Novel Attempts for Plain Bearing Solutions in Wind Turbine Drivetrains
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Abstract
Gearbox failures cause a significant amount of downtime hours and maintenance costs in wind turbines. In most cases, bearing failure is the root cause of gearbox breakdown. Alternative bearing technologies such as plain bearings have the potential to remarkably improve turbine durability. Hence, there is a lot of development activity in the area of plain bearing solutions for wind drivetrains. In cooperation with a gearbox producer and a start-up company that develops direct drive turbines, Miba launched an R&D project on the development of plain bearings for the rotor.

This paper will respond to typical reservations against plain bearings and explain Miba’s basic approach, the simulation tools applied, the type of bearing material tests performed and the results and findings. The paper also describes how reverting to a wide range of multilayer material compositions and sophisticated surface technology helps to cope with the high requirements on bearings. Finally, an outlook on solutions for gearbox bearings and momentum rotor bearings and the corresponding validation programs is presented.

Bearing Failures
Various approved surveys (Figure 1) prove that gearbox failures cause a significant amount of downtime hours and maintenance cost in wind turbines. As a result, the cost of energy (CoE) increases, substantially reducing the profitability of wind turbines. In most cases, bearing failures are the root cause of gearbox breakdown. Alongside improved roller bearing technology, advanced gearbox and turbine concepts and sophisticated control systems, alternative bearing technologies have the potential to remarkably improve turbine durability.

Figure 1: Failure rate and downtime of wind turbine components (source: Durham University, presented at EWEC 2010)

Roller element bearings (REB) are state of the art for gearboxes and for rotor bearings in wind turbines. They are sensitive to shocks, excessive load, downtime, micro movements and vibrations. REB in smaller gears have become more reliable over the years, thereby increasing lifetime. However, distortions have an exponential effect on peak loads relative to size and the small contact area of the rollers leads to non-permissible Hertzian stresses. But oversized bearings in conjunction with a wide load profile result in slippage of roller elements at moderate loads and during starting periods. The consequence is micropitting and smearing on the surface of the raceways. Complimentary downtime, micro movements and vibrations result in false brinelling. All this pre-damage agglomerates and propagates under high loads/cycles resulting in bearing failure well before the calculated fatigue lifetime is reached.

Hence, the lifetime in the multi-MW segment is only seven years on average and even less in many cases. Near/off-
shore applications will again pose a big challenge for REBs. Further upscaling is necessary and accessibility for servicing is limited. In consequence, reliability becomes even more important. Considering that all REB-specific failure modes are not relevant for plain bearings, it is obvious that this bearing technology demonstrates high potential in the multi-MW segment in terms of making drivetrains more durable and reliable.

**Plain Bearings**

**Status Quo**

Hydrodynamic plain bearings have been state of the art in large diesel engines for decades. These bearings have to cope with high dynamic loads and numerous starts and stops (dry friction) in combination with high temperatures and abrasive dirt particles. Loading, increasing demands on service lifetime and other specific requirements such as corrosion are continuously increasing. In contrast to REBs, there is no fatigue lifetime calculation available based on the effects of alternate loading and diminishing factors for all kinds of operating conditions. Engineering, design and calculation work, including lifetime estimation for

program engineering (CE) model that strongly considers the specific application and operating conditions. Miba's CE approach features an engineering environment based on three major areas: load calculation and oil film simulation, bearing material testing and field testing. In the course of a comprehensive lifecycle program, both test rig bearings and field engine bearings are being investigated on a systematic schedule in metallographic and chemical laboratories. The empiric data gathered is being used to close the “calculation and simulation – test rig testing – field testing” loop and to develop a data base for lifetime estimation for each application. Figure 2 illustrates Miba's concurrent engineering and lifecycle program.

**Potential Plain Bearing Failure Modes**

*Sliding speed too low …*

This reservation seems to be justified when typical engine bearing sliding speeds are compared with bearing locations in wind drivetrain applications: 

>10 m/s in crank bearings compared with 1–1.5 m/s in rotor bearings and only 0.5–0.8 m/s in first stage planet gear wheels. Using advanced hydrodynamic simulation tools and extended specific

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**Figure 2**: Miba concurrent engineering processes and lifecycle
bearing test rig tests, it has been proven that a hydrodynamic oil film is created at much lower speeds than expected. Figures 3 and 4 depict results of a planet gear bush simulation at overload, with a surface sliding speed of 0.6 m/s. Even under those conditions, the hydrodynamic peak pressure is rather moderate compared with engine bearings. The oil fill ratio shows a fully filled gap between the bearing bush and pin at the main loaded area (100% hydrodynamic – no dry friction), and how the gap is filled again after the expansion of the clearance gap.

Enormous number of start-stop cycles in wind drivetrains…

Long-term experience with bearings in diesel and gas engines has shown that bearings also have to cope with solid body contact and dry friction. During each start and stop sequence, the full mass load, from the whole crank drive (crank shaft, connecting rods, pistons) up to huge 2-stroke engines, rests on the bearings without any hydrostatic support.

Auxiliary engines on vessels have to cope with numerous starts each day. The load profile of emergency generators in nuclear power plants is characterized by fast starts (zero to full load) and stops during the entire lifecycle – a bearing failure is simply not an option. New requirements from start-stop systems in passenger cars and hybrid truck engines pose a great challenge for bearings, but solutions are available to cope with such conditions.

All these tribology requirements cannot be fulfilled by standard bearing technology. The availability of advanced simulation tools and specific material tests helps us to understand the mechanism of pre-damage and to take the right measures. In combination with a broad range of lining materials and surface technologies, suitable solutions can be developed.

Figure 5 shows a specific test program and the parameters applied for evaluating start-stop properties of different engine bearing materials tested at a bearing clearance of 1/1000. Test results – wear in µm after 1800 cycles – are shown in Figure 6 while Figure 7 shows a test bearing with a synthetic overlay (Miba Syntheoc® with a layer thickness of 25 µm) and the corresponding test results. Both clearly illustrate the effect of a synthetic layer.
Specific load too high for planet stages:
A planet bearing is subject to the highest load compared with the other gear stages and this in combination with the lowest rotational speed, at least for the first stage. Unit loads for typical first planet stages of a wind gearbox are much above 10 MPa at nominal load. In modern gearboxes, unit loads are up to 50 MPa at overload conditions. Together with edge loading from moments and elastic distortion of the adjacent parts, local peak loads for planet bearings are tremendously high. In non-engine applications such as industrial and marine gearboxes, soft bearing materials such as Babbitt and aluminum alloys are typically used in order to cope with start-stop cycles and edge loading. As a result, the maximum unit load is limited to 5–10 MPa. Harder lead-bronze based materials could bear a higher load. But only due to high tribologic demands in combination with a soft overlay. A typical galvanic overlay will most probably not fulfill lifetime requirements due to high wear. And lead-bronze based materials do not withstand high corrosive wind gearbox oils, but lead is not an option in a wind drivetrain anyhow.

In non-automotive combustion engines, unit loads of crank bearings are between 30 and 70 MPa and even up to 90 MPa. These are illustrated in detail for each engine size and the corresponding bearing types in Figure 8. The values depicted reflect the nominal load for which the engine is designed. Unit loads will increase further, driven by ever increasing environmental requirements and the reduced total cost of ownership (cost per kWh). High temperatures (>>100°C), abrasive particles and numerous start-stop cycles complete the full picture of the operating conditions.

<table>
<thead>
<tr>
<th>Engine bore [mm]</th>
<th>Specific unit load [MPa]</th>
<th>Bearing type Lining material + overlay(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>120–130</td>
<td>60</td>
<td>Bronze lining + galvanic overlay</td>
</tr>
<tr>
<td>150–180</td>
<td>70–90</td>
<td>Bronze lining + AlSn Sputter (PVD) + Synthec®</td>
</tr>
<tr>
<td>200</td>
<td>30–35</td>
<td>Bronze lining + galvanic overlay</td>
</tr>
<tr>
<td>320–350</td>
<td>30–40</td>
<td>Bronze lining + galvanic overlay</td>
</tr>
<tr>
<td>400–500</td>
<td>25</td>
<td>AlSn alloy</td>
</tr>
<tr>
<td></td>
<td>35</td>
<td>Bronze lining + galvanic overlay</td>
</tr>
</tbody>
</table>

High stationary load – creeping
Stationary load on bearings can be rather high, particularly on main bearings. If soft plain bearing material is chosen in order to cope with dry friction, misalignment and edge loading, the lining or the solid bearing will creep under high stationary specific load. The
same holds for non-homogenous bearing materials such as AlSn or Babbitt alloys. Embedded Sn in a cast matrix will simply be squeezed out depending on the function of load and time.

Conclusion: Stationary load has to be considered for the basic layout of the bearing location (specific unit load) and/or choice of material combination (tough and homogenous alloys). Figures 9 and 10 depict the difference between a soft, non-homogenous AlSn alloy and a homogenous AlSn sputter (PVD coating) on a tough CuZn alloy without soft phases.

Figure 9: Embedded Sn in cast AlSn alloy

Figure 10: AlSn Sputter (PVD) on CuZn alloy with Synthec® layer on top (black)

In-depth testing of selected materials has been performed to ensure the validity of the conceptual bearing solutions regarding impact of high stationary and dynamic loads, temperature and time.

Lifetime requirement cannot be achieved:
At a first glance, a lifetime requirement of 20 years and 200,000 operating hours appears unachievable if you come from the combustion engine world. On average, a big 4-stroke diesel engine has a service life of 30,000 operating hours (around 5–10 years). The bore is typically 450 mm while the bearings diameter is around 420 mm. Based on this comparison, a wind turbine lifetime requirement does not appear to be achievable with hydrodynamic plain bearings. A more detailed investigation of the load profile of diesels and wind turbines allows a more realistic comparison of the load cycles (fatigue and wear) as well as the sliding path over time (wear). This results in the following picture:

<table>
<thead>
<tr>
<th>Load cycles:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wind turbine rotor:</td>
</tr>
<tr>
<td>(12 rpm, 200’000h)</td>
</tr>
<tr>
<td>144 mill revs./cycles</td>
</tr>
<tr>
<td>Wind GBX 1st planet:</td>
</tr>
<tr>
<td>(40 rpm, 200’000h)</td>
</tr>
<tr>
<td>480 mill revs./cycles</td>
</tr>
<tr>
<td>Diesel bore 450:</td>
</tr>
<tr>
<td>(600 rpm, 30’000h)</td>
</tr>
<tr>
<td>1008 mill revs./2016 mill load cycles</td>
</tr>
<tr>
<td>(all cycles incl. inertia)</td>
</tr>
<tr>
<td>Wind GBX HSS:</td>
</tr>
<tr>
<td>(1500 rpm, 200’000h)</td>
</tr>
<tr>
<td>18000 mill revs./cycles</td>
</tr>
<tr>
<td>Diesel bore 170:</td>
</tr>
<tr>
<td>(1400 rpm, 45’000h)</td>
</tr>
<tr>
<td>3780 mill revs./7560 mill load cycles</td>
</tr>
<tr>
<td>(all cycles incl. inertia)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Sliding paths:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wind turbine rotor:</td>
</tr>
<tr>
<td>(bearing diameter 2200 mm)</td>
</tr>
<tr>
<td>993’600 km</td>
</tr>
<tr>
<td>Wind GBX 1st planet:</td>
</tr>
<tr>
<td>(bearing diameter 220 mm)</td>
</tr>
<tr>
<td>336’000 km</td>
</tr>
<tr>
<td>Diesel bore 450:</td>
</tr>
<tr>
<td>(bearing diameter 420 mm)</td>
</tr>
<tr>
<td>1.43 mill km</td>
</tr>
<tr>
<td>Wind GBX HSS:</td>
</tr>
<tr>
<td>(bearing diameter 140 mm)</td>
</tr>
<tr>
<td>7.92 mill km</td>
</tr>
<tr>
<td>Diesel bore 170:</td>
</tr>
<tr>
<td>(bearing 180 mm)</td>
</tr>
<tr>
<td>2.31 mill km</td>
</tr>
</tbody>
</table>
Start-stop cycles
- Wind drivetrains: According to information given by wind turbine and GBX producers, the total number of start-stop cycles is between 15,000 and 20,000 depending on operating conditions.
- Engines: As stated beforehand, some engine applications must withstand more than 10 starts each day. Over its service life, a diesel or gas engine may be subject to thousands or more start-stop cycles. Most of these engines are started extremely fast, from zero to full nominal power.

Conclusion
- The number of load cycles is a good indication of both fatigue lifetime and wear. This comparison shows that bearings in diesel engines are subject to even more load cycles during their service life than are bearings in wind turbine applications.
- The total accumulated sliding path within the service life of bearings can be another good indication of expected wear, except in the case of high-speed stage engines which have more accumulated sliding kilometers than wind drivetrain bearings. As depicted above, GBX HSS unit loads are many times lower than that of comparable engine bearings.
- Start-stop cycles occur under dry friction conditions and are therefore another good indicator of expected wear. Many engine applications have to cope with similar numbers of starts as wind drivetrains.
- And once more, bearings in diesels additionally have to cope with extremely high dynamic loads (combustion cycle and inertia), at high-temperatures and in combination with abrasive particles.

High friction losses …
Each stage of a wind gearbox is characterized by quite different load situations and rotational speeds. Hence, the friction losses for both roller elements and plain bearings are rather different for each gear stage. Advanced hydrodynamic simulation tools are capable of calculating both hydrodynamic and dry friction losses for each stage. Figure 11 depicts friction losses for each stage of a typical 3 MW gearbox at nominal load:

<table>
<thead>
<tr>
<th>Gear stage</th>
<th>Planet 1</th>
<th>Planet 2</th>
<th>LSS</th>
<th>IMS</th>
<th>HSS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power loss of each stage (radial + axial)</td>
<td>2 kW</td>
<td>5 kW</td>
<td>7 kW</td>
<td>12 kW</td>
<td>16 kW</td>
</tr>
<tr>
<td>Total power loss all stages</td>
<td>&lt;2% power loss</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 11: Hydrodynamic and dry friction losses for each stage of a 2–2.5 MW gearbox

Conclusion
- As expected, slow-speed stages such as intermediate and sun wheel stages and particularly planet stages are very efficient with plain bearings. Due to the high rotational speeds of high-speed stages, the hydrodynamic friction losses are rather high, of course, and most probably remarkably higher than that of roller bearings.
- Wind turbines operate within a wide range of loads, so efficiency has to be taken into account under load situations other than just nominal load.
- And most important: The pros and cons of the whole concept have to be taken into consideration. E.g., if a gearbox equipped with plain bearings loses less than 1% in efficiency compared to REB, this loss will be compensated many times over by higher durability and less repair and maintenance work. Not to mention that a smart plain bearing solution for a newly designed gearbox utilizing the potential of this technology will save weight and room.

High oil demand and oil pressure …
As long as the plain bearing solution is a pure hydrodynamic acting bearing, both oil demand and oil pressure are rather low. Oil demand for the slow-speed stages, in particular, is extremely low. Slow rotational speeds in combination with a bearing clearance in the range of 1–2/1000 automatically limit the oil flow. The bottom line is that even if a 2-3 MW gearbox is equipped with plain bearings on all gear stages, the expected total consumption is less than 40 l/min. Hence, no additional oil pump system is required in contrast to a contemporary gearbox with pressurized oil supply.
Development Activity

Early in 2008, together with a renowned gearbox specialist, Miba decided to launch an R&D project focused on planet stages, which have been considered the most critical and demanding bearing locations. At the same time, a start-up company developing a new 3 MW+ direct drive turbine with plain bearings for the rotor decided to choose Miba as their development partner.

First investigations have been completed: A consideration of unit loads and pv (unit load x sliding speeds) calculations and comparison with known applications. The next feasibility step was to apply existing elasto-hydrodynamic oil film simulation (EHD) tools, specifically developed for combustion engines and wind-application-specific material testing, for pre-selected material combinations at slow sliding speeds.

Hydrodynamic Simulation

This initial purpose and objective of applying EHD simulations was to gather more information on the effect of wind-specific load conditions on the oil film, particularly at peak loads, and to derive design concepts that help optimize the performance of plain bearings in a wind drivetrain. Combustion engine specific EHD tools can be used for high-speed, intermediate and sun-wheel stages without modifications. For the planetary stages and the rotor bearing, the big challenge is the slow rotational speed and the subsequent impact of the necessarily small calculation increments that result in extremely long and unstable calculation runs. As a result, Miba An analogous model has been developed to enable EHD simulation for the planet bearing as well. Basic results regarding oil film creation, oil film pressure and film thickness have already been illustrated in the previous chapter.

Figures 12 and 13 demonstrate the non-progressive characteristic of hydrodynamic plain bearings in contrast to roller bearings. The impact of overload and moments (resulting in local peak loads mostly at the edges) on peak oil film pressure is more or less linear and therefore rather moderate at a first planet stage even at 300% load (Figure 12). Minimum oil film thickness decreases in a regressive manner with load (Figure 13). This characterizes the ability of a plain bearing to withstand shock and peak loads much better than a roller bearing.
Conclusion: High moments cause edge loading, but less than expected. Plain bearing characteristics also help to reduce edge load, while a minor shape adaptation at the ends of the bearing bush or pin results in a significant improvement in oil film thickness and in particular to a reduction in asperity contact. Figure 15 demonstrates the ability of shape correction to remarkably reduce the wear caused by dry friction.

Other simulations, which cannot be depicted in this paper due to confidentiality, demonstrate that the design of the bearing and the choice of material as well as the material combination are inevitable for a reliable plain bearing solution in wind drivetrains. A change in material properties and or thickness of the bearing bush, bearing clearance and elasticity on the ends has a non-negligible impact on the bearing performance.

Material Testing
EHD simulation results, wind-specific frame conditions such as slow rotation, huge moments which result in edge loading and numerous start-stop cycles as well as highly corrosive gearbox oil types narrow down the possible bearing materials and material combinations. Accordingly, a test program for a wind drivetrain bearing application with the focus on the first planet stage has been carefully set up to test potential materials. A description of the test rig description and the test parameters are shown in Figures 16 and 17. The main result of wear rate over time is depicted in Figure 17. The tested materials included the CuZn alloy base material, the AlSn20 Sputter (PVD coating) lining material, the running and the Syntheo® running-in layer.

Summary and interpretation of test results:
- Syntheo®: This synthetic PAI-based layer can be used as a running-in layer to improve the ability to cope with edge loading and as a running layer in certain applications. The objective of the test series was to evaluate the synthetic layer in both applications. The results show typical running-in wear of a few microns after ~100 hours. After several 100 hours, wear levels out and no further wear can be measured. This is a good indication that Syntheo® can be the right choice as a pure running layer, particularly for highly loaded bearing locations with low rotational speeds such as planet and sun-wheel stages in gearboxes and rotor bearings. Dry running tests (oil supply cut off) prove that there will be accelerated wear, but it will not harm the system. Both the bearing and the pin remain in good conditions and there is no evidence of smearing or pre-seizure or any other
suspicious traces or patterns. Start-stop properties of this bearing material were illustrated in the previous chapter.

- Sputter AlSn20Cu: This AlSn20Cu based PVD coating has been developed for highly loaded bearings with high lifetime requirements. First used in passenger cars, then for heavy duty truck engines, they are now also used for high-speed diesel and gas engines. Similar to Synthec®, no further wear can be measured after the running-in phase. Typical applications would be radial bearings for the high-speed and intermediate stages as well as highly loaded planet stages.

- CuZn alloy: The purpose of testing the pure base material was only to make sure that once the running layer(s) disappear(s) there will not be any seizure, not even under high load. It is obvious that the wear rate is much higher compared with Synthec® or Sputter, but the bearing and the pin remain in good condition. There is no evidence of smearing or pre-seizure or any other suspicious traces or patterns. Although the tested CuZn alloy is known for its excellent tribologic properties (wear resistance under high unit load and slow rotational speed), this test result is good evidence that planet bearings would most probably not be feasible without adequate surface technology.

Gearbox Bearing Concept
Derived from our EHD simulations, material tests and design work, there is a smart concept available for planet plain bearings (including the oil supply system) for testing purposes as well as for serial use. For high-speed, intermediate and sun-wheel stages, conceptual design solutions have been realized for both radial axial bearings to be tested in prototypes. Design concepts are matter of confidentiality and thus not part of this paper.

Rotor Bearing Concept
A unique product development derived from the first gearbox bearings development work is a direct substitution for a double-row tapered roller bearing for the rotor which is depicted in Figure 18. Exchangeability of the bearing pads without removing the rotor or even loosening any main bolts is part of the patent claims Miba has applied for. This important feature ensures low maintenance costs even if wear is higher than expected.

Based on the conceptual design, the bearing is a hybrid between a radial and an axial hydrodynamic system. Hence, the huge challenge is oil film simulation as there is currently no standard tool available. As a result, efforts have been focused on optimizing the design for load distribution.

FEM simulation
Peak loads on a double-row tapered momentum bearing are extremely high when taking into consideration ultimate load cases such as a 25-year wind gust including yaw misalignment. The meshing for this FEM simulation takes into account the rotor, stator, hub and the bearing consisting of outer and inner rings, as well as the bearing pads and their fixation. Results are being used to modify the bearing design towards more elasticity in order to distribute the load over a maximum number of bearing pads and towards achieving good load distribution for each single bearing pad. Within several iteration loops, it is possible to substantially reduce peak contact pressure on the bearing pads. The progress within three iterations in the case of extreme load is illustrated in Figure 19. Figure 20 gives an impression of the load distribution over the bearing circumference and on the bearing pad for a typical fatigue load.
case (nominal load including standard deviation).

![Figure 19: Reduction of peak contact pressure within three iteration loops](image)

Figure 19: Reduction of peak contact pressure within three iteration loops

![Figure 20: Distribution of load](image)

Figure 20: Distribution of load

**Outlook**

**Simulation**

Important and basic information could be achieved as one part of the feasibility of plain bearings in wind drivetrains. For gearbox bearings, more in-depth simulations provided valuable data for design work for all bearing locations in wind gearboxes. Simulation tools for wind specific applications will be developed further, in particular for the momentum rotor bearing and the planetary stages in wind gearboxes.

**Material Testing**

The simulation results and the specific bearing material testing provide sufficient confidence that both gearbox and rotor bearings are feasible under normal and under extreme operating conditions (overload, shock loads) based on the technology presented within this paper. To make sure that the presented plain bearing concepts will also hold for low idling and numerous start-stop sequences, Miba is working on a modified bearing test rig suitable for simulating low idling and start-stop operation under high load as well.

**Gearbox Bearings**

Planet gear wheel bushes were successfully tested for the first time on a back-to-back test rig in June 2010. With some modifications to the planet bearings and with additional prototypes for radial and axial plain bearing for the high-speed stage, testing will be continued in the second quarter of this year, including simulation of extended overload conditions. This gearbox testing program is the basis for a decision to use a plain bearing gearbox on a tower for field testing.

Once plain bearings have been validated and approved, the next important step is to design a new gearbox, taking into account the design rules and limits of plain bearings for each stage and exploiting their basic potential advantages. From today’s point of view, a multi-megawatt gearbox will have a more compact design using plain bearings and will help to substantially reduce weight.

**Rotor Bearing**

A prototype bearing for a 3 MW+ direct drive turbine will be produced this year, and delivered early in 2012. The validation program starts with functional testing at ground level and will be later continued on a tower. After field testing has been successfully completed in 2012, a zero series will be delivered sometime in 2013.
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