“FlexPad” - Innovative conical sliding bearing for the main shaft of wind turbines

To cite this article: Tim Schröder et al 2019 J. Phys.: Conf. Ser. 1222 012026

View the article online for updates and enhancements.
“FlexPad” - Innovative conical sliding bearing for the main shaft of wind turbines

Tim Schröder*, Georg Jacobs1; Amadeus Rolink1; Dennis Bosse1
1 Chair for Wind Power Drives, RWTH Aachen University
* Corresponding author: tim.schroeder@cwd.rwth-aachen.de

Abstract. This paper shows an innovative approach concerning a new type of rotor main bearings for wind turbines. The conical sliding bearing with its flexible support structure is designed for the specific loads and operating conditions of wind turbines. It is robust, compact and offers superior serviceability since individually replaceable bearing pads obviate the disassembling of the whole drivetrain during repair/maintenance.

After a short description of the bearing design the results of functional testing under real conditions in full-scale on a 1 MW system test bench are presented. The achieved load-bearing properties, the mixed friction performance and the start-up behaviour will be discussed.

1. Introduction

Wind turbines (WT) count as one of the most important techniques of renewable energy production and have a constantly increasing market share within the electric power sector. Besides the continuously increasing size and performance of wind turbines, the improvement of reliability and reduction of costs belong to the biggest current technical challenges. Especially roller bearings face their technical limits in the drivetrain of wind turbines, with a conspicuous high failure rate. Therefore, reliable, cost efficient and easy to maintain alternatives are of interest.

For the main shaft bearing this can be achieved by a sliding bearing which is, due to a conical shape of the sliding surface, capable to handle axial and radial loads in a very compact structure. By means of extensive Elasto-Hydrodynamic (EHD) simulations a new main bearing concept for wind turbines was found. Since the shaft of wind turbines is loaded with heavy tilting moments special design measures were required to avoid edge wear.

The contribution will describe the design process of this main bearing and experimental results of a full-scale demonstrator test.

2. Approach

![Figure 1. WT with double row tapered roller main bearing [www.skf.com]](image)
WT drivetrains with double row tapered roller main bearings, like shown in Figure 1, have major advantages compared to a conventional three-point support as for example the drivetrain can be designed very short, which leads to lightweight components (esp. shaft) and a short load path from the rotor to the tower. On the other hand the big diameter of these bearings in combination with the high number of rollers leads to fatigue issues due to the high number of overrollings. Also accounted by the big diameter, it is a big challenge to handle tolerances in the manufacturing process and system deformations during operation. Both have a big influence on the pressure distribution and consequently also on the bearing lifetime [1, 2]. The application of sliding bearings instead of roller bearings offers the chance to dismantle these barriers, which motivated the investigations on a conical (v-shaped) sliding main bearing.

EHD simulations of a pre-dimensioned rigid conical sliding bearing [3] showed distinctive issues concerning edge wear and high contact pressures already in the early design process.

Figure 2. Load taking capability with uniaxial loads

In Figure 2 it is clearly visible by moderate hydrodynamic pressures that a conical sliding bearing is capable to handle axial loads and radial loads applied in the middle of the shaft, as in this case the bearing gap “closes” parallel. However, the bearing is not able to handle radial forces under lever or any bending moments (Figure 2, right) as the shaft tilts slightly within the bearing clearance. This effect is amplified by the short supporting width of the two cones and leads to high contact pressures resulting in edge wear and an early malfunction [5, 6]. Obviously it is essential for a WT main bearing to take all (6-DOF) loads occurring from the wind, so that specific measures are needed to solve this issue.

A flexible designed support structure of the bearing pad allows the pads to follow the movement of the shaft. By a slight deformation of the support, a smooth pressure distribution without edge wear and a long durability is ensured (Figure 3).

Figure 3. Flexible Support structure of the bearing pads

Depending on the actual load distribution on each bearing segment, the support structure will yield with the outer flexibility (2) and the inner flexibility (3) to a combined double flexibility (1). By this a “parallel” and well lubricated gap is ensured.

Once the functional principle of this, “FlexPad” named, sliding bearing has been proved by EHD calculations, a 1MW wind turbine has been chosen as a demonstrator platform to design, simulate and test this bearing in full-scale. Therefore, the hub-loads are calculated by a Co-Simulation of Simpack and Simulink, considering the WT controller and the wind conditions according to IEC 61400 [3,4]. Concerning this loads extensive EHD calculations have been performed in order to investigate the design.
parameters of the bearing and especially the ideal stiffness combination of the “inner” and “outer” flexibility [5, 6].

![Pressure distribution during dynamic simulation of IEC design load case (DLC) 1.1](image)

**Figure 4.** Pressure distribution during dynamic simulation of IEC design load case (DLC) 1.1

Figure 4 shows exemplarily the pressure distribution during a dynamic EHD simulation of IEC design load case 1.1 (NTM - Normal Turbulence Model) at the nominal wind speed of 12 m/s. It is visible, that the hydrodynamic pressure is distributed evenly over the loaded pads without any contact pressures and highest hydrodynamic pressures under 100 bar.

After concluding the design process, the bearing and the related conical shaft is manufactured and ready to be tested on a 1 MW WT system test bench. The left picture in Figure 5 shows the assembled bearing without the shaft and picture in the middle gives a detailed view on the bisected bearing demonstrating the flexible support structure of the bearing pads.

![Pictures of manufactured bearing; bearing on a 1 MW WT machine carrier on System test bench](image)

**Figure 5.** Pictures of manufactured bearing; bearing on a 1 MW WT machine carrier on System test bench
3. Concept-Validation Measurements

In order to investigate the new bearing concept under real conditions, an extensive measuring campaign has been carried out. The right picture in Figure 5 shows the “FlexPad” bearing during the mounting process on the machine carrier of the 1 MW system test bench, where it replaces the original roller main bearing.

This system test bench is equipped with a hydraulic load application unit. This unit is capable to emulate the forces and bending moments resulting from the wind exposed WT rotor and load values can be set according to the results of the hub-load analysis [3, 4].

To validate the new bearing concept, as well as the related EHD-Simulation models, extensive tests from synthetic uniaxial loads (static) to real dynamic loads have been carried out.

3.1. Test Setup

As the “FlexPad”-bearing is designed as a momentum-bearing (similar to SKF Nautilus, Figure 1), it takes all the loads directly without a second bearing point. That is why it is possible to simplify the test setup (see Figure 5). The Gearbox and Generator is dismounted from the machine carrier and the shaft of the main bearing is driven by a servomotor mounted at the gearbox position.

![Figure 6. Test Setup; temperature measurement points, displacement transducer](image-url)

In the test setup the bearing is equipped with a variety of different sensors to investigate the load taking capability under operation. Two temperature sensors are applied just behind the sliding surface of each pad. One sensor at the outer ring and one on the inner ring. This ensures the ability to detect critical thermal situations and gives information about the load distribution over the whole bearing.

In addition six displacement transducers are positioned at six upper pads to measure the deformation of the FlexPad arm under load.

Moreover the bearing is equipped with a torque sensor between the servomotor and the bearing to investigate the friction in the bearing.

During operation force and travel of each hydraulic cylinders is monitored by load cells and position sensors.

The bearing is lubricated with synthetic oil (PAO) of the viscosity index 320. The bottom pads are below the oil level (immersion lubrication) and some of the upper pads are additionally supplied by an external oil supply (pressure less).
3.2. Load-bearing performance

As this bearing design is new and only pure radial sliding bearings or pure axial sliding bearings are common in industry, the characteristics of the load-bearing behaviour are of major interest. Therefore, multiple tests from synthetic uniaxial loads (static) to realistic multiaxial load sequences (static and dynamic) have been conducted.

The temperature sensors, of which two are installed per pad, are best suited to evaluate the load zone. Also under pure hydrodynamic conditions the gap between the shaft and the pad is getting smaller in the loaded zone, which results in more liquid friction and a higher heat input into the pad. Of course also critical operating conditions due to mixed friction or edge wear can be detected at an early stage.

Figure 7. Temperature distribution during uniaxial thrust test (left: rear side of the bearing with pad #H1-H16, right: front side of the bearing with pad #V1-V16, simulation result)

For a better understanding of how the load is distributed over the bearing, simple synthetic loads are investigated at nominal rotational speed (28 rpm). In the case of pure thrust the conical bearing takes the load with the front cone.

The thrust is ramped up in 7 steps up to 101 kN, whereby every step is hold for 10 minutes of operation. Figure 7 shows the temperature distribution, in which the dotted lines indicate the temperatures of each pad at the beginning of the tests ($t_0$; 0 kN thrust) and the continuous lines the temperatures at the end of the last load step ($t_1$; 101 kN thrust). The blue line indicates the temperatures at the outer ring measurement points and the red line shows the inner ring measurement points.
The measurements proof the expectation, that the front pads become warmer as they take the load. Figure 7 shows also that the pads are very equally loaded due to the very similar temperature distribution, which is an indicator for proper position tolerances of the bearing pads. The outer ring measurement points generally become a bit warmer (~2-3°C), as the bearing diameter is larger at this location resulting in a higher circumferential speed. Visible is also the gravity influence on the 1700 kg heavy shaft. In the measurements as well as in the simulation the rear bottom pads become slightly warmer.

**Figure 7.** Temperature Distribution

![Temperature Distribution](image)

**Figure 8.** Temperature Distribution during uniaxial bending moment, Simulation result

Figure 8 shows the next simple synthetic load as pure bending moment is applied to the bearing. The bending moment is ramped up in 11 steps up to 100 kNm, whereby every step is hold for 10 minutes of operation. The continuous lines again show the temperatures of the different measurement points at the end of the last step. The loaded zone changes to top front and bottom rear as expected and predicted by the simulation model. After completing the uniaxial tests, more tests with combined loads were conducted in order to cautious approach to the maximum design load values as follows:

- $M_y + F_x$ (Bending moment + Thrust)
- $M_y + F_z$ (Bending moment + Radial force)
- $M_y + F_x + F_z$ (Bending moment + Thrust + Radial force)
- $M_y + M_z + F_x + F_z$ (static design load)

The measurements showed that combined loads lead to a combined contact pattern. All tests had in common that the load zone became larger at higher loads, but edge-wear never took place during all the tests as neither a critical temperature rise was measured nor any wear was detectable after the tests. Also the measured friction torque increased only slightly in each additional load step, whereas this is explainable with the smaller bearing gap and higher liquid friction due to the higher load.
3.3. Deformation of FlexPad Arms

Since the load-bearing performance under all relevant load scenarios is proofed successfully and edge wear is effectively prevented by means of the flexible support structure, now attention is to be payed to the deformation of the flexible structure.

Two exemplary load conditions are selected for this purpose: The pure bending moment \( M_y \) (left column) is selected, because in this case the minimum number of pads is loaded and the maximum absolute deformation is reached. Even combined loads do not reach this maximum deformation, as the additional load is always distributed over more pads.

The second regarded load condition is a realistic highly dynamic load. Here predictions about the adaptability to changing conditions can be made.

As shown in Figure 9 the deformation of the FlexPad arms is measured by LVDT displacement sensors on the back of six upper pads.

![Figure 9](image)

**Figure 9.** Deformation of FlexPad arm under pure \( M_y \) bending moment (left) and realistic highly dynamic load (right)

The results at pure bending moment show a deformation of 0.7 mm at maximum load measured at the front 12 o’clock pad position (Sensor \( S_{V1a} \)). The Sensors at front 10 and 2 o’clock position (\( S_{V3a} \) & \( S_{V14a} \)) measure with 0.5 mm and 0.4 mm a lower deformation corresponding to the load zone.

According to an FE-calculation a deformation of 1 mm leads to a von Mises stress of 190 N/mm\(^2\) at the most stressed area of the pad supporting structure, which is about 20 % of the yield strength, hence the measured deformation is uncritical and fatigue issues should not occur. Nevertheless, further investigations will be carried out on this topic.

During static operational points the sensors detect also some slight rotational speed-dependent movement (~30 µm), which indicates that the FlexPad bearing adapts also manufacturing tolerances like roundness deviations of the shaft. Especially in comparison to roller bearings where this results in constraining forces between the rollers and the races, this is a big benefit.

The rear sensors (\( S_{H1i}, S_{H13a}, S_{H1a} \)) do not measure any deformation as they are placed on completely unloaded segments.

The measurement under dynamic loads shows that the bearing adapts quickly to new load situations, which is important for the application in wind turbines. Even though calculations concerning the fatigue under long term dynamic loading are yet to be made. The dynamic loads couldn’t be run up to the highest
loads, because the used hydraulic pressure unit wasn’t able to deliver the requested amplitudes of up to 85 kN in combination with the requested frequencies of up to 1.25 Hz. During all measurements the FlexPad arms show a full elastic recovery and entirely uncritical load-bearing and temperature characteristics.

3.4. Mixed Friction limit

Compared to rolling bearings, low rotational speeds are particularly critical for sliding bearings, as the hydrodynamic separation of the surfaces depends on the circumferential speed. Therefore investigations on the mixed friction limit have been conducted at 60 % synthetic thrust ($F_x$) and bending moment ($M_y$). This represents a normal operation point of the considered wind turbine. During the test the rotational speed is reduced from nominal speed (28 rpm) to a stepwise decreased minimum value. In each of the ten steps the minimum value is hold for 10 seconds.

Figure 10 shows the results of this investigations. In the detailed view of the last four steps in the bottom graph it is visible that the measured friction torque drops uniformly with the dropping rotational speed. This is explainable by the lower liquid friction at lower rotational speeds. From step 1 to 9 the lowest friction measured is still getting lower each step but in step 10. In step 10 the rotational speed is ramped down from nominal rotational speed (28 rpm) to 3.6 rpm. Despite the lower rotational speed, the friction torque during the holding phase is not lower than in the previous phase and it also rises during the holding phase, indicated by the dotted black lines. This is an indication that the hydrodynamic pressure build-up is no longer able to completely separate the surfaces, so that mixed friction occurs. At the mean diameter of the conical shaft this speed corresponds to a circumferential speed of 0.127 m/s. Normal operation points are far away from this rotational speed so that full lubrication is ensured under nominal conditions. According to the test setup, there is a roller bearing inside of the load application unit, whose friction torque is also measured by the torque sensor, so that the absolute values shown in Figure 10 are overestimated.
3.5. Wear during WT start ups

Despite a very early transition to hydrodynamic lubrication, wind turbines pass through the mixed friction region during start-ups. According to a SCADA-data analysis of eight 1.5 MW onshore wind turbines over two years, each turbine had about 1000 start-ups per year counted by means of the closing operations of the circuit-breaker.

In order to be prepared for this circumstances, which are incidentally quite harsh for sliding bearings, special sliding materials were developed and investigated \[3, 4, 6, 7\]. New thermal spraying coatings have been developed which are featured, for example, by solid lubricants incorporated into the material matrix. Besides this technology enables the coating of complex surfaces, like the conical surface of the sliding pads.

To investigate the properties of the bearing and the coating during the start-ups, this conditions were simulated on the 1 MW system test bench. Start-up loads and the rotational speed ramp according to the load simulation were applied to the bearing in repetitive. The cycle is truncated at 7.5 rpm due to cycle time reduction, as it was proofed before, that a higher rotational speed is uncritical.

![Figure 11](image)

**Figure 11.** Start-up loads (top left), exemplary cycle showing friction-torque and the rotational speed (top right); oil sump temperatures during all 12 tests (bottom)

In the top right graph in Figure 11 one exemplary cycle showing friction-torque and the rotational speed ramp. As soon as the rotational speed ramp starts, the measured friction torque increases rapidly. In this phase a higher speed results in more friction by reason of asperity contact. From a speed of 1.6 rpm, the hydrodynamic separation of the surfaces begins and the friction torque drops again to the level of nominal operational points.
The lower 4 graphs show the series of maximum values of each cycle for the 12 conducted tests, which were run either with 1000 or 500 Cycles in a row. For a better perceptibility the first six and the last six tests are shown in different graphs.

It can be seen that the peak values of the mixed friction are very similar over the test cycles apart from test one and the beginning of test 2. In this two cycles probably running-in and surface adaption processes have taken place. In following ten cycles, the difference in the curves can be explained by different ambient conditions. E.g. in the bottom graphs, the oil sump temperature at the beginning of the tests is slightly different or in test six the gate to the test hall have been opened for a longer duration (winter, 5°C outside temperature). This directly correlates with the measured friction values. In total 8000 start-up cycles have been completed, which corresponds with about 40 % of the lifetime start-ups of a wind turbine. After the tests the bearing has been disassembled and the optical impression of the bearing surfaces corresponds with the measurement results. The highest loaded bearing pads showed a slight smoothing of the surfaces but without excessive wear.

4. Summary

This contribution shows an innovative approach concerning a new type of rotor main bearings for wind turbines. The conical sliding bearing design with its flexible support structure of the bearing pads combines the benefits of established double row tapered roller bearings with the benefits of sliding bearings. When a bearing fault occurs, for example, the bearing pads are replaceable individually without the need of disassembling the drivetrain, in contrast to roller main bearing applications. In order to investigate the new bearing concept under real conditions, an extensive measuring campaign has been carried out on a full-scale 1 MW system test bench. Among other matters the load-bearing properties, the mixed friction limit and the start-up behaviour have been investigated. All tests carried out delivered very promising results. Furthermore the described problem of edge wear is successfully solved by the innovative flexible support structure of the bearing pads.

Acknowledgments

The authors would like to thank the German Federal Ministry of Education for the financial funding and the possibility to do research on this topic.

References