Investigation of the individual load distribution of a blade bearing test rig by means of finite element simulation

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Abstract. To carry out fatigue tests on the blade bearing test rig used in this work, knowledge of the maximum contact pressure and position of the contact ellipse in each rolling contact is necessary. Since these cannot be measured directly, a detailed FE-model of the blade bearings and the test rig components is set up. Previous studies use non-linear spring to simplify the raceway ball interaction. In this paper each rolling contact is modelled with a surface to surface interaction. This allows the evaluation of the maximum contact pressure and the distance between the contact area and the edge of the raceway. Furthermore, measurements of the relative displacement between the outer and inner ring for different load steps were carried out and show a good correlation with the simulation results.

1. Introduction
Blade bearings in wind turbines allow a rotation of the rotor blades in pitch direction and thus a control of the occurring loads and power output of the turbine. In the worst case, a defect of the blade bearings can lead to a failure of the power control or to losing a rotor blade.[1] Therefore a robust design of the blade bearings and thus avoidance of a failure is absolutely necessary. The blade bearing is predominantly subjected to high dynamic bending moments, induced by aerodynamic and weight forces, while the bearing stands still or oscillates with small angles. [2, 3]

In the majority of wind turbines of the 3 MW classes, double-row four-point contact bearings are used as blade bearings. As shown in Figure 1, the load is transmitted between the inner and outer ring via two rows of balls. The angle \( \alpha \) at which the loads are transmitted, and thus the position of the center of the contact ellipse between the rolling element and raceway, changes with increasing load. In case of an overload, the contact ellipse runs out of the raceway. This results in an impermissibly high surface pressure in the area of the raceway edge, so called edge loading. [4, 5] Additional damage mechanisms which must be taken into account in bearing design process are wear and fatigue. [6] Investigations have shown that common bearing standards such as DIN ISO 76 or 281 [7] does not result in a realistic lifetime calculation and
therefore should not be used during the design process. [8, 9] One reason for this is the unconventional oscillating movement of the bearings. Additionally, the service life on which the design standard is based was determined on comparatively small bearings on standardized test rigs such as FE8. In comparison blade bearings are much bigger. Furthermore, other materials and manufacturing processes are used. The fatigue behavior of the blade bearings should be investigated on original bearings.

In this work a test rig as shown in Figure 2 is used to carry out wear and fatigue tests of real size blade bearings to get a deeper understanding of the occurring failure mechanisms. The test rig is able to apply a static axial load to the bearings and an attached pitch drive allows an oscillation movement of the bearings.

The fatigue behavior depends largely on the level of surface pressure and the associated stress condition inside the raceway material. Compared to its size, the blade bearing has a comparatively low stiffness. Therefore, the load distribution and thus the individual load on each rolling element is highly dependent on the stiffness properties of the attached components. [10] A detailed FE model of the test rig is used to determine the internal load distribution. Naturally, fatigue damage occurs more quickly under high loads. To keep the test times short, higher loads than those occurring in reality are used. However, it must be ensured that no edge loading occurs.

For assembly the blade bearing, the rolling elements are guided through a small opening in the outer ring. This so-called soft spot is a weakness in the raceway. In real operation, the soft spot is positioned in such a way that it is loaded as less as possible. In this work, the influence of the soft spot is taken into account and the load application on the test rig is optimized in order to keep the load applied here as low as possible.

2. Methodology

2.1. Raceway test rig and test methodology

The loads occurring in the field at the blade bearing result in a load distribution on the individual rolling elements. In order to apply these multi-axial and dynamic loads on a test rig, a very complex test rig design and actuators are necessary. In the test method used here, it is assumed that the load direction from which the individual load on the rolling elements results is irrelevant to fatigue behaviour. The main influence factors are the maximum contact pressure and the number of load cycles. In the test concept used here, the load is applied only in axial direction, which enables a comparatively simple test rig design and a defined maximum contact pressure. The fatigue tests are carried out on two load levels (2500MPa and 3000MPa) and the number of load cycles until a damage occurs is determined. The results can be used to create an S-N curve of the bearing. The connection to the real turbine is established with the linear damage accumulation method according to Palmgren and Miller.

Figure 2: The test rig is used to carry out wear and fatigue tests of two real size blade bearings.
With the test rig shown in the Figure 2 two original blade bearings are tested simultaneously. Between the inner ring of both bearings a spacer ring and a stiffening ring are located. All of these parts are firmly connected with bolts. On each outer ring four stiffening plates are attached. Between the stiffening plates 56 hydraulic cylinders are located which can apply an axial load of maximal 5.6 MN on the bearings. To minimize the load on the soft spot, no load is applied in this area. Since an asymmetrical load application leads to an ovalization of the bearings and thus a strong inhomogeneous load distribution and edge loading, no load is applied every 90° as well. The bearings can be oscillated by a pitch drive. Typically, oscillation angles < 5° are used to produce wear damages and oscillation angle > 5° to produce fatigue damage. The maximum oscillation speed is 9°/s.

In double-row four-point bearing, the load is divided between both rows of balls, as shown in Figure 1. The division ratio depends on the stiffness of the test rig but also on the manufacturing accuracy and is therefore difficult to predict. To avoid this inaccuracy, only one row of balls is filled during the tests. In addition, this allows a higher local load in each rolling contact. This reduces the duration for fatigue tests.

The geometry parameters of the blade bearings are given in Table 1.

### Table 1: Geometry parameter of the double row four-point bearing

| bearing diameter | Raceway diameter | Ball diameter | amount of balls |
|------------------|------------------|---------------|----------------|
| 2.3 m            | 47 mm            | 45 mm         | 140            |

2.2. Validation measurement setup

Direct measurement of the maximum contact pressure in the rolling contact and the position of the contact ellipse is currently not possible due to poor accessibility. A determination is only possible based on simulation models. To validate the FE model, the displacement between the outer and inner ring due to the applied axial load is measured and compared with the FE model.

![Figure 3](image1.png)
**Figure 3:** Measurement setup of the displacement measurement. The vertical displacement between the outer and inner ring is measured using mechanical dial gauges.

![Figure 4](image2.png)
**Figure 4:** Displacement of the bearing rings in the FE model. The colours represent the displacement in vertical direction (blue low, red high).

As shown in Figure 3, the dial gauges are fixed on the inner ring and the relative vertical displacement of the outer ring is measured for different axial loads. The dial gauges have a measuring accuracy of 5 μ. Figure 4 shows the simulated displacement of the outer and inner ring under axial load. It can be seen that the outer ring bend under axial load. This leads to different vertical displacements on the outer
and inner edge of the outer ring. Therefore, measurements are taken once on the outer side and once on the inner side of the outer ring, like shown in Figure 4. Measurements have shown that a measurement error caused by slight bending of the inner ring and thus tilting of the fixed point is very small and can be neglected. The measurement is carried out at several points along the circumference.

In the FE model the relative displacement of a node on the inner and outer ring in y direction along the circumference is evaluated, as shown in Figure 4.

2.3. FE Simulation model
A FE-simulation model of the test rig, as shown in Figure 5, is build up in Abaqus. All parts including the rolling elements, attached parts and bearing rings are modelled as flexible bodies. For meshing, linear hex elements were used for all components.

To reduce the computational time and complexity of the FE model, the rolling element - raceway interaction is usually simplified with non-linear springs. [11–13] This modelling approach allows a sufficient determination of the load distribution on the individual rolling elements. However, a determination of the position of the contact ellipse and thus a prediction of edge loading is not possible. This paper utilizes a surface-to-surface contact interaction between all rolling elements and raceways of the whole bearing. This makes it possible to predict edge loading. A penalty constraint enforcement method was used in the contact formulation. Here the contact force is proportional to the normal penetration of the ball and raceway. A linear penalty stiffness in the range of the elastic modulus of steel was used.

Figure 5: FE Simulation model of the test rig. Partitioning of the balls and raceway and centering springs (top left). Contact pressure of a single contact (bottom left)

A rigid-body motion of the balls can lead to bad convergence behavior of the model. To prevent this, springs are modelled between balls and raceway. Here a low spring stiffness is necessary to do not influence the simulation results.

The forces of the screws on the outer and inner ring are modelled with pre tensioned spring elements. The axial hydraulic cylinders are modelled as constant axial force elements. The applied force can be varied, and the contact pressure, contact area, and displacement for each rolling element can be analyzed.

Table 2 shows global parameters of the model size and required computing resources.
Table 2: FE-model size and calculation time

| total number of nodes | total elements in contact interaction | memory needed to minimize I/O | total computation time |
|-----------------------|---------------------------------------|------------------------------|-----------------------|
| 13.14m                | 1.6m                                  | 245 Gb                       | 20.5 h                |

3. Results

3.1. Validation of the relative displacement of the FE-model

Figure 6: Relative vertical displacement for simulation and measurement for different axial loads over the entire circumference of the bearing. Between the inner ring and outer edge of the outer ring (left) and between inner ring and inner edge of the outer ring

Figure 6 shows the measured and simulated relative displacement between the inner ring and the outer edge of the outer ring for three different axial loads. This shows a strong fluctuation along the circumference. In the center of the loaded areas (45°, 135°, 225°, 315°) a large increase of displacement is visible. The highest displacement of 1030 μm is located at 315° at an axial load of 4.48 MN. In the area of no introduced load (0°, 90°, 180°, 270°) a reduced displacement up to 60% compared to the highly loaded area occurs. Furthermore, the gaps between the plates attached on the outer ring of the bearings lead to lower local stiffness at this location and consequently to a lower local force and displacement. A comparison between 90° and 270° shows an asymmetry of displacement. The reason for this is the stiffening effect of the pitch drive mount. This leads to a more uniform deformation inside this angular region. Due to a poor accessibility, it was not possible to perform measurements in this area.

The simulated relative displacement between the inner ring and the inner edge of the outer ring, shown in Figure 6, shows a similar course over the circumference. In principle, the displacements are reduced by up to 30% compared to the outer edge. The reason for this is the tilting of the outer ring, shown in Figure 4.
Especially in the area of strong deformation, simulation and measurement show a good correlation at all three load ranges. Stronger deviations are visible in the lower loaded areas. The reason could be non-linear effects such as bearing clearance or by production-related deviations in geometry and thus uneven load flow.

3.2. Maximum contact pressure

Figure 7: Maximum contact pressure of each raceway ball contacted for the inner and outer ring for difference axial loads. For the upper bearing (left) and the lower bearing (right)

Figure 7 shows the maximum contact pressure of each rolling contact for the inner and outer ring of both bearings. The highest pressure of 3.5 GPa occurs in the area of 315° at 4.48 NM axial load. Like shown in the displacement, a reduction of the contacted pressure is visible in the area of (0°, 90°, 180°, 270°). The pressure is reduced by up to 30% in the area of 0° and 180°. The screws in the outer and inner ring leads to an uneven stiffness in the rings. This leads to a crisscross pattern in the contact pressure of adjacent rolling elements. Since the amount of screws and balls are not equal. In comparison to the rolling elements, the geometry of the raceway of the inner ring is convex. Due to the lower osculation, the maximum pressure is generally slightly higher on the inner ring than on the outer ring for the same normal force. This effect is more dominant at low loads than for higher loads.

The force of the upper bearing is a bit higher in the area of 90° degrees due to the increased stiffness of the attached pitch drive mount.
3.3. Position of the contact ellipse and prediction of edge loading

As shown in Figure 8, the distance $\Delta E$ between the outer edge of the contact ellipse and the raceway edge was calculated to predict edge loading. The maximum resolution of the distance corresponds to the grid width of the mesh. If the distance is zero, the contact ellipse is cut off, the contact surface is reduced and thus the maximum surface pressure increases.

Figure 8: Determination of the distance $\Delta E$ between the contact ellipse to the raceway edge

![Figure 8](image)

Figure 9 shows the distance between the contact ellipse of the outer rings and the raceway edge for different axial loads. Upper bearing left, lower bearing right.

![Figure 9](image)

Figure 9: Distance between the contact ellipse of the outer rings and the raceway edge for different axial loads. Upper bearing left, lower bearing right.

Figure 9 shows the distance between the contact ellipse on the raceway of the outer ring and the raceway edge for different axial loads. On the one hand, the gradual change results from the discrediting by the mesh. On the other hand, like seen in the contact pressure, the screw connections result in discontinuous load distribution and therefore discontinuous position of the contact ellipse. For low loads, the contact ellipse runs in the middle of the raceway and a sufficient distance of 4-6 mm to the raceway edge. For increasing axial loads, the contact ellipse moves towards the lower raceway edge, particularly in the highly loaded areas ($45^\circ$, $135^\circ$, $225^\circ$, $315^\circ$). At a total axial load of $7 \text{ MN}$, the contact ellipse reaches the raceway in the highly loaded areas and edge loading occurs. This load level should be avoided during the tests.

4. Conclusion

In this work, a FE-Simulation model of a blade bearing test rig was set up. The model focused mainly on the stiffness behaviour of the bearing and surrounding parts and the ball raceway interaction. A
novelty is a detailed surface to surface contact interaction for all rolling element - raceway contacts. The evaluation of the maximum surface pressure shows that areas with reduced load could be created by a targeted positioning of the hydraulic cylinders and take advantage of locally varying stiffness of the surrounding parts. The soft spot can be positioned in these areas to avoid unintentional fatigue in this area.

By evaluating the position of the respective contact ellipse and its distance to the raceway edge, edge loading can be predicted. A beginning edge loading occurs from an axial load of 7 NM. This high load should not be exceeded during the fatigue tests to prevent unintentional fatigue in the area of the edge. To validate the FE model, the relative displacement between inner and outer ring was measured for different axial loads. A comparison with the simulation results shows a very good agreement especially in the heavily loaded areas.

In future work, the bearing model created in this paper can be integrated into an FE model of the hub and blades of a wind turbine to predict edge loading under extreme load cases.

5. Acknowledgments
The results of this paper has be carried out in the project HBDV – “Design of highly loaded slewing rings”. Funded by the German Federal Ministry for Economic Affairs and Energy (BMWi).

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