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Correlation of Planetary Bearing Outer Ring Creep and Gear Load Distribution in a Full-Size Wind Turbine

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Abstract. In this work, bearing outer ring creep in the planetary gear of a wind turbine gearbox is investigated. So far, gear creep has only been investigated experimentally on very small bearings under simplified load conditions. This paper presents experimental measurement results from a full-size 2.75 MW wind turbine. The tests were performed on a system test bench capable of applying loads in all six degrees of freedom at the rotor hub. The objectives of this investigation are to provoke and measure bearing outer ring creep and to identify correlations to the gear load distributions acting on the planetary gear. After a motivation, a literature overview on bearing ring creep is given. Influences on and occurrences of ring creep are described and the two mechanisms roller-induced creep and gear creep explained. Then, the test setup and the evaluated experiments are presented. The influence of torque, rotational speed and additional non-torque loads on gear load distribution and bearing outer ring creep is shown, discussed and put in relation to findings from literature. Key findings are that outer ring creep speed increases with global rotational speed and torque. Also, gear creep – acting in the opposite direction of roller-induced creep – was found at the generator-sided bearing ring for loads below 40% nominal torque. The planet gear load distributions with sun and ring gear were highly uneven and opposed to each other at low torques while leveling out at high torques due to the load-dependent deformations of sun gear, planet carrier and ring gear. The average gear load distribution over all ring gear positions showed to be influenced only by torque and to be independent of non-torque loads. In accordance, extreme non-torque loads were found to not influence the creep behavior of the two planetary bearing outer rings.

1. Introduction and motivation
The share of renewable power has increased rapidly in recent years, with wind power exemplarily constituting the second largest share amongst all energy sources in the European Union today [1]. As the cost pressure on the entire supply chain of wind turbines (WT) increases to be competitive with conventional energy sources, improving the reliability of WTs has become one of the major challenges today [2]. To improve the reliability of today’s complex WTs, understanding the detailed causes of damages becomes essential. Due to high downtimes per failure caused by time-consuming repair, mechanical damages in the gearbox are of particularly high interested in this context.

Bearing ring creep – especially bearing outer ring creep – is one of the mechanical issues in planetary bearings which on average require two to three costly repairs over the design life time of a WT [3]. This usually involves a replacement of the entire gearbox as up-tower repairs are often not possible. Bearing ring creep is a phenomenon occurring especially in WT planetary bearings due to rapidly increasing gearbox torque density [4] and high specific loads. Together with elastic surrounding structures [5] due to lightweight constructions, deformations within the gearbox increase. With gear teeth engagements being sensitive to misalignments, uneven gear load distributions can be the result.
The phenomenon of bearing ring creep has been investigated extensively on small scale model test benches by means of simulations and experiments [5] and is well understood within the investigated parameter range. However, uneven gear load distribution as well as multiple-row bearing concepts, as commonly seen in WT planetary gears, have been identified as highly relevant influences on ring creep [5], but have neither been tested under realistic load conditions nor in a full-size wind turbine. Therefore, a metrological investigation as presented in this paper is necessary.

2. Literature overview bearing creep

This chapter will give a brief literature overview regarding bearing ring creep, whereby the focus will be on the investigation of bearing outer ring creep – also referred to as just outer ring creep. More in-depth information can be found in the cited literature [5].

Outer ring creep can be divided into two independent mechanisms which provoke opposing tangential movements of the bearing ring in its housing. These movements can lead to fretting corrosion, setting free particles that might cause problems in other places of a gearbox. Also, misalignment and ultimately breaking of components can be caused, leading to failures of entire gearboxes [5].

Simulating ring creep is possible and has been done [5]. However, literature shows that the calculated creep speed is extremely dependent on the friction coefficient between the bearing outer ring and the planet gear as well as on the press fit. Even very small changes in these values drastically change the obtained creep values. As the exact friction coefficient and press fit are not known for the investigated setup – as in most real systems – no creep simulation was performed.

2.1. Roller-induced creep

Roller-induced creep refers to a mechanism that leads to small, irreversible tangential movement of a bearing ring in/on its seat. This occurs in the rotational direction of the entire rolling element set. Several influencing factors have been investigated in recent years [6] [7]. Increasing load, loose fits, thin bearing rings and circumferential load (opposed to point load) amplify the effect while higher bearing clearance, wider bearing rings, more rollers and increased friction in the bearing seat weaken ring creep, to only name the main factors.

The mechanism of roller-induced creep is generally described as a “caterpillar-shaped” deformation of the bearing ring due to radial load. In [8], this is exemplarily shown for a simple plate model. Figure 1 schematically depicts this situation for a bearing outer ring. The upper gear in figure 1 shows the initial position with a radial load acting on the rollers, but without a rotation yet. The bearing outer ring shows the “caterpillar-shaped” deformation (exaggerated in this figure), leading to a small gap in between rollers.

Looking at the lower gear, the rollers have moved by an angle $\alpha$ and so have the gaps in between them. Due to the prevailing radial and tangential stress situation in the bearing set, the press-fit is overcome locally by tangential strain which leads to a very small movement $\Delta \alpha$ of the outer ring relative to its housing, in this case a gear wheel. One of the mechanical prerequisites identified in [8] is that the press fit must be overcome along the entire bearing width to enable this tangential movement at all. As the described sequence occurs with every overroll, roller-induced creep increases with rotational speed and time.

2.2. Gear creep

In addition to roller-induced creep, gear creep can occur if the bearing housing is a gear. The radial and tangential load situation in each gear tooth engagement and the resulting tooth bending lead to the formation of an additional gap in the area of each teeth engagement. When set in rotation, this gap is set into motion as well. Other than roller-induced creep, gear creep drives the bearing ring against the rotational direction of the rolling element set. Figure 2 shows the rotational direction of the bearing ring (red dot) relative to the gear wheel (blue dot) for roller-induced creep (middle) and gear creep (right). The rotational direction of the rolling element set (black dot) is the same in all pictograms.
In [5] it was found out that gear creep already occurs at lower loads than roller-induced creep. However, the speed of roller-induced creep is higher. Therefore, at a certain torque/load level, the two mechanisms compensate and at higher loads, roller-induced creep dominates.

In addition to tangential movements, gear creep can also cause axial movements in bearing seats if the gear has a helix angle other than 0°. In this case, tangential gear creep is more likely to be seen at the bearing (rotor- or generator sided bearing) where the “pre-loadzone” tooth engagement ends due to the gear’s helix angle and direction of rotation. For this, the “pre-loadzone” tooth engagement is defined the one that a tooth passes through before rotating into the bearing load zone.

In the outlook of [5], the load distribution along the gear width was identified as a highly relevant factor to be investigated, especially for multi-row bearing concepts. Figure 3 schematically depicts an even and uneven load distribution on the two gear engagements of a planetary gear. The arrows on the outside represent the resulting axial, tangential and radial forces on the planetary gear acting from the...
sun and ring gear. The multiple arrows on the inside represent the resulting bearing load distribution. In the right pictogram, the resulting gear meshing forces are uncentered due to misalignments in the gearbox, for example tilting of the planet gear and torsion of the sun gear.

3. Test setup and experiments
At the Center for Wind Power Drives (CWD), a three-point suspension, 2.75 MW generic research nacelle with a gearbox ratio of 62.8 was investigated on a system test bench. The test bench allows the load application in all six mechanical degrees of freedom at the WT’s hub [9]. Any generic load can be applied and the obtained measurement results are highly reproducible. Figure 4 shows the research nacelle mounted on the system test bench at the CWD.

![Figure 4. Generic 2.75 MW research nacelle on system test bench at CWD, RWTH Aachen.](image)

Amongst roughly 300 sensors in this turbine, the measurement equipment included rotating strain gauges on the sun and stationary strain gauges on the ring gear at four positions over the circumference [10]. By this, the load distribution in both planetary gear’s contacts can be evaluated over the gear width and over the ring gear’s circumference. In figure 5, the strain gauge positions on the sun and ring gear are shown.

![Figure 5. Strain gauge positions for gear load distribution evaluation [11].](image)

Also, distance sensors were installed in the planet carrier, facing the planetary gear and its bearing outer rings from both sides [10]. In figure 6, the positions of the distance sensors used for the presented creep evaluation are shown. The described setup allows the axial positions of the planetary gear and its bearings’ outer rings to be measured directly. By calculating the difference of these signals, axial creep can be detected.
To measure tangential creep, grooves were machined into the axial face sides of the gear and the outer rings which pass the distance sensors once every rotation. This allows the calculation of rotational speed by evaluating peak times in the distance signal. The data evaluation concept is schematically shown in figure 7. Axial creep is calculated directly from distance differences, tangential creep by comparison of passing periods of the grooves.

When no creep occurs, the time difference between a “planet gear peak” and an “outer ring peak” ($\Delta t_1$) remains constant over time. If the time difference increases ($\Delta t_2$), tangential creep occurred. In the shown example $\Delta t_2 > \Delta t_1$ which means that the bearing ring took more time for one revolution than the planet gear. Hence, the bearing ring is performing a relative rotation in the planet gear against the direction of rotation which would be interpreted as gear creep. As $\Delta t_1 > \Delta t_2$ the creep speed increased, for example due to increased load.

For this investigation, experiments from three different load scenarios were evaluated. Every evaluation included the gear load distribution at sun and ring gear as well as the calculation of outer ring creep. The qualitative time series of the evaluated load scenarios are depicted in in figure 8.
Each load scenario was chosen to obtain a different set of information. The objective of the first load scenario (“Constant Operating Points”) was to evaluate the influence of pure rotor torque on the gear load distribution and the general creeping behavior. With the help of the second load scenario (“Torque ramps at constant speeds”), the influence of rotational speed on creep was evaluated. The objective of the third load scenario (“Loads in 6 degrees of freedom”) was to evaluate the influence of non-torque loads on gear load distribution and creeping behavior.

Outer ring creep was not accelerated in any fashion during these tests. Rotational speeds, torque and non-torque loads were within operating range of the investigated turbine. As ring creep is a very slow and most importantly cumulative process and literature shows no influence of dynamic loads on ring creep, dynamic loads were not investigated. In addition, the presented detection and evaluation method would not be able to process tangential creep phenomenon on very small time scales to see the influence of dynamic loads.

4. Results

This chapter presents the results of the chosen load scenarios with respect to gear load distribution and creeping behavior. For every test, gear load distribution was evaluated at four positions on the ring gear (see figure 5) and the gear load distribution of the sun gear was reduced to these areas for easier comparison.

4.1. Constant Operating Points

In figure 9 the gear load distribution for 40% and 100% nominal rotor torque is shown. For better overview, only the “180°-(low) position” as defined in figure 5 is shown. The blue crosses represent the sun gear measurement, the red crosses the ring gear measurement. One of the sun gear strain gauges had to be interpolated due to technical issues during data acquisition. The percentage values of each color are the sum of the three strain gauge measurements of this side (rotor or generator) divided by the sum of all six measurements of this component. Therefore, this value – in a simplified way – represents the load carrying share of this half of the tooth. The pictogram qualitatively shows the load distribution for the evaluated torque level for both tooth engagements.

It can be seen that at lower torque (40%) the sun gear carries more load on the rotor side while the ring gear carries more load on the generator side. For 100% nominal torque, this constellation inverts.
In figure 10, the load shares for 40%, 60%, 80% and 100% nominal torque with no additional non-torque loads can be seen. Here, the average load shares over all 4 ring gear positions is given. The load share percentages on the rotor- and generator-sided halves of each gear engagement add up to 100%.

Figure 10. Average rotor- and generator-sided load carrying share at 40-100% nominal torque for sun (left) and ring (right) gear.

It can be seen that the sun gear’s generator-sided load-carrying share increases with torque. This effect correlates with the torsion of the sun gear. The ring gear’s behavior is the opposite as the generator-sided load-carrying share decreases with torque. This behavior is mostly related to the torsion of the planet carrier and the resulting misalignment of the planet gear. The effect that the shares level out with increasing torques proves the effectiveness of the applied tooth modifications.

In figure 11 the creeping behavior for the four evaluated torque levels is shown for the two bearing outer rings. The y-axis are scaled differently due to different total creeping. The left axis shows the tangential creep in degrees, the right axis shows tangential creep in mm at the bearing seat. Positive creep values represent roller-induced creep, negative values represent gear creep.
The data clearly shows higher creeping for higher torque. Also, all curves show a linear behavior. Therefore, the creeping appears to be constant over time. For the rotor-sided bearing, only roller-induced creep was found. However, the generator-sided bearing shows negative tangential creep for 40% and 60% nominal rotor torque and therefore gear creep. This result correlates with information from literature as the generator-sided bearing is where the gear engagement with the sun gear ends before entering the bearing load zone as described in chapter 2.2. At 100% load the rotor-sided bearing shows almost twice as much positive creep as the generator-sided bearing which is partially "held back" by the still ongoing, but overcompensated gear creep.

Figure 12 shows the creep speed for both bearings at the investigated torque levels (40-100% nominal torque). In addition, the creeping speed at a very low torque level of about 13% nominal torque was evaluated to be zero. The figure sums up the results from figure 11 by clearly showing the influence of torque on both bearing rings’ creeping speed.

4.2. Torque ramps at constant speeds
In the experiments evaluated in this figure, pure torque ramps were applied at various constant speed levels. The torque was increased at a speed of 0.1 kNm/s at the generator side up to its maximum of 24.7 kNm. As the gear load distribution changes in dependency on torque (as shown in chapter 4.1), it was evaluated at short time periods of the torque ramp at torque levels correlating to the “constant operating points”. As expected, the gear load distribution at those torque levels was the same and is therefore not shown again here. However, this result proves that the evaluation method gives proper
results even for much shorter time periods. In figure 13 the tangential outer ring creep for both bearings is shown for constant rotor speeds of 4.4 rpm, 8.8 rpm, 13.1 rpm and 17.5 rpm.

The curves are not linear as the torque constantly increases. Also, it can clearly be seen that higher rotational speed of the rotor also leads to higher absolute creep values. For lower speeds, this effect appears to be linear while the curve at 17.5 rpm does not follow this pattern. However, creeping speed also increased compared to 13.1 rpm and therefore the curve qualitatively shows the expected behavior. The curves of the generator-sided bearing also qualitatively show the expected behavior; gear creep at lower loads, roller-induced creep at higher loads and an increase of creep with increasing rotational speed. The slightly unsteady curves indicate that the evaluation method for tangential creep becomes less steady for short time intervals. Nevertheless, the overall course of all curves appears reliable.

4.3. Loads in 6 degrees of freedom – “6DOF”
This chapter investigates the influence of non-torque loads at the rotor gear load distribution and ring creep behavior on the load carrying share. Figure 14 shows the rotor-sided load carrying share of sun and ring gear under different “6DOF” load scenarios. As in figure 10, the average shares over all four ring gear positions are shown. The blue and red bar do not add up to 100% in this figure as each bar represents the rotor-sided share of one component. More “6DOF” load scenarios were investigated and show the same behavior.
The first load scenario “Reference” contains the same data as the red lines in figure 10 for 100% nominal torque. Both other load scenarios include an additional negative force in z-direction of 500 kN, representing the rotor weight. It can be seen that regardless of externally applied non-torque loads, the load carrying shares are almost unchanged compared to the reference scenario with pure torque. This is explicable as non-torque loads at the rotor hub are mostly diverted into the machine carrier by the main bearing and planet carrier bearing. Consequently, mostly torque is transmitted to the planetary gears, leading to the behavior shown in figure 10.

Figure 15 shows tangential outer ring creep for both bearings’ outer rings for the reference load scenario (pure torque) and the load scenarios “A” and “B” from figure 14.

Compared to the “Ref” load scenario, both “6DOF” load scenarios show higher, but equal creeping behavior, despite very different external loads. A direct comparison of “Ref” and “6DOF” only gives limited information as the “6DOF”-experiments were done roughly 9 months after the “Ref”-experiment. In the meanwhile, a large number of experiments with different scopes of investigation was done. During this time, the outer rings most likely continued creeping and by this, conditions in the gap, such as such as the press fit and the friction coefficient – an effect known from literature [7] – might have changed.

4.4. Axial creep

As mentioned, axial creep of the bearing outer rings was also evaluated in this work. However, axial creep only occurred during the very first tests (speed ramps) as soon as the torque was higher than 50% nominal torque. As soon as this level was reached, axial creep occurred during two tests until the axial clearance of both bearings’ outer rings was used up. Both bearings creped out of the planet gear. This result does not correlate with literature findings [5] where axial creep of two bearings’ outer rings was found to be out of the gear for one bearing and into the gear for the other. Further investigation regarding this deviation is necessary.

4.5. Optical inspection of selected components

In this chapter, selected pictures of components of the investigated 2.75 MW gearbox after 12 years of operation are presented. Table 16 shows marks of tangential movement on one outer ring seat in a planet gear and on the outer bearing ring dismounted from this planet. As bearing ring creep can also occur on bearing inner rings, marks on the bearing inner ring as well as on the planet pin are given as well.
Table 16. Marks of tangential movement on inner (left) and outer (right) ring seats of planet bearing.

| Outer ring seat planet gear | Outer ping seat planet bearing | Inner ring seat planet bearing | Inner ring seat planet pin |

Axial movements do not leave visible marks as on the components as the total creeping distance is very small. However, the resulting axial forces that build up lead to wear. Table 17 shows the marks and wear caused by the axial forces on the bearing outer ring face side as and on the holding circlip ring. Massive wear can be seen on the softer circlip ring which in this case has lost about half its thickness.

Table 17. Marks of axial forces on outer ring face and circlip ring.

| Outer ring face planet bearing | Circlip ring with massive wear |

5. Conclusion and outlook
In the presented work, a full-size 2.75 MW generic research nacelle was investigated on a system test bench with the focus on outer ring creep in the planetary bearing. The gear load distribution acting on the planetary gear was evaluated under different load cases. While the average load share for each bearing remained at about 50% for all load cases, the load distribution at the tooth engagements with the sun and ring gear were shown to be highly dependent on rotor torque. Even high applied non-torque loads at the turbine’s rotor hub did not have a significant influence on the average load share of the bearings.

For selected load cases, outer ring creep was evaluated. A clear dependency on rotor torque was shown for both bearings’ outer rings with 100% nominal rotor torque leading to about 1.8° and 40% nominal torque leading to only 0.4° of creep at the rotor-sided bearing. Gear creep – leading to tangential movements in the opposite direction of roller-induced creep – was found at the generator-sided bearing’s outer ring for 60% and even more for 40% nominal rotor torque. The clear dependency of tangential creeping speed on rotational speed was confirmed. However, linearity could only be shown with some uncertainty. The evaluation of tests with high and different applied non-torque loads at the rotor showed no influence on creeping behavior.

In further work, the influence of varying bearing load shares far different from 50% will be evaluated. As the influence of non-torque loads on outer ring creep was shown to be negligible, these experiments will be done on a component test bench, which reduces testing cost drastically. The results from this work can then be used to help improve the design of gear flank modifications in wind turbines in order to obtain gear load distributions favorable to avoid or at least to reduce outer ring creep. This would lead to no or less wear in the press fit and extend the lifetime of gears and bearings in wind turbine gearboxes.
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