Plain bearings for wind turbine gearboxes

An update on experience


Abstract

As a leading gearbox supplier in the wind business ZF is continuously investigating and optimizing core gearbox technologies. As such ZF is investigating alternative bearing solutions, i.e. hydrodynamic plain bearings. With this paper ZF presents an update on its technology development activities and shares experiences into the public domain to support initiatives like VDMA, to prepare a standard for plain bearings used in main gearboxes of wind turbines. Starting from a selected set of plain bearing designs assembled on the most significant locations in a wind turbine gearbox and tested according relevant wind turbine conditions. This paper will share insights in the behavior of different plain bearing designs by using standardized calculation methods, like DIN 31562, to benchmark initial working conditions of these bearing concepts.

Introduction

Wind energy represents a major part of the actual energy mix. The major part of wind turbines is equipped with a geared drive train. Current wind turbine gearbox designs - in the power range from 2 up to 6 MW - typically consist of a combination of planetary stages with at least one high speed helical gear stage. The majority of state-of-the-art designs are equipped with roller bearings. Plain bearings have decade's long track records in applications such as turbo machinery, marine and automotive. The introduction of plain bearings into wind turbine gearboxes has a number of drivers like reliability, torque density increase and other potential benefits like noise reduction. During the development of ZF’s plain bearing program and many supplier discussions, the need for a standardized praxis for a plain bearing design guide for wind turbine gearboxes became clear. A decent listing of important attention points for plain bearing designs and a clarification of relevant wind turbine operational and loading conditions have been already started in the VDMA consortium. By this paper ZF would like to share insights regarding tested bearing designs and their behavior during different operational conditions.
By applying representative load cases during gearbox test runs, the bearing behaviour was parameterized by measuring oil supply rates and bearing temperatures, recording noise and vibrations levels and monitoring the lubricant cleanliness condition. These test results, together with the specific bearing design, the shaft location and loading conditions, are used as input for a journal bearing calculation to evaluate the behaviour by parameters like lubricant film thickness and friction power. The outcome of the investigation leads to an evaluation procedure based on the parameter and a boundary definition for an operational condition, e.g. where the bearing reaches unfavourable lubrication regimes.

**Plain bearing designs**

In several 2MW gear units ZF installed different designs of plain bearing solutions, applied on different shaft locations to evaluate for each the functionality by performing a representative test protocol. The sectional drawing of a wind turbine gearbox using plain bearings shows the applicable locations. The table highlights the different applied bearing concepts.

![Fig. 1: ZF Wind Power 2MW plain bearing gearbox](image)

<table>
<thead>
<tr>
<th>Shaft position</th>
<th>Plain bearing design</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Concept I</td>
</tr>
<tr>
<td>Planet</td>
<td>X</td>
</tr>
<tr>
<td>LSIS</td>
<td>X</td>
</tr>
<tr>
<td>HSIS</td>
<td>X</td>
</tr>
<tr>
<td>HSS</td>
<td>X</td>
</tr>
</tbody>
</table>

Different design concepts were worked out and rated regarding risk, functionality and cost, leading to three main design concepts. In the following table these designs are characterized into two groups for the parallel stages, as high speed (HSS), high speed intermediate (HSIS) and low speed intermediate shafts (LSIS), and for the planetary stages as the planet position.

For the parallel stages conventional materials like white metal are used for the first two designs. More advanced multi-layer materials are chosen for the third design. These
materials have a higher load caring capacity and are thereby resulting in a more compact bearing.

Table 1: Concept overview for radial loaded HSS and HSIS bearings

<table>
<thead>
<tr>
<th>Concept</th>
<th>Characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>I &amp; II</td>
<td>B/D: 0.5 - 1, Specific loads: 8 – 15 MPa, Lubricant film thickness: 5 – 6µm, Material: white metal on steel backing, State-of-the-art: turbo-machinery</td>
</tr>
<tr>
<td>III</td>
<td>B/D: 0.3 – 0.6, Specific loads: 15 – 35 MPa, Lubricant film thickness: 2 – 3µm, Material: multi-layer coating on steel backing, State-of-the-art: combustion engines/ crankshafts</td>
</tr>
</tbody>
</table>

The main difference between the two designs is a higher power density, realised by selecting a different B/D ratio and for some positions a different material. The planets are radially beared by a floating sleeve, which is a bronze sleeve that can freely rotate between the planet gear and the planet shaft. This bearing creates two lubrication films, one between the planet gear and the floating sleeve (outer film) the other between the floating sleeve and the planet shaft (inner film). The sleeve itself is driven by the fluid friction of both inner and outer lubrication film, resulting into a speed range between planet speed and 0 rpm.

Table 2: Concept overview for planetary bearings

<table>
<thead>
<tr>
<th>Concept</th>
<th>Characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>I &amp; II</td>
<td>B/D: 0.7 – 1.4, Specific loads: up to 20 MPa, Lubricant film thickness: 4 – 6 µm, Material floating sleeve: bronze material, State-of-the-art: cement mills, wheel drives in construction machinery</td>
</tr>
</tbody>
</table>
Wind turbine operational conditions

Plain bearings are historically used in applications that are running with only a limited set of operating points. This is in large contrast with a wind turbine application. To design and validate a reliable plain bearing solution it is from utmost importance that all different working conditions for the plain bearings are investigated further.

To define the correct test protocol, typical wind turbine conditions as mentioned in the torque speed duration distribution (TSDD), are considered. The TSDD describes for every torque-speed relation the occurrence rate.

Fig. 2: An indication of a torque speed duration distribution for a wind turbine

The TSDD above shows mainly 3 different areas of application. The first area, at the left bottom corner, loads and speed are limited. In the second area the turbine starts up and tries to connect onto the grid or the rotor is slowing down when the turbine is going to stand still and is already disconnected from the grid. This is an operational area where speed is available without any torque. At the last part of the TSDD, also known as the power curve or operational mode, the speed of the turbine drive train is sufficiently high to start generating power. In this area at the right upper corner, the mechanical loading condition occurs which is defined with nominal load and speed condition including the drive train efficiency. Short temporally conditions such as high overloads, short overspeeds, high loads at limited speeds are not shown on this example of the TSDD but can happen and special attention during the design phase is recommended. These last conditions are not taken into account in this analysis.
Table 3: Example of a load revolution distribution (LRD)

<table>
<thead>
<tr>
<th>Turbine Mode</th>
<th>Bin</th>
<th>Load % to $T_{mech}$</th>
<th>Speed % to $n_{nom}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Generation</td>
<td>1</td>
<td>140%</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>2</td>
<td>120%</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>3</td>
<td>112%</td>
<td>97%</td>
</tr>
<tr>
<td></td>
<td>4</td>
<td>100%</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>5</td>
<td>92%</td>
<td>101%</td>
</tr>
<tr>
<td></td>
<td>6</td>
<td>66%</td>
<td>100%</td>
</tr>
<tr>
<td></td>
<td>7</td>
<td>46%</td>
<td>96%</td>
</tr>
<tr>
<td></td>
<td>8</td>
<td>31%</td>
<td>96%</td>
</tr>
<tr>
<td></td>
<td>9</td>
<td>10%</td>
<td>52%</td>
</tr>
<tr>
<td>Start-up/Stop</td>
<td>10</td>
<td>0% (1)</td>
<td>48%</td>
</tr>
<tr>
<td></td>
<td>11</td>
<td>0% (1)</td>
<td>24%</td>
</tr>
<tr>
<td></td>
<td>12</td>
<td>0% (1)</td>
<td>6%</td>
</tr>
<tr>
<td></td>
<td>13</td>
<td>0% (1)</td>
<td>3%</td>
</tr>
<tr>
<td></td>
<td>14</td>
<td>0% (1)</td>
<td>1%</td>
</tr>
</tbody>
</table>

(1) only dead weight considered no torque

The load-speed data as input for the assessment is retrieved out of the TSDD using different bins as mentioned in the table aside. As an alternative, the load – speed bin data of a load revolution distribution (LRD) as mentioned in IEC 61400-4 [4] can be taken. Such an LRD is generated out of the TSDD using the “Miner” method [4] and represents in a simplified way the power curve of the wind turbine.

**Testing results**

All different bearing concepts mentioned in table 1 and 2 were build-in into different gearboxes and tested on the ZF back- to-back test rigs. In the test protocol the actual operation conditions relevant for a turbine were applied, like the complete power curve including overloads till 140% and transient events like low and high speed idling, as indicated in table 3. Further additional events such as overspeeds till 120%, emergency stops, operation at oil sump warning limit resulting in low oil viscosities and cold start-ups have been also taken into account and resulted in a test time of at least 300hrs. During these tests, parameters like oil supply and bearing temperatures as well as pressures were monitored for the different operational conditions, and will be used as input for the calculation assessment.

After the tests, the gearboxes were partially or completely disassembled and the components are inspected and re-evaluated by measurements. For several bearings smoothing effects of the surface, caused by running-in, were observed and proved by roughness measurements. Nevertheless no wear was detectable by evaluating the geometrical measurements performed before and after testing. On all specimens, differences seen in topography were neglectable.
Calculation assessment

To estimate the lubrication thickness and the coefficient of friction the calculation model according to DIN 31652-1 [1] is used. This method is chosen because equations mentioned in DIN 31652-2 [2] are well documented and therefore the outcomes and trends can easily be used as a benchmarking in the public domain. The input data of the calculations is based on test measurements gathered during representative operational conditions for the different plain bearing locations and concepts. An additional lubrication film parameter is taken up in the evaluation and is highlighted in the next chapter.

Evaluation parameter: Film thickness ratio (Lambda)

The distribution of speed-torque combinations on wind turbine level leads to different operating conditions characterized by the combination of viscosity, velocity and load. On each bearing location these parameters have an influence on the behavior of the plain bearing resulting in film thickness build-up and in friction torque. Depending on the operation condition, a bearing can run in different lubrication regimes. The most desired regime is the hydrodynamic (HD) lubrication regime where a full fluid lubrication film build-up is reached and no asperity contacts occur. Unfortunately this regime is not always reached for each wind turbine operational condition and for every bearing arrangement. Nevertheless for mechanical operational conditions the plain bearings will operate in the hydrodynamic lubrication regime.

The typical lubrication regimes for a sliding contact are already several times discussed in the public domain as for example in [5] where a categorization is made in 4 different regimes known from the Stribeck curve. As a variant on the Stribeck curve, where on the abscissa typically the dimensionless Hersey number is used which is a function of the absolute viscosity, the rotational speed and pressure, a dimensionless film thickness ratio lambda (λ) can be used as suggested in [5]. This film thickness ratio (λ) is a function of the minimum film thickness (h_{min}) and the roughness of both sliding surfaces.

\[
\lambda = \frac{h_{\text{min}}}{\sqrt{R_{q,1}^2 + R_{q,2}^2}}
\]

Where, \( R_{q,1} \) = rms surface finish of journal
\( R_{q,2} \) = rms surface finish of bearing

The film thickness ratio (λ) will be used to evaluate the bearing behavior to the below mentioned lubrication regimes. As it is expected aswell for hydrodynamic and elastohydrodynamic (EHD) lubrication regimes, no or limited wear of the bearing components

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will occur. These lubrication regimes are required to obtain an optimal operational condition of a plain bearing. According to this method the different lubrication regimes can be roughly categorized as proposed in figure 3.

![Stribeck curve](image)

Fig. 3: Lubrication regimes indicated on the Stribeck curve using the film thickness ratio ($\lambda$).

**Film thickness calculations**

For the different design concepts, the lubricant film ratio is calculated incorporating load bins according table 3 mentioned as power generation conditions.

![Graphs of lubricant film ratio](image)

Fig. 4: The calculated film thickness ratio ($\lambda$) for different operational conditions.
A speed range from grid connection till the nominal operational speed (from 52% till 104% \( n_{\text{nom}} \)) and operational torques between 10% till 140% of \( T_{\text{mechanical}} \) are used.

The graphs in figure 4 show the influence of these conditions onto the film thickness ratio (\( \lambda \)).

The film thickness ratio (\( \lambda \)) is the lowest for \( T_{\text{mechanical}} \) and overloads based on the analysis.

The film thickness is increasing when the speed and corresponding load are decreasing.

The concepts showing the highest potential for torque density increase are operating under the lowest film thicknesses. At mechanical load level, the lambda values are around 3 till 5 for concept III and are increasing for Concepts I and II with values larger than 15.

In the next overview, a Stribeck curve is generated per bearing position using the operational points of table 3. In de arched zone at partial and EHD regime, the method of DIN 31652-1 [1] does not take these lubrication regimes into account and therefore calculated friction coefficients will vary from the actual values.

Similar conclusions for the curves in figure 5 can be made because concept II and III will operate much closer to the partial lubrication regime compared to concept I design which always operated in the HD regime.

As mentioned already, the lubrication film is increasing when speed and the corresponding load are decreasing while running down the power curve and results in good hydrodynamic conditions until the point of grid connection. When the turbine is disconnected from the grid,
the torque will disappear and the bearings will only be loaded by dead weight of the shaft and gearing components. In order to check the critical operational conditions at idling and start/stop modes, the speed is further lowered till the lubrication film ratio is back near the partial lubrication regime. Calculations are performed for the different bearing locations using a bearing temperature of 60°C which is conservative but still relevant in such operational conditions.

![Graph showing lubricant film ratio (λ) at idling conditions](image)

**Fig. 6**: The calculated film thickness ratio (λ) for idling and start/stop conditions

Out of the results can be concluded that from a generator speed of approximately 20RPM and above, the lubricant film ratio is sufficiently raised to run into the hydrodynamic lubrication regimes. For the HSS and the planet outer film, these regimes are already reached at a generator speed of approximately 10RPM. Below the mentioned speeds, the bearings are running in partial lubrication regimes where the specific pressures are still within the prescribed experience values of 2.5MPA according to DIN 31652-3 [3], so wear shouldn’t be an issue in these conditions.

**Summary**

When the behaviour of a plain bearing is evaluated, it is noticed that the film thickness ratio is the lowest in the range of mechanical loading for normal operational conditions. For operational conditions situated on the power curve between cut-in and nominal operation, the film thickness ratio (λ) is sufficiently high enough with the result that the plain bearings are operating in the hydrodynamic lubrication regime. Local contact wear during start-up and stop events will be limited, because the available speeds are sufficient to bear the dead weight of the geared components and shafts. Idling at HSS speeds lower than 20 RPM; the film thickness ratio is decreasing and the bearings are starting to operate into the partial
lubrication regime because the sliding speeds are getting too low. At these conditions asperity contacts start to occur but due to the low speeds the frictional losses are also low and very limited amount of wear could be expected.

The most critical loading conditions in respect to the film thickness ratio ($\lambda$) are at $T_{\text{mechanical}}$ and during overload events where sliding speeds are expected to be normal or even higher. This results in high friction losses for the journal bearings. The tested samples did not reveal wear nor damage even when the bearing operated continuously for several hours at torques till 140% of the mechanical load level.

As an evaluation rule for plain bearings running in proper lubrication regimes, the lubricant film ratio lambda shall be evaluated at mechanical operational conditions. For this condition a critical lubricant film ratio can be proposed: $\lambda_{\text{critical}} = 4$.

A further improvement can be made when the journal is sliced in a certain amount equally sectioned lamina where for each a corresponding lubricant film ratio $\lambda_i$ can be calculated. For each lamina $\lambda_i$ should be higher or equal to $\lambda_{\text{critical}} = 4$. With this method shaft misalignments and journal bending can be taken into account. This is not implemented in the DIN 31652-1 [1] but can be calculated by more advanced rating tools.

For bearing concepts using materials which are less sensitive to the operation near mixed friction lubrication regimes, $\lambda_{\text{critical}}$ could be lowered to 3. Nevertheless operating in such regimes should be thoroughly evaluated because the risk of (local) asperity contact becomes high.

To obtain a good plain bearing design, additional sanity checks need to be carried out for transient operational conditions and events as there are, overload and overspeed, high loads at limited shaft speeds, cold start up,....

References