Cage Loads of Wind Turbine Blade Bearing

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Abstract

Slewing bearings of wind turbine blades pitch system allow the required oscillation while transferring complex dynamical loads from the rotor blades towards the hub. Most common for this application are double rowed four-point contact ball bearings consisting of two rings and two rolling element sets either equipped with a ring cage or with spacers. By equally distributing the rolling elements along the circumference, the bearing cage balances the loads along the raceways and thereby actively prevents damage mechanisms. However, cage stress analyses imply further optimization potentials leading to higher load capacities and the prediction of cage damaging mechanisms via dynamic simulations. This contribution presents a simulation procedure that calculates rolling element dynamics at the presence of a ring cage by taking the perpendicular movement of the contact ellipses due to the elastic deformations of the bearing rings into account. The procedure is carried out exemplary by the load spectra of a 3 MW reference wind turbine and the actual geometry of a 2.4m diameter bearing.

Keywords. Wind turbine multibody simulation, blade bearing cage, finite element simulation

1. Introduction

In modern wind turbines a well-established bearing type for pitch applications is the double rowed four-point contact ball bearing. These bearings combine cost-effectiveness and high carrying capacity due to their ability to geometrically adapt to high dynamically changing bending moments and axial loads. However, growing rotor sizes in order to increase the system performance of wind turbines has led to the necessity to further enhance the bearings load capacity and sustainability [1]. Thereby, adjusting the shape parameters of the rolling element set is effective though also affecting adjacent components. As an example, increasing the diameter and the number of rolling elements reduces the circumferential spacing required by the bearing cage to separate the rolling elements, hence result in weakening of the cage structure. In this case the damage mechanisms of large slowly oscillating slewing bearings are not sufficiently researched yet and due to their low stiffness technical design standards are not applicable or lead to deviations between calculations and measurement data [2]. Furthermore, research on cage loads naturally concentrates on high speed applications since bearing cages are considered being the critical component at the presence of high centrifugal forces [3].

This contribution presents a simulation approach to determine cage stress and the rolling elements behavior by taking the complex dynamic load spectra and the structure of blade bearings into account. In the present approach a mechanical multibody model of a collective pitch controlled (CPC) 3MW
wind turbine that integrates a real shaped blade bearing is executed to calculate the rolling element load distribution [1]. The dynamic behavior of the rolling elements is described by tracing their contact points on the raceway surface numerically [4, 5]. The dynamic simulation uses a macro-elastic cage model, described by Hahn to calculate the cage pocket forces by comparison of the transient angles of the rolling elements with the cage [6]. Throughout the contribution, a time period is investigated to describe a certain contact event.

2. Approach

The contributed approach shows the impact of load spectra at the blade root on the cage stress during a certain event of rolling element-cage pocket contact. By this means a procedure is described to calculate the dynamic behavior of the rolling elements at the presence of a ring cage, displayed in Figure 1. For the analyses the actual geometry of a double rowed four-point contact ball bearing is added to the mechanical WT model.

![Figure 1. Simulation procedure](image)

Starting with the mechanical WT model$^1$ described in Section 3 the dynamic load spectra ($F_l, M_l$ and $\alpha_{pitch}$) at the blade root are obtained taking the wind field into account. The rolling elements begin equally distributed along the circumference. For each time step their transient position is provided as input data by the first loop in the procedure. Combined with the dynamic load spectra at the blade root, the blade bearing model$^2$ is executed to calculate the rolling element normal forces, further described in Section 4. The dynamical behavior of each rolling element is calculated by integration of the contact vectors circumscribing linear speed profiles between the raceways [4]. By execution of a finite element simulation of the rolling element-raceway contact$^3$ the specific contact angle $\alpha$ is obtained. The model solves the static equilibrium of the ring load distribution acting on the rolling element set between the raceways [2, 7, 8]. The rolling element-cage dynamic model$^4$ parametrizes the bearing geometry and numerically approximates the equations of motion at a constant sampling rate of 100Hz. At the end of the procedure the macro-elastic model of the ring cage$^5$ was implemented to enable the analytical calculation of cage pocket contact forces, due to the second loop providing the cage stiffness for each load case [6]. Upon obtaining the cage pocket contact forces of certain contact events during the dynamic simulation a finite element model of the ring cage is executed to display the cage loads. The specifications and model parameters are listed in Table 1 at the end of this contribution.

3. Mechanical wind turbine model (mbs)

For the determination of load spectra at the blade root, a multibody co-simulation was carried out using SIMPACK and AERODYN [1, 9]. The aerodynamic loads acting on the rotor blades are determined using the Aerodyn v13 code provided by NREL [10]. The model consists of modal reduced flexible blades transferring the wind loads to the blade root, a CPC controller and a blade bearing model. The rotor blades are rigidly coupled to the inner ring of the blade bearing. The outer ring is coupled to the hub. In this contribution the generic wind turbine model of the Center for Wind Power Drives was adjusted by realistic blade shapes that belong to the 3MW reference wind turbine investigated in the publicly funded project HBDV$^v$ (Analyses of highly loaded slewing bearings). The reference wind turbine produces power in a wind range between 3m/s and 25m/s. The referenced wind field used for the load definition is based on DLC1.1 at wind conditions according to wind class 1a, IEC61400 [11]. The contributed cage model is executed at rated wind speed (13m/s). During this state, Figure 2 displays high amplitudes of pitch angle followed by no pitch activity due to transition between rated power and underrated conditions.
4. **Blade bearing model (mbs)**

To calculate dynamic rolling element loads the referenced double rowed four-point ball bearing was implemented in SIMPACK. The model consists of an outer ring rigidly connected to the inertial coordinate system with 0 DOF and an inner ring with 5 DOF coupled to the inertial system to prevent rotation around its middle axis. Both rings are connected by force elements equally distributed along the raceway to substitute each rolling element. A graphic representation of the blade bearing MBS model and the rolling element load distribution is displayed in Figure 5.
The dynamic load spectra acting on the inner bearing ring had been implemented by input functions. The dynamic rolling element loads during oscillation are calculated by interpolation taking the instantaneous position of each rolling element into account. Therefore, a cascade has been set up between the blade bearing model and the dynamic code. Currently, the blade bearing model is based on rigid bearing rings.

5. Integral dynamic model

Generally, rolling element dynamics are determined by calculation of the contact points to describe the angular speed along the circumferential diameter. For high-speed bearings this is accomplished by solving the dynamic equilibrium of contact forces due to accelerations and inertial effects [13]. In contrast to unidirectional rotating bearing cages, blade bearings are oscillating relatively slow superimposed with dynamic loads of high magnitude. Their bearing cages are generally idle in case of normal oscillation of the rotor blades if the circumferential displacements of the rolling elements are compensated by the cage pocket clearance. Furthermore, the maximum oscillation speed of the rotor blades is relatively slow \( w_{\text{max}} = 8^\circ/s \) and approximately \( w_{\text{obs}} = 4^\circ/s \) during the observed contact event. However, rolling elements of blade bearings reach high load magnitudes leading to circumferential displacements of rolling elements as well [2]. This implies high cage pocket contact forces of rolling elements remaining in the load zone during pitch oscillation.

Rolling element dynamic

In the kinematic routine the angular position of each rolling element is accumulated numerically by considering their instantaneous rotational speed. The kinematic analyses require a description of the vectors between the raceway system and the contact points of each rolling element with the inner and the outer raceway curvature. To analyse the dynamic behaviour the essential part of various published models is the calculation of these contact points determined by the angle and the deflection of the pressure ellipses between rolling element and raceway surface [5]. To meet the static equilibrium contact angles rise due to displacements, deformations and misalignments of the rings caused by bending moments and axial loads [2, 7]. In this contribution the contact points are calculated by solving the static equilibrium of the rolling element raceway contact via finite element method. The approach is based on simplifications, such as the race control hypothesis that assumes no relative motion between the rolling elements and the raceway surface [13]. The model further addresses highly loaded blade bearings at low oscillation speed. Therefore, dynamic accelerations due to centrifugal and inertial effects are neglected. Figure 6 shows the geometry of the rolling element-raceway interaction used in this approach, generally described by Gupta in “Advanced Dynamics of Rolling Elements” [4].

![Figure 6. Parameterization of the rolling element-raceway contact [4]](image)

To determine the contact vectors the dynamic model described by Gupta was adjusted to the shapes of the raceways of four-point contact ball bearings. Furthermore, it has been assumed that the inertial and the raceway coordinate systems have the same position. Due to the overlapping load zones of blade bearings (Figure 5), the resulting rolling element speed is calculated by vector addition of both diagonal
speed profiles defined by their contact angles. The circumferential speed of the linear profiles is described by tracing the contact angles for each raceway according to equation 1.

\[ \vec{v}_{R,i} = \frac{D_m \cdot \pi \cdot n_R \cdot \vec{r}_{IR,i}}{\vec{r}_{IR,i} + \vec{r}_{OR,i}} \]  

(1)

Thereby, \( \vec{r}_{IR}, \vec{r}_{OR} \) are the contact radii towards the bearing center axis, \( n_R \) is the rotational speed of the inner ring and \( D_m \) is the rolling diameter. The kinematic model neglects the displacement of the rolling elements geometric center due to their elastic deformation towards the center of the raceway curvature \( \vec{r}_{BR} \). Considering this effect requires an integration of the flattening effect due to the compression of the rolling elements by means of hertzian theory or alternatively by finite element contact simulation. Figure 7 (a) shows a comparison of the contact angles at the upper row (RW1) for a ring load distribution in negative direction of the blade root coordinates. Starting from the mounted contact angle at 47° both angles deviate while arising due to different deformations of the outer ring (OR) and the inner ring (IR). Thereby, the contact angle \( \alpha \) is obtained by means of finite element simulation of the force vectors between the ring planes orthogonal towards the rotational bearing axis displayed in Figure 7 (b).

**Cage speed**

There are various existing methods to calculate the cage speed either couple the cage to the rolling elements or to the shear effect of the lubricant. Since the contributed model currently neglects tribological losses methods coupling the cage to the rolling elements are considered. In case of modelling the cage coupled to the rolling elements the cage speed can either be determined by the leading rolling element, or by weighting the rolling element normal forces propagated for high speed applications by Wagner and Krinner et al. [13]. Alternatively, the cage speed in the present approach is calculated by the mean rotational speed, according to equation 2.

\[ \bar{\omega}_c = \frac{\sum_i \vec{\omega}_{R,i}}{i} \]  

(2)

The mean rotational speed provides more manageable results for an initial approach. In case of oscillatory rotation and multiple rolling element cage pocket contacts determining the cage speed by the leading rolling element is inapplicable. To compare the suitability of the weighting approach with the mean rotational speed further studies are necessary.

**Sampling rate**

In the kinematic model the equations of motion are integrated using a constant step size of 10 ms determined by the sampling rate of the wind turbine MBS model. Generally, required sampling rates are
10² times higher for rolling element raceway contacts, according to Weinzapfel and Sadeghi [3]. However, these frequencies are considered for high speed applications and the maximum time step size is limited due to the required number of discrete elements in the elastic cage model leading to divergent simulations. By contrast, higher sampling rates may produce better results solving the dynamic equilibrium between the traction forces at the rolling element raceway contact and the elastic reactions of rolling element-cage pocket contact forces.

**Macro-elastic cage model**
Flexible bearing cage models show significantly smaller rolling element-cage forces (approx. 75%) than rigid cage models [3, 14]. The present approach thus considers the cage macro-elasticity and the elasticity of the rolling element-cage pocket contact for each numerical step. Thereby, the macro-elasticity is described by discretizing the cage into \( z = 139 \) segments in order to form a serial connection of spring-damper elements (SD element) connected by nodes at their cage pocket center. The thus obtained SD-Polygon is implemented in the dynamic simulation to calculate the contact forces based on the dynamic deflections inside of each cage pocket. The SD-Polygon enables the analytical approximation of cage loads. Figure 8 (a) shows the SD-Polygon exemplary for a cage comprising 10 cage pockets, described by Hahn [6]. The detail level of the discretization is adapted by the number of SD-elements replacing the cage structure. In the present approach the cage is discretized by one node per cage pocket center, displayed in Figure 8 (b).

![SD-Polygon, 10 nodes](image)

![Longitudinal cage Section, 5 discretized segments](image)

**Figure 8. Model representation of the cage discretization [6]**

The stiffness matrix \( K_T \) of a single segment is simulated by finite elements according to equation 3.

\[
K_T^{-1} \cdot \overrightarrow{F} = \overrightarrow{u}
\]  

(3)

Vice versa, the contact forces are calculated numerically according to equation 4. Thereby, \( \overrightarrow{u}_t \) is the contact deflection and \( \overrightarrow{F}_{N,t} \) is the contact force.

\[
\overrightarrow{F}_{N,t} = K_T \cdot \overrightarrow{u}_t
\]  

(4)

During the contribution only the first entry of the force vector \( \overrightarrow{F} \) in circumferential direction of the cage is simulated to approximate the stiffness matrix for each SD-element. Figure 9 shows the FEM results of the deformations in circumferential direction \( u_\theta \) that determine the segment stiffness. In case of a rolling element cage pocket contact the stiffness matrix is approximated by the FEM results of the contact pair.

![FEM-analyses of segment and contact stiffness](image)

**Figure 9. FEM-analyses of segment and contact stiffness**
**FEM cage model**
The global FEM model of the bearing cage aims to show the cage stress as a result of contact forces. The cage model consists of 139 cage pockets that are connected along the axial guiding clearance to the cylindrical system of inertia with one tangential DOF and three rotational DOF. By this means the rotation of the discrete elements along the circumference is allowed in case of tangential contact forces. The calculation of the deflections depends on the assumption of a mean rotational cage speed compared with the differential rolling element speed. The FEM model thus allows a certain tangential elasticity but prevents the global cage rotation by a fixed coupling connected to the cage pocket center at a segment at the opposing side of the cage.

### 6. Results
In this section a certain event of rolling element-cage pocket contact is described. Through the simulation various contact events had been identified as a result of angular displacements of the rolling elements relative to the cage. At begin of the observed time period the rolling elements are distributed along the circumference following several pitch cycles. Then, the pitch controller had been initializing rated wind speed after idling with a pitch impulse of approximately 4°/s, displayed in Figure 10. While decreasing pitch angle 10 of 139 rolling elements pinched the cage at time step 224.87s. The pitch angle that has led to multiple rolling element cage contacts during the time period is given in Figure 11.

![Figure 10. Angular pitch speed](image1)

![Figure 11. Pitch angle](image2)

The amount and magnitude of cage pocket contacts during the observed time period is given in Figure 12. The diagrams show the maximum cage pocket force during different contact events, even though multiple rolling elements contacted. During the simulation procedure the rolling elements had been traced by fixed indices to record their migration. Thereby, the rolling elements that participated at the contact are identified in the diagrams. During the contact event (at 224.87s) the rolling elements hit the cage from opposing load zones in negative pitch angle direction. The comparison of both diagrams shows the cage had been driven by the rolling elements alternately according to the pitch angle. Furthermore, cage pocket contacts are initialized by the leading rolling element inducing a characteristic load amplitude followed by a transition of the leading rolling element. Since the current model assumes simplified rolling element slip behavior the amplitudes decrease step wisely. The analyses of the frequency and magnitude of cage pocket contacts implies the prediction of cage wear as a result of pitch activity. Furthermore, during the observed time period no contrary cage pocket contact forces leading to cage strain have been identified [15].

![Figure 12. Cage pocket contact events; left: negative pitch rotation, right: positive pitch rotation](image3)
The rolling elements contacted the cage next to the maximum of the load zone near 0°, relative to the blade coordinate system. Figure 13 shows the load distribution of rolling element normal forces had been calculated by a multibody simulation of the blade bearing, introduced in Section 4. Thereby, the rings are modelled rigidly and each rolling element force was linear interpolated since the loads depend on transient rolling element angles during dynamical motion. Due to the assumption of rigid rings the deformations of the blade bearing are neglected.

The distribution of cage pocket forces during the contact event is given in Figure 14. Before the event occurred, the rolling elements had migrated to an angular distribution relative to the cage pocket center. As a result of the pitch rotation in negative direction the rolling elements close enough to the cage pocket front contacted. The orientation of the cage pocket fronts switches with the oscillation angle. Accordingly, the instantaneous displacements of the rolling elements relative to the cage pocket centers are given in Figure 15. The rolling elements oscillating alongside the load zones show discontinuous displacements. The other displacements agree with the formation of opposite load zones near 0° and 180° as described by [2, 7]. The leading rolling element during the contact has been number 5 at 1.87°.

Figure 16 shows the cage load case implemented in the finite element model. The contact forces are applied by continuum distributing couplings with a specified influence radius by the length of their corresponding hertzian pressure ellipse. Accordingly, the von Mises stresses are displayed in Figure 17. In front of the cage pocket contacts in rotational direction the cage is compressed. In the trail of rolling element 13 the cage is strained. The analyses show the characteristic stress peaks due to notch effects.
and shear stress at both axial sides of the cage pocket centers. Thereby, the maximum stress is at the peaks one cage pocket ahead, even if the maximum contact force is at pocket five.

Figure 16. Cage load case during contact

Figure 17. Cage stress during contact

Table 1. Specifications and model parameters

| Parameter                      | Value                  |
|-------------------------------|------------------------|
| bearing type                  | four-point ball bearing|
| cage type                     | ring cage              |
| bearing diameter (mm)         | 2400                   |
| ball diameter (mm)            | 45                     |
| number of rows                | 2                      |
| no. of rolling elements       | 139                    |
| mounted contact angle (°)     | 47                     |
| cage geometry (mm)            | 60x6x2400              |
| cage pocket diameter (mm)     | 47                     |
| WT power (MW)                 | 3                      |
| rotor diameter WT (m)         | 104                    |
| friction coefficient          | 0.03                   |
| e.-modulus (GPa)              | 210                    |
| density (kg/m³)               | 7800                   |
| sampling rate (Hz)            | 100                    |
| controller type               | CPC                    |

7. Conclusion

In this contribution a simulation procedure is presented that allows the investigation of cage stress induced by load spectra of wind turbine blades blade bearing. The procedure consists of a mechanical wind turbine multibody simulation that integrates the real blade shape of a 3 MW wind turbine mounted on a double rowed four-point contact ball bearing. To describe the rolling element dynamics the free contact angles are equated with the transient contact points for each raceway curvature. The structural finite element simulations of the bearing and the cage are based on real shaped geometries. Cage pocket contact forces had been calculated by a macro-elastic cage model implemented in the dynamic model. The results show a cage load case that had been caused by the pitch system during the initialization phase at rated wind speed. It was shown that cage stress is often caused by multiple rolling element cage pocket contacts. Thereby, transitions of the leading rolling elements occur that agree with the literature. Furthermore, it was shown that load peaks at the cage pocket center occur due to compression in front of the contacting rolling elements. Due to the necessity to further increase the sustainability and the load capacity of blade bearings, optimizing the bearing cage design potentially allows adjusting the rolling element set. The presented results are at an early state and requiring further investigation. For this purpose, blade bearing tests are scheduled on samples at the blade bearing test rig of the Center for Wind
Power Drives, including measurements of the stress peaks by strain gauges. Upon validation of the cage loads, evaluations on further design load cases are suggested to show blade loads that produce maximum cage stress.

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HBDV – Design of highly loaded slewing bearings

8. References

[1] Berroth J, Jacobs G, Kroll T and Schelenz R 2016 Investigation on pitch system loads by means of an integral multi body simulation approach J. Phys.: Conf. Ser. 753 112002
[2] Schwack F, Stammler M, Poll G and Reuter A 2016 Comparison of Life Calculations for Oscillating Bearings Considering Individual Pitch Control in Wind Turbines J. Phys.: Conf. Ser. 753 112013
[3] Weinzapfel N and Sadeghi F 2009 A Discrete Element Approach for Modeling Cage Flexibility in Ball Bearing Dynamics Simulations Journal of Tribology 131 21102–11
[4] Gupta P K 1984 Advanced Dynamics of rolling elements (Berlin: Springer), ISBN: 978-3540960317
[5] Harris T A and Kotzalas M N 2006 Rolling Bearing Analysis - 2 Volume Set (Boca Raton: CRC Press)
[6] Hahn K 2005 Dynamik-Simulation von Wälzlagerkäfigen (Berichte aus der Konstruktionstechnik) (Aachen: Shaker), ISBN: 3-8322-3760-7
[7] Daidié A, Chaib Z and Ghosn A 2008 3D Simplified Finite Elements Analysis of Load and Contact Angle in a Slewing Ball Bearing Journal of Mechanical Design 130 421
[8] Krynke M, Kania L and Mazanek E 2011 Modelling the Contact between the Rolling Elements and the Raceways of Bulky Slewing Bearings KEM 490 166–78
[9] Roscher B, Werkmeister A, Jacobs G and Schelenz R 2017 Modelling of Wind Turbine Loads nearby a Wind Farm J. Phys.: Conf. Ser. 854 12038
[10] Moriarty P J and Hansen A C 2005 AeroDyn Theory Manual: Technical Report, National Renewable Energy Laboratory (Golden, Colorado (US))
[11] Wind turbines - Part 1: Design requirements (IEC 61400-1:2005 + A1:2010); German version EN 61400-1:2005 + A1:2010 0127,1 (VDE-Bestimmung VDE 0127-1) 2011st edn (Berlin: Beuth)
[12] Germanischer Lloyd Industrial Services 2010 Guideline for the Certification of Wind Turbines (Hamburg: Germanischer Lloyd)
[13] Wagner C, Krinner A, Thümmel T and Rixen D 2017 Full Dynamic Ball Bearing Model with Elastic Outer Ring for High Speed Applications Lubricants 5 17
[14] Ashtekar A and Sadeghi F 2012 A New Approach for Including Cage Flexibility in Dynamic Bearing Models by Using Combined Explicit Finite and Discrete Element Methods Journal of Tribology 134 1
[15] Schul C 1997 Einfluß der Baugröße auf die Lebensdauer feststoffgeschmierter Kugellager (Fortschritt-Berichte VDI Reihe 1, Konstruktionstechnik, Maschinenelemente vol 283) (Düsseldorf: VDI-Verl.), ISBN: 3183283018