Feasibility study for the use of hydrodynamic plain bearings with balancing support characteristics as main bearing in wind turbines

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Abstract. The failure of a wind turbine’s main bearing leads to high maintenance costs which affects the economic competitiveness of wind energy as renewable energy source. Therefore, research is conducted to replace today’s used roller bearings by plain bearings. These can be designed segmented so that in case of a failure the segments can easily be exchanged on tower. As part of a former research project a 1 MW plain bearing with conical sliding surfaces was developed and tested on a full-scale system test bench. This contribution presents a methodology how this concept can be developed further and optimized to design a bearing in 3-point suspension or a moment bearing for high power classes.

1. Motivation and objectives
The drivetrains of wind turbines are manifold and demand different characteristics of their main bearings. To tackle these demands, various main bearing concepts have been established in the wind industry, all of them having their specific conceptual capabilities, advantages and disadvantages. However, all these concepts are implemented with rolling bearings, which must be replaced at great expense in the event of failure. In order to increase the availability of the turbines and to reduce the maintenance costs, plain bearings get in the focus of research and industry due to their advantage to be designed as a segmented bearing. This allows an exchange of the bearing segments on tower. As a result of the cost advantages in the maintenance of plain bearings, the use of plain bearings as main bearings in wind turbines is currently an important topic in the wind industry. However, there are only few publications on this topic available. Jonuschies investigates the potential and application limits of radial plain bearings as main bearings [1]. Thomsen develops numerical models for the simulation of hydrodynamic bearings and points out the significance to allow an alignment of the bearing segments to the shaft [2]. This becomes evident when designing a plain bearing for the main bearing of a wind turbine because due to the weight of the rotor and high bending moments the shaft inclines during operation. Therefore, in the former research project “WEA-GLiTS”, that was carried out at the Chair for Wind Power Drives (CWD), a plain bearing concept, designed as moment bearing, was developed.
and experimentally validated. The alignment of the segments is achieved via a double-flexible segment support structure [3].

Following the work of the WEA-GLiTS project, the objective of this article is to present a methodology on how hydrodynamic plain bearings for use as main bearings in wind turbines can be designed for different bearing concepts using segmented pads and balancing support structure characteristics.

Within the framework of the methodology an elasto hydrodynamic (EHD) simulation is carried out within the scope of a feasibility study for a preliminary bearing design in a three-point suspension for the 2.75 MW research turbine of the CWD. Since the elastic deformation of the sliding segments also changes the lubrication gap geometry, this has a major impact on the hydrodynamic pressure built-up and the load distribution in the bearing [4,5]. For this reason and especially for the application in wind turbines, the elastic deformation must be taken into account in the design process. With the results of the EHD simulation the preliminary bearing design can be evaluated with regard to the load distribution. Furthermore, the system behavior can be analyzed and the relevant influencing parameters for optimizing the bearing design can be derived.

In a further section, a bearing design of a moment bearing for large turbine sizes (up to 6 MW) with the double-flexible segment support structure is presented to which the presented methodology can be applied. These bearings offer the advantage of a compact design. Especially due to the complex and expensive production of such roller bearings [6], a hydrodynamic plain bearing is considered as a cost-effective alternative for a big diameter rotor main bearing. In addition to that a concept with reduced complexity and applicability for numerous drive train concepts is presented, namely the spherical pad bearing design. Based on the consequent further development and taking into account the technical and economic needs of several wind turbine drive trains, the two presented concepts offer promising plain bearing solutions for rotor main bearings.

2. FlexPad Bearing Concept

The function of the main bearing of a wind turbine is to transmit the non-torque loads of the main shaft from the wind field to the machine carrier. Because of the high bending moments, the shaft tilts within the bearing. If a plain bearing is used as the main bearing whose sliding segments are rigidly mounted on the bearing housing, the segments cannot adapt to the displacement and deformations of the shaft and locally small lubrication gaps occur. These small gaps located at the outer edges of the bearing lead to high pressures (edge loading) and possible failure of the bearing if no sufficient break-in can be assured. To avoid edge loading a double flexible segment support structure was developed in the research project WEA-GLiTS. Figure 1 illustrates the functionality of the FlexPad called flexible arm support structure. In this concept the plain bearing segments are mounted on a special designed flexible arm. Due to the external and inner flexibility of the arm, the segment can perform a parallel displacement and follow the displacement of the shaft (Figure 1, 3) which prevents the occurrence of edge loading.

![Figure 1. Principle of the double flexible segment support structure[3]](image)

The geometry of the flexible arm has amongst other parameters (arm length, inlet contour etc.) a major influence on the pressure distribution. Figure 2 shows the simulated pressure distributions of a FlexPad moment bearing with two different arm geometries. The bending stiffness of the arm of bearing A does not allow a proper alignment of the segment and high pressure peaks occur. In comparison the arm
stiffness and the inlet contour of each segment of bearing B to the right are well designed and the hydrodynamic pressure is more evenly distributed for this bearing. Finally, the functionality of this bearing concept and its patented support structure has been validated in experimental tests with 1 MW bearing demonstrator (Figure 3).

Figure 2. Pressure distribution for two bearing designs

Figure 3. 1 MW FlexPad bearing demonstrator [3]

3. Methodology

The objective of this article is to present a methodology to transfer the innovative double-flexible support structure to other main bearing concepts. This will be demonstrated by a feasibility study based on EHD simulation for a preliminary design of a bearing in three-point suspension. In particular, the model structure and the level of detail of the EHD model will be discussed in this section.

Bearing design for three-point suspension

In order to transfer the double flexible support structure to a plain bearing in three-point suspension the first step is to analyze the drivetrain system. Figure 4 shows a schematic sketch of a drivetrain of a wind turbine in three-point suspension. In this configuration, the wind loads are absorbed at three points, namely the main bearing and the torque arms on both sides of the gearbox. In today’s wind turbines double-row self-aligning spherical roller bearings are used as main bearing [6]. These bearings are designed with a spherical raceway due to the deflection of the shaft. Since in a three-point suspension the non-torque loads are absorbed in the gearbox as second bearing point, the gearbox performs a small vertical and horizontal displacement. In case of designing a plain bearing for this arrangement, the displacement of the gearbox has a great influence on the load distribution in the main bearing and needs to be considered in the EHD simulation.

Figure 4. Schematic sketch of a drivetrain in three-point suspension
For this boundary condition a preliminary design of a conical FlexPad bearing was derived (see Figure 5). The geometry of this preliminary design is derived from the geometry of spherical roller bearings. This double conical bearing design has a mean diameter of 890 mm and consists of 32 equal pads, 16 on each side, that are connected to the bearing housing via the previous described double-flexible support structure.

By designing the number of bearing segments two contradictory effects need to be considered. On the one hand the number of segments cannot be too low because in this case some segments are just partially located in the loaded zone and a parallel displacement of the bearing segment cannot take place. This enables the occurrence of high-pressure peaks (edge loading). On the other hand, an increasing number of bearing segments leads to higher manufacturing and assembly costs. In the former research project WEA-GLiTS simulations have shown, that 16 pads at each cone are a good compromise to fulfil the mentioned requirements. Finally, the optimal number of pads is dependent on numerous aspects, e.g. bearing size, operating conditions, manufacturing and assembly possibilities.

Figure 5. Preliminary design of a FlexPad bearing for a three-point suspension

EHD-Model

In order to conduct an EHD simulation for the presented preliminary design in Figure 5 first a multi-body simulation model needs to be set up. This model consists of two flexible bodies, namely the main shaft and the main bearing. To reduce the number of degrees of freedom (DOF) a mixed modal/static reduction is performed. For the shaft in total 50 modes are considered in the simulation. For the plain bearing segments, a staggered reduction is conducted. During this procedure, the modes are calculated one after the other for each segment. This ensures that there is no need to consider an extremely large number of global modes for the deformation of each segment and for each segment the same number of DOF is considered. The hydrodynamic pressure is calculated by solving Reynolds' differential equation in each time step with automatic step size control. For the EHD simulation the software FIRST developed by IST is used.

The influence of the gearbox displacement is considered by spring elements. The spring characteristics allows a maximum displacement of the shaft of 1.5 mm in radial direction. This displacement is in the same order of magnitude as the gearbox displacement that was measured on CWD’s system test bench for the 2.75 MW research turbine with three-point suspension [7]. However, this model approach is a simplification, which is considered sufficient for the analysis here.

The static load case for this simulation is derived from multibody simulations for a 1 MW turbine in power production operation according to the specifications of the IEC 61400 standard [8]. These simulations were conducted from the cut-in wind speed to the cut-out wind speed and the maxima of the 95% percentiles for each load (Fx, Fy, Fz, My, Mz) were combined to a static load case. The loads were
scaled with a scaling factor up to this turbine size. This procedure is sufficiently accurate for the feasibility study in this paper, because for the evaluation of the load distribution it is not the exact level of the loads but rather the combination of the individual non-torque loads that is important. The input parameters are summarized in the following table.

| Fx [kN] | Fy [kN] | Fz [kN] | My [kNm] | Mz [kNm] | n [1/min] | Oil         | T [°C] |
|---------|---------|---------|----------|----------|-----------|-------------|--------|
| 151.65  | 9.02    | -130.58 | 150.52   | 119.41   | 18.7      | ISO VG 320  | 45     |

4. Results

As result of the conducted simulation Figure 6 shows the calculated hydrodynamic pressure distribution for the geometry shown in Figure 5 in a three-dimensional plot. For a better clarity just the hydrodynamic pressure distribution on each segment without the shaft and the bearing are shown. First off all it can be concluded that no pressure peaks due to edge loading occur which is an indication that the sliding segments can perform a properly parallel displacement.

The left plot shows the hydrodynamic pressures from the view in downwind direction and reveals the loaded zone of the bearing. Due to the radial force $F_z$ in negative $z$-direction the bottom segments are more loaded than the upper ones. Furthermore, due to the thrust load the shaft moves in axial direction so that the gap between the shaft and the pads of the rear cone closes. Consequently, all pads of the rear cone are loaded and act as an axial bearing. The maximum pressures are around 80 bar what can be carried by most of the common used plain bearing materials [9].

The plot to the right in Figure 6 which displays the hydrodynamic pressures from a side view shows that for this design just the segments of the rear cone are loaded. The pads of the front cone are all unloaded. This phenomenon is known from double-row spherical roller bearings where the bearing row on the gearbox side carries more load than the one on the rotor side. The reason for this pattern is the axial displacement of the shaft with the consequence that the gap between the front pads and the shaft increases so there is almost no hydrodynamic pressure build-up. Figure 7 shows the three-dimensional gap for the simulated bearing. The gap of the front cone is larger than 5 mm which is much too high for a hydrodynamic pressure build-up.
The axial displacement of the shaft is facilitated by the geometry of the bearing housing. Figure 8 reveals that the bearing housing deforms due to the axial load. A stiffer design of the housing would counteract this behavior.

A closer look on Figure 8 shows that the rear segments deform more than the bearing housing due to the axial load. An increase of the stiffness of the bearing support arms and a smaller bearing clearance would reduce the axial displacement of the shaft. A further step would be the design of an asymmetrical bearing with a higher inclination angle of the back cone what could be beneficial for the load distribution between the two cones. This approach is known from spherical roller bearings and is a topic in current research [10,11].

In the further procedure it would be interesting to see how the optimized bearing design performs in other operating conditions (start-up, idling). For this purpose, further simulations are necessary, whereby the material properties of the bearing material may also have to be considered.

5. Moment Bearing

After demonstrating the functionality of the FlexPad concept in demonstrator tests for a small 1 MW turbine in an earlier project, the next step is to transfer this bearing concept to moment bearings for larger plant sizes (up to 6 MW). In comparison to other bearing concepts the moment bearing offers the advantage of a compact design. This compact design is achieved by a large shaft diameter, which leads to high demands on manufacturing accuracy and high costs for the currently used roller bearings. In contrast to roller bearings, these disadvantages do not apply to plain bearings in such an extent, since plain bearings can be segmented. The preliminary design of a segmented plain bearing in moment design is shown in Figure 9. This bearing concept was inspired by commercially available rolling bearings in moment design (e.g. SKF Nautilus), which are used in wind turbines up to 6 MW [12].
This design consists of a non-rotating outer ring, the segments mounted on the outer ring and the rotating hollow shaft. The plain bearing segments are designed to ensure parallel displacement of the sliding surface under axial and radial loads as well as under bending moments. The large outer diameter of the rotor shaft causes high circumferential speeds, which has a positive effect on the hydrodynamic pressure build-up. Furthermore, the bearing is designed in such a way that the segments can be exchanged without disassembling the rotor shaft.

Currently this bearing has the status of a concept study. For evaluation, further development and to validate the use of the double-flexible segment support structure for this bearing type, EHD simulations with corresponding rotor loads for this turbine size must be carried out in the same way as shown for the bearing in three-point suspension. With these results, the segment supports can be dimensioned and designed with the aim of an optimum load distribution. In addition, fatigue calculations for the design of the sliding segments should be carried out.

6. Spherical pad bearing

Based on the promising results of plain bearings as solutions for rotor main bearings and the successful above presented feasibility study, the spherical pad bearing represents a further development of the FlexPad bearing concept including the advantage of a reduced number of parts. The deflection of the shaft can be compensated by rotational alignment of the inner pad-ring while still allowing the support of radial and axial loads with reduced radial shaft displacement. The hydrodynamic characteristics and load carrying capability can be optimized by adjusting the micro geometry of the spherical bearing which is producible with state-of-the-art manufacturing tools. By reducing complexity and manufacturing afford the in Figure 10 depicted preliminary design study represents a promising economical and technical feasible solution for multi-point drive train suspensions. The within a feasibility study conducted EHD simulations of a multi-MW drive train indicate a promising hydrodynamic load carrying capacity under combined radial and axial loads, as indicated in Figure 11.
7. Conclusion and Outlook

In this paper an innovative double-flexible segment support structure for plain bearings as main bearings in wind turbines was presented and approaches to the application of this support structure for different bearing concepts were discussed. For this purpose, in a feasibility study a preliminary design was developed and it was shown that this segment support structure can also be used for a three-point suspension without the occurrence of edge loading. With the help of an EHD simulation the hydrodynamic pressure distribution in the bearing could be calculated and analyzed. Furthermore, the load transmission and deformation of the bearing could be evaluated and approaches for the optimization of the bearing within the design process could be determined. Moreover, an outlook for the design of a moment bearing for large turbine sizes with such a double-flexible segment support structure was presented.

The preliminary studies in this paper show as a key result that, based on the 1 MW prototype, the double flexible segment support structure can also be used for main bearings in a three-point suspension as well as moment bearing in the 6 MW power class. Regarding the moment bearing design further simulations needs to be conducted to confirm the applicability of the double flexible support structure to this concept. In addition to that, a further development of the evaluated FlexPad, the spherical pad bearing concept, and its feasibility was presented. The compact design reduces complexity and represents versatile solution for numerous drive train concepts of wind turbines.

Based on the results in this paper, these concepts can be further developed. For this purpose, in the case of the bearing for the three-point suspension, the angles of inclination of the front and rear cone as well as the stiffness of the segment support structure and other bearing parameters such as bearing clearance can be optimized. Furthermore, the design and simulation of a bearing with only segments in the loaded zone in order to save costs is worth to be a topic for further investigations. For the preliminary design of the moment bearing, EHD simulations must also be carried out to evaluate the load distribution and deformation of the bearing. After the principle feasibility study of the spherical pad bearing was proven, the further design development is at hand.

Finally, cost estimations of these concepts are necessary in order to compare them with today’s used rolling bearings. However, at this stage of development, there is not sufficient data available regarding manufacturing costs and number of units in order to make a valid cost estimate and then compare it with the series product rolling bearing.
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