r_{max}$ | Shear strain amplitude |
| $\Delta\varepsilon_n$ | Principal strain amplitude |
| $\varepsilon_a$ (mm) | Strain amplitude |
| $\varepsilon_e$ (mm) | Elastic strain |
| $\varepsilon_p$ (mm) | Plastic strain |
| $\varepsilon'_f$ | Fatigue extension coefficient |

ISO 281. The contact load, contact geometry parameters, and material parameters were taken into account. The results showed that the reliability of the inner ring is less than that of the outer ring, which was consistent with empirical observations. Göncz et al. [10] calculated the static carrying capacity of three-row roller slewing bearings combined with ring flexibility, non-parallel ring displacement, clearance, and raceway hardening. The vector approach was used to describe the geometry of the bearing and the rolling motion, providing a convenient engineering method for the early design of the slewing bearing. To calculate the fatigue life of a double-row ball slewing bearing, Potočnik et al. [11] proposed a procedure that described the geometric parameters as vectors to consider irregularly shaped bearings. Different calculation theories were used for the fatigue life calculation, and different results were compared with each other. Lai et al. [12] analysed the hardened depth and carrying capacity, and the plastic compression and subsurface destruction were verified by standard contact fatigue tests. Their paper did not consider the dynamic carrying capacity, which deserves further investigation. Kunc et al. [13] researched the bearing capacity with a hardened raceway and experimented with a low cycle carrying capacity. This model assesses the origin and expansion times and the location of cracks that reduce the service life for a rolling rotational connection. Olave and Sagartzazu [14] established a finite element model of a four-point-contact ball slewing bearing in order to analyse the load distribution considering ring geometry effects. Their results were consistent with the theoretical calculations. Daidié and Chaib [15] established a finite element model of a slewing bearing to analyse the changes in load distribution and contact angle. These authors evaluated the change in contact angle and experimentally validated the model effectiveness. Göncz and Drobne [16] established a finite element model of a three-row roller slewing bearing in order to analyse the load and stress distributions. Using the stress-life method, these authors obtained the slewing bearing working life. Göncz and Potočnik [17] used ABAQUS to establish a model of the propagation of a two-dimensional contact fatigue crack. The expansion of the initial contact fatigue crack on the slewing bearing surface was simulated. The analytical results were reliable and could be used more frequently in subsequent analytical applications. Zaretsky et al. [18] used the Monte-Carlo method to test the fatigue life of a rolling bearing with a certain fixed speed. The bearing service life under different bearing steels was compared. Potočnik et al. [19] studied the fatigue life of a double-row ball slewing bearing, as well as the influence of ring deformation and the hardened layer depth.

The critical areas of stress-strain and fatigue life on a raceway with a hardened layer are observed several millimetres below the raceway contact position. The yaw bearing diameter can be several metres. If the whole model is refined to consider this area, the grid number will reach tens of millions. Hundreds of contact pairs can easily lead to non-convergence in the actual analysis. Therefore, using non-linear springs instead of a ball yields the maximum contact load. The relationship between the ball and raceway under an external load is based on Hertz contact theory. Calculation results are compared with the results of the empirical calculation equation. A local finite element model of the ball and raceway is established to observe the influence of different hardened depths on the stress and life of the yaw bearing. Although the non-