



Assessment of ring deformations and internal grease pressure for seal design in a 5 m three-row roller wind turbine blade bearing

Matthis Graßmann¹, Marlene Bantz², Frank Schönberg², Arne Bartschat¹

¹ Fraunhofer Institute for Wind Energy Systems IWES, Large Bearing Laboratory, Am Schleusengraben 22, 21029 Hamburg, Germany

² Freudenberg FST GmbH, Höhrerweg 2-4, 69469 Weinheim, Germany

Correspondence to: Matthis Graßmann (matthis.grassmann@iwes.fraunhofer.de)

Abstract.

Large wind turbine blade bearings require reliable sealing systems to retain lubricants and prevent contamination throughout their service life. Seal performance is strongly influenced by relative movements of the bearing rings and by internal grease pressure. This study investigates ring deformations and internal grease pressure in a grease-lubricated 5 m three-row roller blade bearing using the Fraunhofer IWES BEAT6.1 test rig. Bearing ring movements are assessed using a validated finite-element model that includes the bearings and adjacent support structures that emulate blade and hub stiffness properties. Internal grease pressure is measured experimentally under representative operating conditions and under extreme scenarios involving closed grease outlets. The simulations predict maximum radial gap variations between -0.29 mm and +0.59 mm and maximum axial displacements of up to 0.41 mm, remaining well below the sealing system's allowable movement range of ± 1.5 mm. Pressure measurements show no significant pressure increase during normal operation, while extreme conditions with blocked grease outlets and no pitch motion result in pressures up to 2 bar. Introducing pitch movement rapidly reduces pressure levels to below 0.5 bar. The results demonstrate that the investigated sealing concept is capable of accommodating the expected ring deformations and pressure levels of a large three-row roller blade bearing. Furthermore, the study provides experimentally validated boundary conditions for the design and assessment of sealing systems in future wind turbine blade bearings.

1 Introduction

Wind turbine blade bearings connect the rotor blade to the rotor hub and enable the rotation (pitching) of each blade about its longitudinal axis. Pitching controls the wind turbine's power output and loads. Blade bearings are grease lubricated. To keep the grease inside the bearing and prevent contamination from the outside, sealings between the inner and outer rings are installed. In operation, the gap between the bearing rings can open and close due to deformations caused by external loading. For the design of the sealings, the minimum and maximum gap is crucial to ensure that the sealings work properly and can prevent any grease contamination and grease leaks even under extreme loads (Harris und Kotzalas 2007). Under operation,



new grease is pumped in the bearing through dedicated channels in intervals, while additional bore holes allow old grease to exit the bearing. This grease usually is collected in containers. However, the behavior of the grease in the bearing is chaotic. Even in the laboratory it is impossible to predict the grease flow and where the grease might leave the bearing. The viscosity of the grease depends on the temperature and its individual formulation. At lower temperature, the grease becomes thicker.

35 Especially the combination of chaotic unpredictable grease behavior and high viscosity may lead to a pressure increase in the bearing or in parts of the bearing.

For the design process, the bearing manufacturer typically specifies the required maximum opening gap, minimum gap and the pressure inside the bearing which the sealings need to withstand or accommodate with. The maximum deformation of the bearing rings can be simulated by the means of finite-element simulations. Different approaches for modelling roller-raceway interactions in large scale roller bearings have been published (Wang et al. 2017; He et al. 2018; Becker et al. 2014; Stammler et al. 2018; Stammler und Schleich 2025). GRABMANN et al. (2026) validated a FE model of a three-row roller blade bearing against experimental data of the ring strains and deformations. They achieved a very comparable ring deformation and a maximum deviation between FE model and experiment of less than 10% neglecting single outliers. However, none of the publications investigate the maximum ring deformation with respect to requirements of the sealing design. For the expected pressure in the bearings, to the knowledge of the authors, very limited information is publicly available. KHAN et al. (2025) investigated the grease flow in a double-row tapered roller bearing as a main bearing of a wind turbine using a multiphase CFD model. They investigated the behavior of the grease inside the bearing under various operating conditions and fill rates below 50%. They simulated the grease distribution after one full rotation of the inner ring of the bearing and examined the occurring pressure towards the sealings. Their results show higher pressure with higher fill rates with a maximum pressure of 0.12 bar for a grease fill rate of 45%. ZHANG et al. (2026) simulated a blade bearing considering the hub and a blade dummy to investigate the deformation of the bearing rings under complex load situations. The bearing is a double-row four-point contact ball bearing with an outer diameter of 1350 mm. The results show a significant ring ovalisation especially of the inner ring due to the inhomogeneous stiffness of the surrounding structures with a maximum relative radial displacement between the bearing rings of 1.08 mm. They further evaluated the resulting gap between the bearing rings to investigate the performance of different sealing designs. Based on their simulations, they discovered the risk of grease leakages with their previous sealing design and showed that the new sealing has better capabilities of withstanding the expected deformations.

40
45
50
55

The very limited information about the pressure in the bearing might lead to very conservative specifications by the bearing manufacturer resulting in increasing challenges for the sealing manufacturer in matching the requirements. This work aims to close this gap by assessing the ring deformation and internal grease pressure for a standard seal design in a 5 m three-row roller wind turbine blade bearing. In this work, the Fraunhofer IWES large blade bearing test rig BEAT6.1 (Bearing Endurance and Acceptance test) is used to measure the pressure in a blade bearing under realistic and extreme conditions. In addition, the gap opening and closing of the blade bearing rings are simulated using a validated finite element model of the

60



bearing and test rig. Section 2 gives an overview of the methodology. The Results are discussed in Section 3. Section 4
 65 concludes the work.

2 Methodology

2.1 Bearing and test environment

The bearing in this work is a grease-lubricated three-row roller bearing with an outer diameter of nearly 5m. Two rows of axial rollers carry the axial component of external loads, while one row of radial rollers carries the radial component. The
 70 bearing is mounted with bolts to the surrounding structures of the Fraunhofer IWES BEAT6.1 (Bearing Endurance and Acceptance test rig). The rig tests two bearings simultaneously. The outer bearing rings are mounted to hub adapters that emulate the stiffness of the rotor hub, and the inner bearing rings are mounted to the force transition element (FTE) that emulates the stiffness of the blade root. These adapters allow for a realistic load situation of the bearing. They are designed to represent the stiffnesses of the surrounding structures of the Fraunhofer IWES reference turbine IWT-7.5-164 (Popko und
 75 Thomas 2018). Six hydraulic cylinders connected to the load platform in a hexapod configuration apply loads in any degree of freedom to the bearings. An additional hydraulic cylinder connects to the inner ring of the lower bearing and allows for pitching the inner part of the test rig including the inner rings of the bearings. The loads at the cylinders are controlled by calibrated load cells. More information about the test rig can be found in (Stammeler 2020). Figure 1 shows a cross-sectional schematic view of the bearing and the test rig. The bearings are displayed in orange, the hub adapters in dark red. The FTE is
 80 the inner part between the two bearings. The load platform is the top white structure with the attached cylinders.

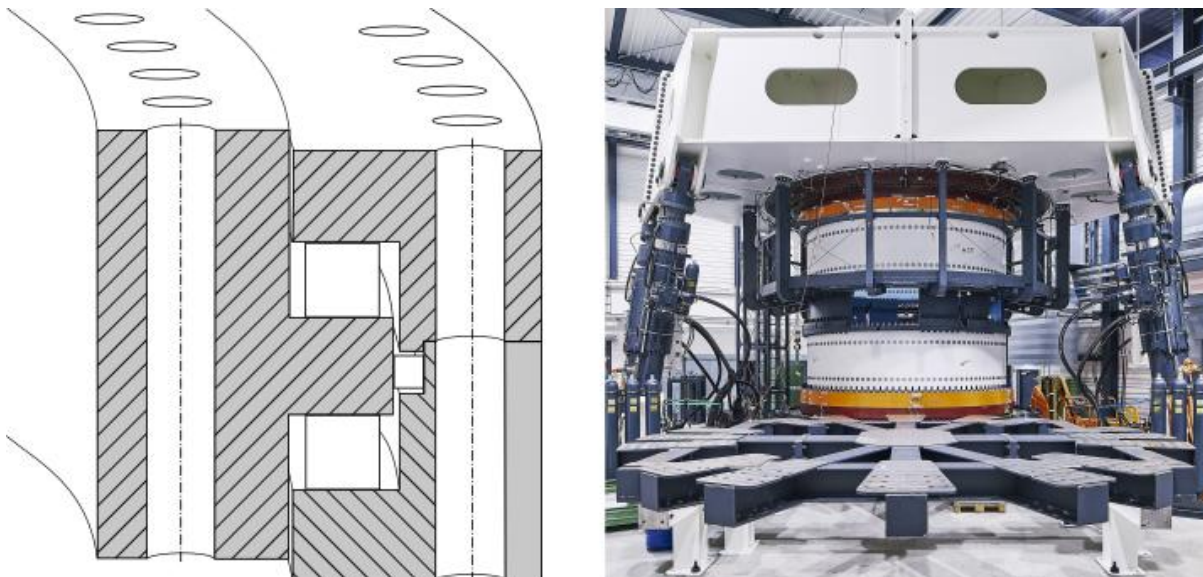


Figure 1: Cross sectional view of a three-row roller bearing with a segmented outer ring (left); BEAT6.1 (right)



The bearing is lubricated with Gleitmo 585k. The grease pump is a Lincoln P603S-15XLF-3Z7-AC-2B1. The pump is set to start a pump cycle every 4 minutes. With each pump cycle, the grease is pumped into the bearings until a working pressure of 240 bar is reached and a pause timer of 4 minutes starts again. This configuration results in the maximum possible grease flow with this pump type to test extreme conditions and remains the same for all the following tests. Both bearings have 18 grease inlets connected to their outer rings and 12 collector bottles connected to grease outlets at their inner and outer rings equally distributed around the circumference. Each bottle has a small hole to minimize the counter pressure. Figure 2 shows grease inlets and a bottle connected to a grease outlet. Prior to the first test of this work, the bearings have reached a calculated fill level of 100%.



Figure 2: Grease inlets and outlets on the outer ring of the lower bearing

2.2 Sealing

Seals are essential components of large rolling bearings, ensuring lubricant retention and preventing the ingress of contaminants throughout the operational lifetime of the bearing. In wind turbine blade bearings, the seal is exposed to significant relative movements between the bearing rings caused by structural flexibility, operational loads, manufacturing tolerances, and internal grease pressure. Consequently, seal design is not a stand-alone process but must be closely integrated with the bearing design and the structural behavior of the surrounding system. Different sealing concepts are commonly applied in large rolling bearings, including radial-radial (rad-rad), axial-radial (ax-rad), and radial-axial (rad-ax)



100 configurations as shown in Figure 3. The classification is performed from the bearing side towards the environment. Traditional rad-rad concepts are characterized by simple manufacturing and assembly and are suitable for applications dominated by axial gap variations. Ax-rad designs provide improved support of the sealing lip through an undercut geometry, resulting in higher resistance against internal pressure, although at the expense of more complex manufacturing and assembly. Rad-ax concepts offer simplified groove geometries and improved pressure stability through an axial support
 105 shoulder but generally allow larger radial movements and smaller axial displacements. Each concept therefore represents a compromise between manufacturability, pressure resistance, and permissible ring movements.

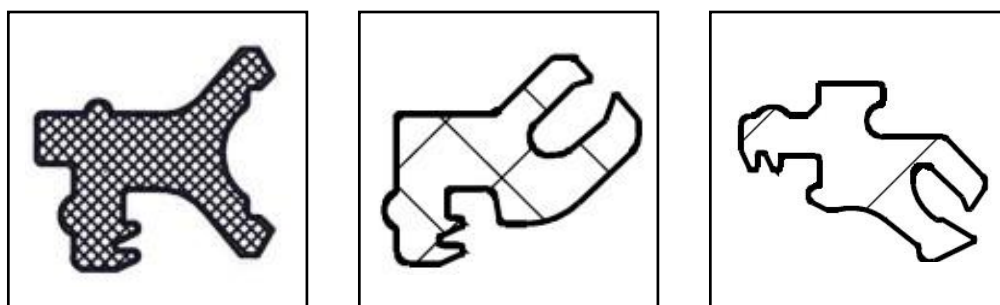


Figure 3: Standard sealing concepts: radial-radial (left), axial-radial (middle), radial-axial (right)

The sealing development process starts with the definition of the operating requirements by the bearing manufacturer. These requirements are typically transferred to the seal supplier through an Engineering Information Sheet (EIS), which contains
 110 all parameters relevant for seal selection and design. The EIS represents the central interface between bearing and seal development and provides the basis for evaluating whether an existing catalogue seal can satisfy the application requirements or whether a dedicated sealing solution must be developed. A comprehensive set of input parameters is required for the design of seals in large rolling bearings. Material-related requirements include the lubricant type,
 115 environmental exposure, temperature range, humidity, and resistance against ageing mechanisms such as ozone or weathering. Equally important are the geometric and mechanical boundary conditions. These include the nominal gap between the bearing rings, manufacturing tolerances, maximum radial and axial displacements, available installation space, groove geometry, and the expected internal grease pressure. Dynamic operating conditions must also be considered, since the seal is required to maintain contact and functionality while continuously accommodating relative ring movements during
 120 operation. The importance of accurately defining the gap displacement becomes increasingly significant with growing bearing dimensions. In large blade bearings, the deformation of the rings is not solely governed by the bearing geometry itself but also by the stiffness of the surrounding support structures. As a result, the actual gap variations experienced by the seal can differ substantially from simple analytical estimations. Bearing manufacturers therefore commonly specify maximum and minimum gap dimensions as well as permissible internal pressures that the seal must withstand throughout
 125 operation.



Besides a large variety of existing seal designs, some applications require specially designed custom solutions. FE simulations play a key role throughout the entire design process (Bacchetto 2013). Seal suppliers evaluate new concepts using detailed FE models of the sealing profile. These simulations incorporate advanced elastomer material models and are used to investigate contact forces, deformation behavior, lip stability, retention forces, and pressure resistance. FE analyses provide quantitative evidence that a sealing design fulfills the requirements specified in the EIS before prototypes are manufactured and tested. Despite the widespread use of seal-level FE simulations, the quality of the design process ultimately depends on the accuracy of the boundary conditions used as inputs. In many cases, FE analyses are performed on isolated seal geometries using prescribed gap values taken from bearing specifications. However, such specifications do not necessarily capture the complex interaction between the bearing rings and the surrounding support structure. Large wind turbine blade bearings are particularly sensitive to these effects, as the flexibility of the hub, blade root, adapter structures, and bolt connections can generate substantial ring ovalisation and local gap variations.

For the bearing investigated in this work, a standard design was selected with allowable radial and axial movements of ± 1.5 mm and a pressure capability of approximately 1.9 bar. Figure 4 exemplarily shows resulting deformations of the sealing for different applied pressures inside the bearing from the FE simulation of the sealing design.

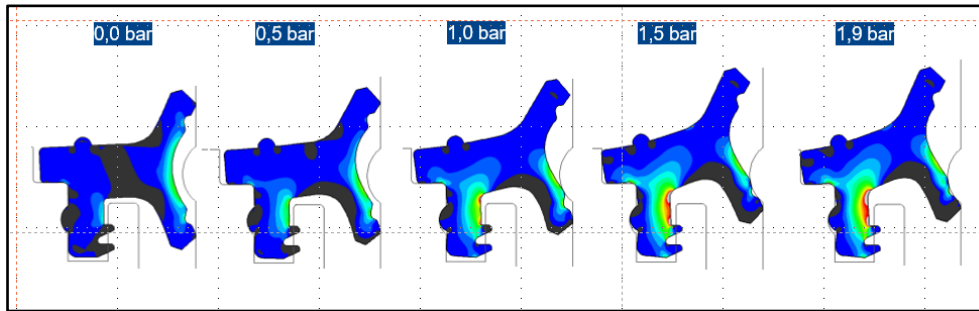


Figure 4: Deformation of the sealing design for different pressures inside the bearing

2.3 FE simulation

In the FE bearing model, the rollers are modeled with nonlinear spring elements to save computational time. Five spring elements represent one axial roller, and three spring elements represent one radial roller. Force distributed constraints connect the springs to the raceways. Three-row roller bearings typically have a segmented ring, in this case the outer ring, for assembling purposes. This segmentation is considered in the FE model and a frictional contact with a coefficient of friction of 0.2 is defined. To evaluate the relative deformation between the rings that the sealing is exposed to, the grooves to mount the sealing are implemented in the bearing rings. Figure 5 shows the mesh of the FE bearing model in a cross-sectional view. The lower segment of the outer ring is displayed in orange, the upper segment in red, and the inner ring in blue. The spring elements are shown in black. Arrows highlight the rows of nodes along the circumference of the rings that are used to evaluate the relative displacements.

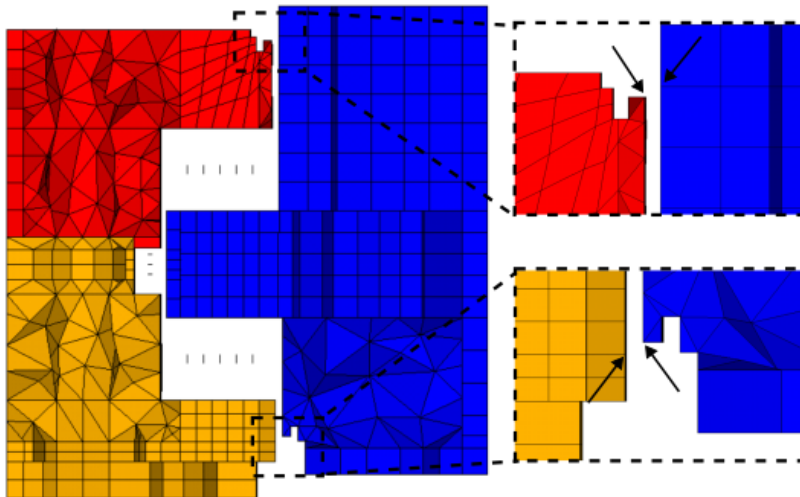
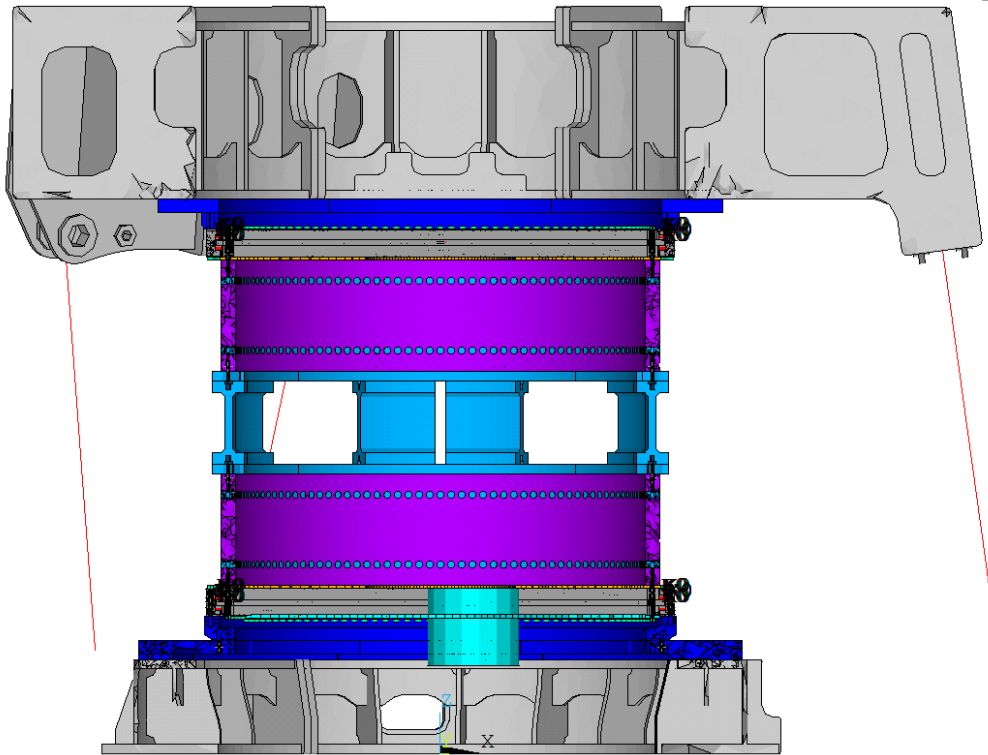


Figure 5: Cross sectional view of the FE bearing model

The FE test rig model, shown in Figure 6, contains every component of the test bench. The components are connected by modeled bolts. Frictional contacts are defined between touching surfaces. Linear actuator elements model hydraulic cylinders allowing for similar loading conditions as in the experiment. In the simulation, gravitational forces and bolt forces are applied in the first load step. The following load steps apply loads to the cylinders to load the bearings. The FE bearing and test rig model is validated against experimental data using strain gauges on the outer surfaces of the bearing rings. The comparison of the simulation with the experiment shows a similar characteristic of the ring deformation. The deviation of the maximum strain is less than 10%. More detailed information about the FE models and the validation can be found in (Graßmann et al. 2026). Two load cases, listed in Table 1, are simulated to investigate the resulting gap between the bearing inner and outer ring and therefore, the ring movements the sealing must compensate. The load case power production describes the blade root forces that act on the bearing for normal operation of the reference wind turbine IWT-7.5-162. Extreme load describes the loads for power production with occurrence of a fault, and the protection system of the turbine intervenes.

Table 1: Load cases for the FE simulations

Load case	Fx in kN	Fy in kN	Fz in kN	Mx in kNm	My in kNm
Power production	322	-270	465	6,505	15,683
Extreme load	634	-337	1,059	8,746	28,628



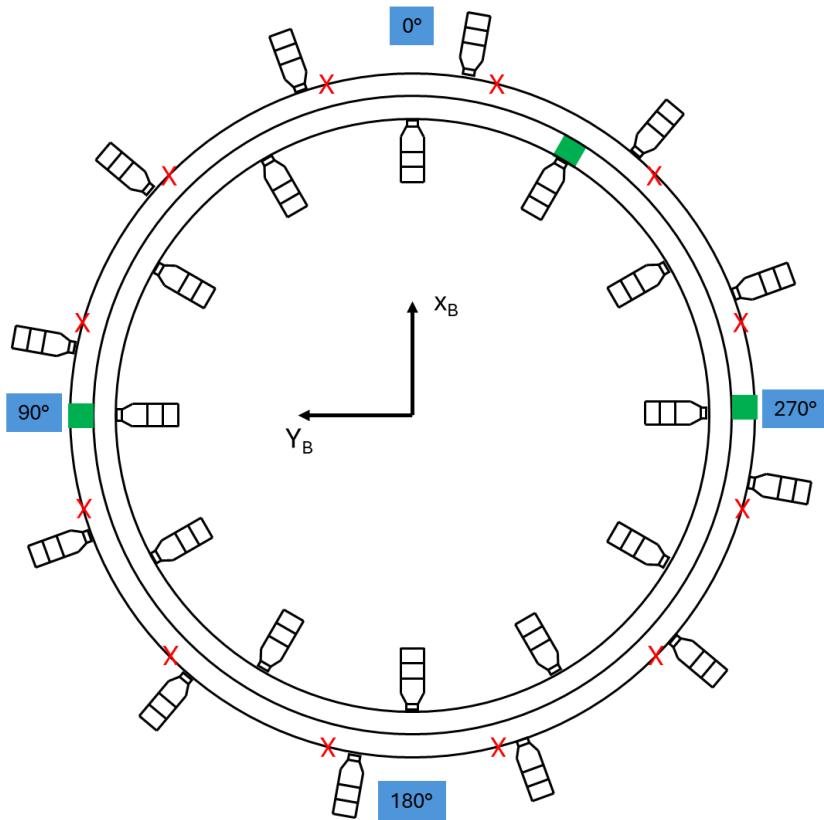
170 **Figure 6: FE model of the BEAT6.1**

2.4 Pressure measurement

To measure the pressure inside the bearing, a piezo-resistive ceramic pressure sensor from RS PRO is used. The sensor can measure relative pressure changes for viscous fluids in a range of 0 bar to 5 bar with an accuracy of $\pm 0.25\%$. In this work, a total of five pressure sensors are installed at the lower and upper bearing. Figure 12 shows an installed pressure sensor on the left.

175 All sensors are mounted through grease outlet holes to the rings of the bearings. In most cases, the sensors are directly screwed into the holes. For the first test, four sensors are installed at the outer ring of the lower bearing at 0° , 90° , 180° , and 270° . From the second test onwards, both the lower and the upper bearing are equipped with sensors. At the lower bearing, the sensors are positioned on the outer ring at 90° and 270° and for the upper bearing on the outer ring at 0° and 180° . One additional sensor is also equipped with a bottle after the sensor to investigate possible changes in the occurrence of pressure

180 when the grease does not face any resistance. In that case, the grease can exit the bearing, and the sensor would detect any pressure changes while the grease flows by. This combination of sensor and bottle to collect outcoming grease is installed at the inner ring at 330° of the lower bearing from test 2 onwards. Figure 7 shows the grease inlets and outlets and the position of the pressure sensors exemplarily for the lower bearing.



185 **Figure 7: Positions of grease inlets (red crosses), grease outlets (bottles) and sensors (green squares) for the lower bearing**

Different tests aim to measure the pressure inside the bearing under different conditions: Test 1 is a wear test that aims at testing real wind turbine behavior. The loads and pitch movements that are expected to be critical for the occurrence of wear on the raceways from the aerodynamical simulations of the IWT7.5 reference wind turbine are used to create that test. Detailed information about the test program can be found in (Stammler 2023). This test functions as a reference test that represents normal operational behavior of a roller-type blade bearing. Test 2 pumps grease in the bearing for six days without any loads or pitch movements. But the grease outlets of the lower bearing are closed except for the one on the inner ring at 330° where the sensor with the bottle to collect outcoming grease is located. Test 3 is directly following test 2 and applies static bending moments with different load levels to investigate the influence of the load on the pressure inside the bearing. The bending moment goes from $M_y = -35 \text{ MNm}$ to $M_y = 35 \text{ MNm}$ in 1 MNm steps. Test 4 is following test 3 and introduces a dynamic M_y ramp to the bearings. The ramp starts at $M_y = -35 \text{ MNm}$ and ramps up to $M_y = 35 \text{ MNm}$ in 1 MNm steps. At each load level, the inner ring of the bearings is pitched $\pm 20^\circ$. For all the tests, the grease outlets of the upper bearing remain open to investigate the differences on the occurring pressure in the bearing with and without the possibility of the grease exiting the bearing. Table 2 lists all the test scenarios to measure the resulting pressure in the bearings.

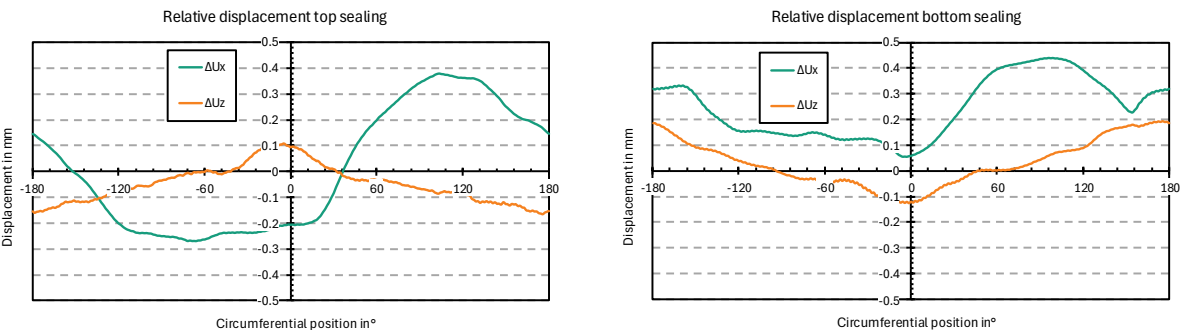
200 **Table 2: Tests for pressure measurements**

Test	Bearing	Grease outlets	Load	Pitch movement
1	Lower	Open	Dynamic loads representing real operational behavior	Dynamic pitch movements representing real operational behavior
	Upper	Open		
2	Lower	Closed, except at IR 330°	None	None
	Upper	Open		
3	Lower	Closed, except at IR 330°	My=-35MNm to My=35MNm in 1MNm steps	None
	Upper	Closed		
4	Lower	Closed, except at IR 330°	My=-35MNm to My=35MNm in 1MNm steps	+/- 20° at each load step
	Upper	Closed		

3 Results and discussion

3.1 FE simulation

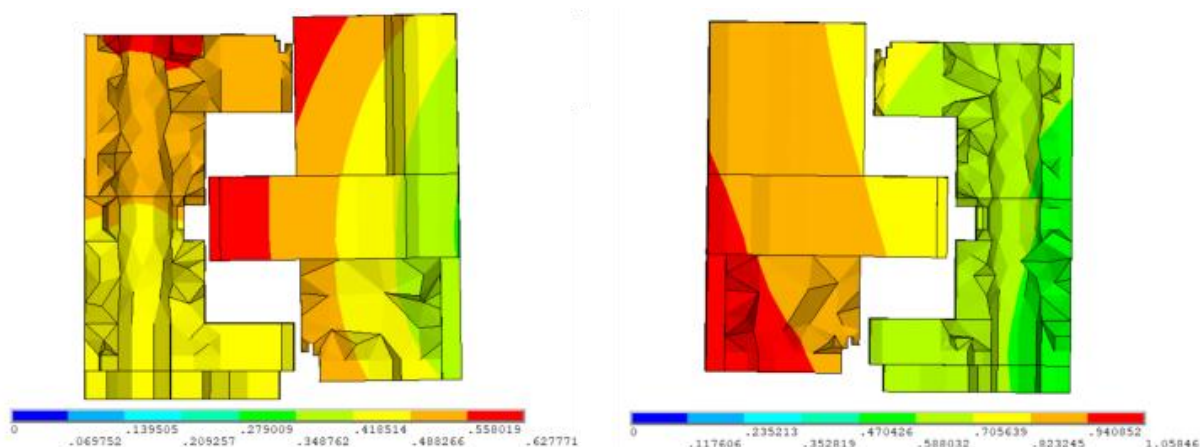
The results of the relative ring displacement along the circumference of the lower bearing for power production is shown in Figure 8. The top sealing is displayed on the left and the bottom sealing on the right. In both cases the radial deformation U_x and axial deformation U_z from the sealing groove is subtracted by the deformation of the opposite ring according to Figure 5. Positive values for the radial deformation ΔU_x indicate a gap opening between the rings and a negative value a gap closing. The maximum gap opening occurs at the bottom sealing with 0.44 mm. The maximum gap closing occurs at the top sealing with -0.27 mm. The maximum axial deformation is 0.19 mm at the bottom sealing.



210 **Figure 8: Relative ring displacement for power production**

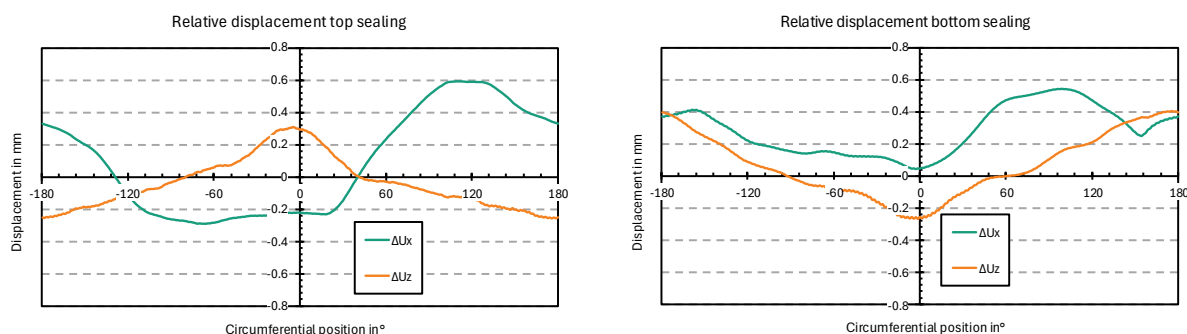


Figure 9 shows the deformation of the rings at -70° where the maximum gap closing occurs on the left and at 100° where the maximum gap opening occurs on the right for the power production load case. In both cases, the tilting of the rings that lead to an opening or closing of the gaps is visible.



215 **Figure 9: Deformation of the rings at -70° (left) and $+100^\circ$ (right) for power production**

Figure 10 shows the relative deformations of the bearing rings for the extreme load case. Here, the maximum gap opening occurs at the top sealing with 0.59 mm. The maximum gap closing is at the top sealing as well with -0.29 mm. The maximum axial deformation is 0.41 mm at the bottom sealing.



220 **Figure 10: Relative ring displacement for extreme loads**

Figure 11 shows the ring deformation again at -70° on the left and $+100^\circ$ on the right for the extreme loads. Compared with power production, the tilting of the rings and therefore the relative displacements of the rings are clearly larger.

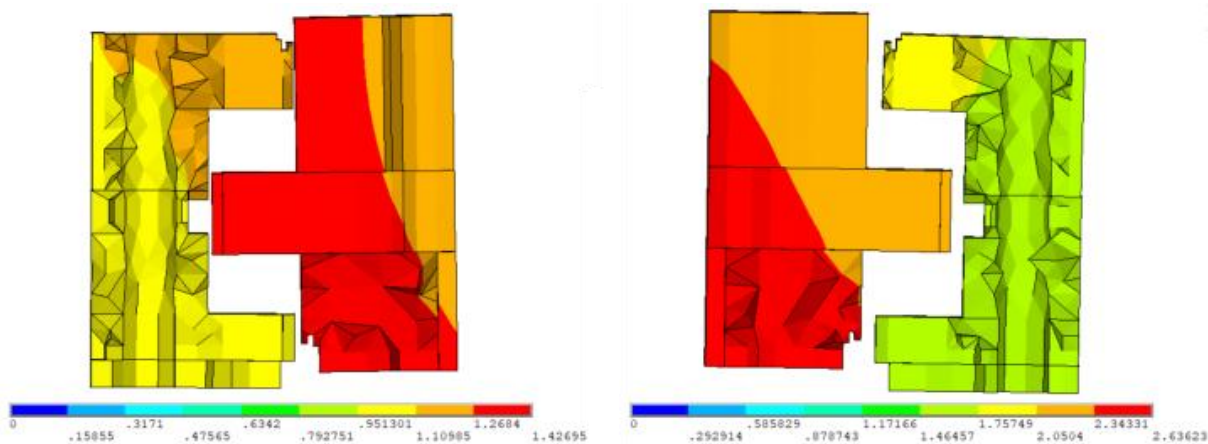


Figure 11: Deformation of the rings at -70° (left) and +100° (right) for extreme loads

Table 3 lists the maximum relative deformation of the sealing for both load cases from the FE simulations. As mentioned earlier, the sealing for this bearing was designed to allow for axial and radial movements of ± 1.5 mm. Hence, all occurring ring deformations even for extreme loads are easily covered by the sealing, preventing any grease contamination or leakage.

Table 3: Maximum relative deformation of the sealing

Load case	Sealing position	ΔU_x in mm		ΔU_z in mm	
		min	max	min	max
Power	top	-0.27	0.38	-0.17	0.11
production	bottom	0.06	0.44	-0.13	0.19
Extreme	top	-0.29	0.59	-0.26	0.31
load	bottom	0.04	0.54	-0.27	0.41

3.2 Pressure measurement

To ensure the capability of the sensor to measure any pressure changes with the grease used, a hand pump is attached to one of the sensors while it is connected to the bearing outer ring. Then, grease is pumped into the bearing by five pump strokes passing the pressure sensor. Figure 12 shows the hand pump attached to the sensor and the resulting measurement of the pressure. The five strokes are clearly visible with a maximum pressure of nearly 0.8 bar. It also shows that the pressure is reduced immediately as the grease flows into the bearing. This validates the capability of the sensor to measure pressures in the bearing with the grease.

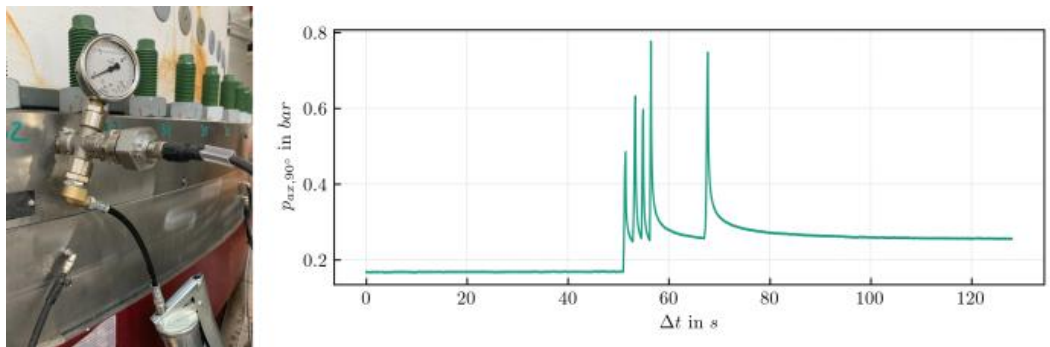


Figure 12: Hand pump attached to a pressure sensor and resulting pressure measurement

Test 1 represents normal operational behavior of a roller-type blade bearing. Four pressure sensors are installed at the lower bearing at the angular positions of 0°, 90°, 180°, and 270°. From the top to the bottom, Figure 13 shows the pitch angle of the test, the bending moment, and then the measured pressures at 0°, 90°, 180°, and 270°. It shows that the pressure is slightly changing over time. However, the maximum occurring pressure with dynamic load and pitch movement is 0.05 bar. Hence, no pressure increase occurs in the bearing under operational conditions even with significantly high amounts of grease pumped into the bearing.

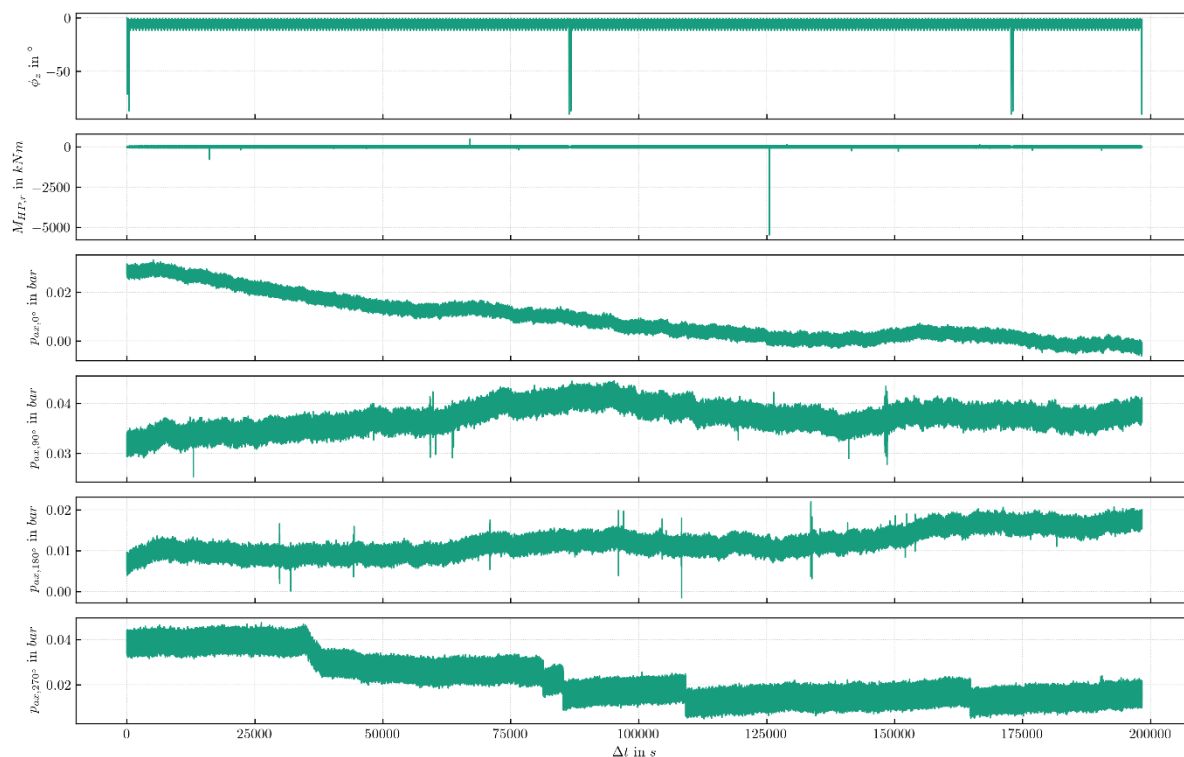


Figure 13: Pressure in the lower bearing during test 1



During Test 2 the grease outlets of the lower bearing (LDUT) are closed except for the one at 330° on the inner ring where the sensor connected to the collector bottle is installed. Figure 14 shows the results of the measured pressure as well as the temperature of the test hall and the bearings. The pressure development shows a significant increase when the grease outlets are closed with a maximum pressure of 2 bar. For the sensor where the collector bottle is attached only smaller fluctuations at a very small pressure can be observed. At the upper bearing, no pressure increase can be seen at all. The course of the pressure measured at 270° at the lower bearing (red line) does not seem to have reached a maximum within the test time which indicates that even higher pressures might be possible under these extreme conditions. The results clearly show that the pressure inside the bearing increases when the grease cannot exit the bearing and the rings are not pitching. However, for the upper bearing (UDUT) where grease outlets are open, no pressure increase is notable. Figure 14 also shows no dependency between the pressure and the temperature. During the test, the temperature of the upper and lower bearing remained almost constant. The temperature of the test hall has some fluctuations due to night and day. However, this characteristic is not shown in the development of pressure in the bearing. The results of test 2 clearly show the possibility of a significant increase in pressure which would influence the design of the sealing. However, the high pressure only occurred under extreme conditions with closed grease outlets and without any movement.

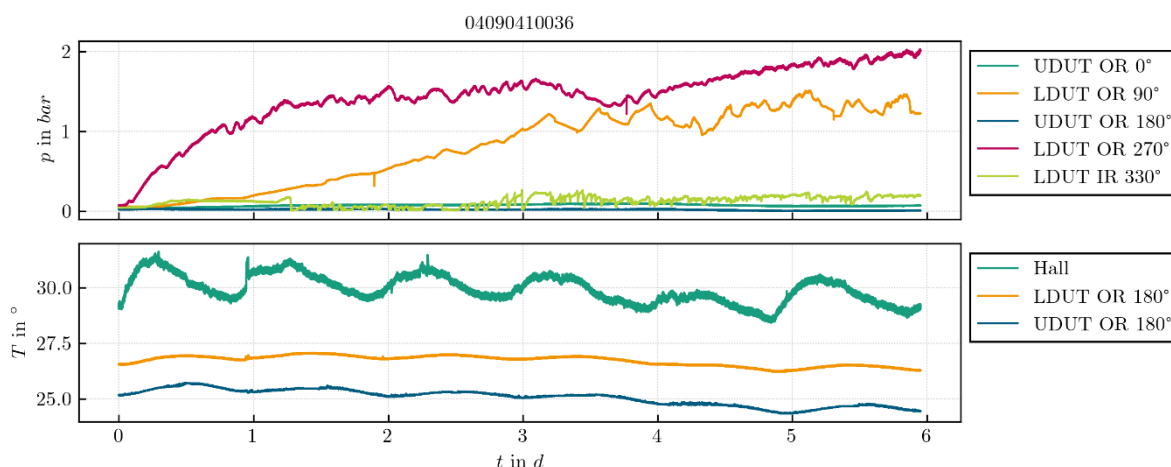


Figure 14: Pressure in the bearings during test 2

Test 3 follows test 2 to investigate the development of the pressure inside the bearing depending on the load level. Figure 15 shows the My ramp at the top and the resulting pressure at the bottom. Moving to $M_y = -35$ MNm slightly increases the initial pressure for the sensors with the high values. Then, for negative bending moments, the pressure almost stays constant. Only with positive bending moments, the pressure drops steadily. At the end of the test, the load goes back to 0 which causes a pressure increase to almost the initial pressure values. It is notable that test 2 stopped with the maximum pressure of 2 bar. However, test 3 starts at 1.1 bar. That means, that the time between the stop of test 2 and the start of test 3 (max 1 hour) without pumping grease in the bearing is enough to already reduce the pressure significantly. The pressure at 270° dropped



from 2 bar to 1.1 bar. The pressure at 90° dropped from 1.2 bar to 0.9 bar. The smaller pressure at 330° on the inner ring stayed at 0.2 bar. Therefore, at high pressures, simply stopping to pump grease in is enough to drop the pressure to roughly 1 bar. In addition, the pressure in the bearing depends on the acting load. However, the load is not enough to persistently reduce the pressure as it increases again as soon as the load is reduced.

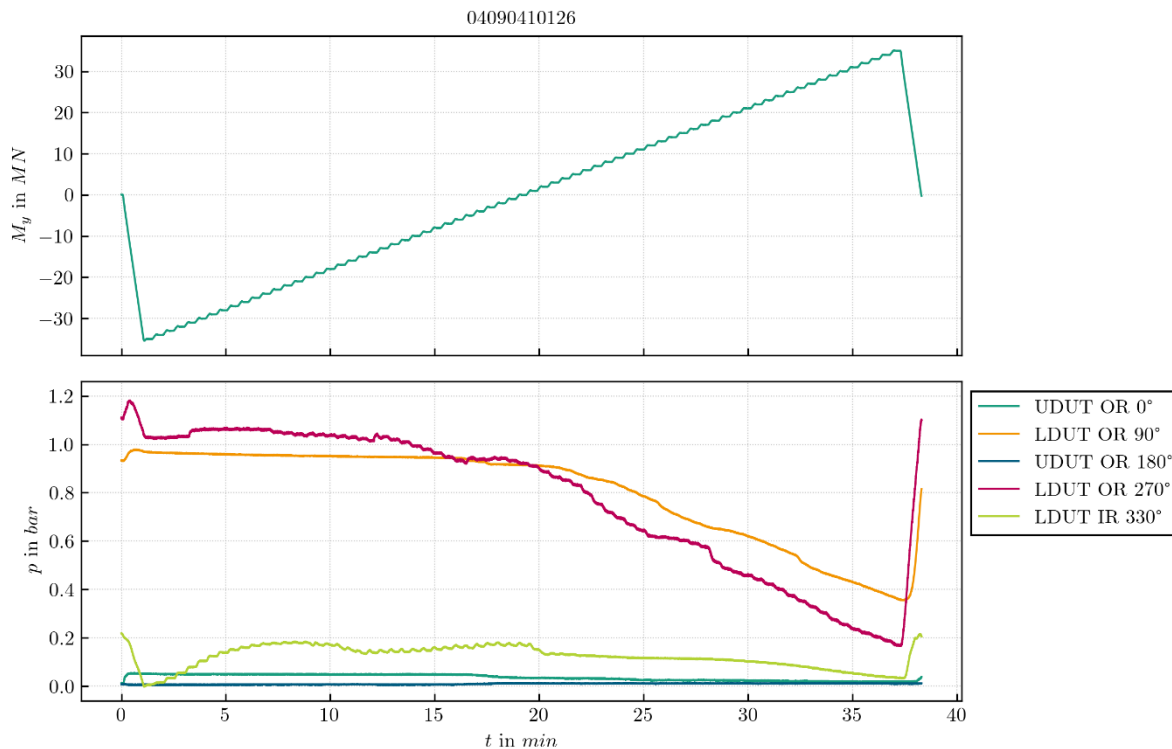


Figure 15: Pressure in the bearings during test 3

Test 4 investigates the pressure development when the bearing rotates at different load levels. Figure 16 shows the pitch movements at the top, the load levels in the middle, and the resulting pressure at the bottom. In contrast to the static load ramp, the pressure in the bearing seems to be reduced persistently when pitch movements are introduced resulting in maximum pressures below 0.5 bar.

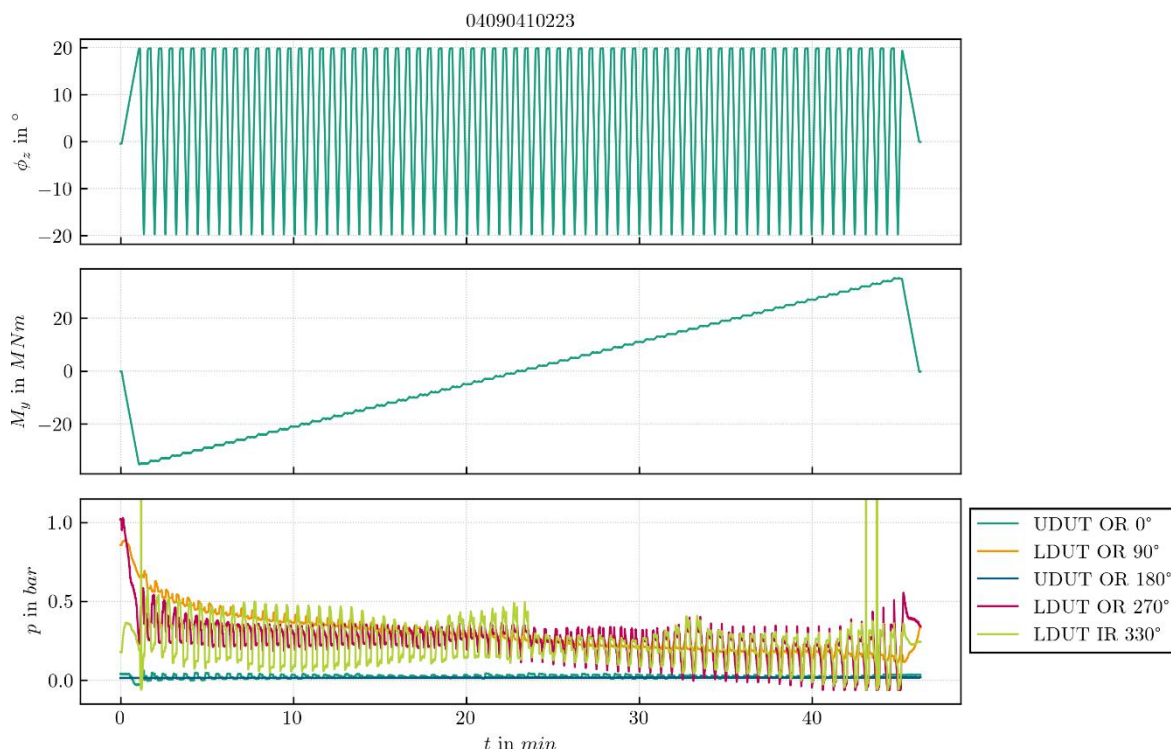


Figure 16: Pressure in the bearings during test 4

Combining all test results, increasing the pressure in a three-row roller bearing is only possible under extreme conditions. With closed grease outlets, without any movements and loads, and pumping as much grease as possible, a maximum pressure of 2 bar is reached after 6 days. It should be mentioned here that the sealing is designed to withstand a maximum pressure of 1.9 bar. However, even at 2 bar the sealing remained in place ensuring comparable test set ups during all following tests. With normal operational behavior of a blade bearing, no significant pressure increase is measured.

4 Conclusion

The design of the sealings for large slewing bearings highly depends on the maximum movements of the bearing rings and the maximum pressure inside the bearing. Especially for large bearings in wind turbines, the stiffness of the surrounding structures influences the ring deformations. The pressure inside the bearing is determined by the operational conditions of the bearings such as load, pitch movements, and grease flow. This work investigates extreme deformation and pressure conditions for a 5 m three-row roller blade bearing with respect to the sealing. The authors have accomplished a set of extreme values a sealing type needs to withstand during its use in a large roller-type blade bearing.



295 This work has used the Fraunhofer IWES blade bearing test rig BEAT6.1 to investigate the bearing ring behavior under normal operating loads and extreme loads. Specially designed adapters ensure realistic stiffnesses of the surrounding structures based on the Fraunhofer IWES reference wind turbine IWT7.5-162. In radial direction, the maximum gap opening for operating loads is 0.44 mm and the maximum gap closing is -0.20 mm. The maximum axial deformation is 0.20 mm. For extreme loads, the maximum gap opening is 0.59 mm and maximum gap closing -0.29 mm. The maximum axial deformation is 0.41 mm. The sealing used in this work was designed to cover axial and radial movements between the bearing rings of
300 ± 1.5 mm. Even under extreme conditions, the maximum occurring gap opening is less than half of the allowable. For the pressure measurements, the maximum amount of grease possible has been pumped into the bearings. The grease outlets of the lower bearing have been closed while the outlets of the upper bearing have remained open. The tests show that no significant pressure increase occurs during normal operational behavior even with the high amount of grease supply. Extreme conditions with closed grease outlets and without pitch movements show an increase in the pressure inside the
305 bearing up to 2 bar. Any loads that are applied to the bearings influence the pressure. However, loads without pitch movements are not enough to persistently reduce the pressure in the bearing. As soon as pitch movements are added, the pressure drops significantly and stays below 0.5 bar. When the grease outlets remain open, no pressure increase can be observed at all. For operational behavior in a wind turbine, no significant pressure increase is expected.

310 This work has investigated the ring deformation and internal grease pressure to assess the design of the sealing for its use in a 5 m three-row roller blade bearing of a wind turbine. The sealing has been a standard design for use in large slewing bearings across different applications. The results have shown that the used sealing can withstand maximum occurring conditions regarding ring deformation and pressure development of the blade bearing.

Further investigations will include different bearing types like ball bearings, different bearing dimensions, as well as different greases and operational loads and pitch movements.

315 **Data availability**

Data will be made available on request.

Author contributions

MG: conceptualization, investigation, methodology, validation, writing (original draft). MB: conceptualization, writing (review and editing). FS: writing (original draft). AB: investigation, writing (review and editing).



320 **Competing interests**

Authors Marlene Bantz and Frank Schönberg were employed by Freudenberg FST GmbH. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

Disclaimer

325 Copernicus Publications adds a standard disclaimer: “Copernicus Publications remains neutral with regard to jurisdictional claims made in the text, published maps, institutional affiliations, or any other geographical representation in this paper. While Copernicus Publications makes every effort to include appropriate place names, the final responsibility lies with the authors. Views expressed in the text are those of the authors and do not necessarily reflect the views of the publisher.” Please feel free to add disclaimer text at your choice, if applicable.

330 **Acknowledgements**

The authors gratefully acknowledge the contribution of Martin Geibel for plotting the measured pressures and providing the diagrams.

Financial support

335 This research received no public funding. The test rig infrastructure, test assembly and validated FE models were part of the research project HAPT2 supported by the Federal Ministry for Economic Affairs and Energy under grant number 03EE2033A.

340

References

Bacchetto, A.: Advanced numerical Advanced numerical simulations for optimized seal designs. SKF.
https://cdn.skfmediahub.skf.com/api/public/0901d196802520c2/pdf_preview_medium/0901d196802520c2_pdf_preview_medium.pdf, 2013.



- 345 Becker, D.; Handrek, T.; Lüneburg, B.: Gekoppelte thermische Finite Element-Analyse und Simulation der Gekoppelte thermische Finite Elemente-Analyse und Simulation der Laufbahnbeanspruchung an einem 6 MW Rotorlager. In: *Tribol. und Schmierungstechnik* 61. Jahrg. 1/2015, 2014.
- Graßmann, M.; Geibel, M.; Schleich, F.: Validation of a finite-element model of a 5 m three-row roller wind turbine blade bearing. In: *Wind Energ. Sci.* 11 (1), S. 251–264. DOI: 10.5194/wes-11-251-2026, 2026.
- 350 Harris, Tedric A.; Kotzalas, Michael N.: Rolling bearing analysis. Advanced Concepts of Bearing Technology. 5th ed. Boca Raton FL: CRC/Taylor & Francis, 2007.
- He, P.; Liu, R.; Hong, R.; Wang, H.; Yang, G.; Lu, C.: Hardened raceway calculation analysis of a three-row roller slewing bearing. In: *Int. J. of Mech. Sci.* 137, S. 133–144. DOI: 10.1016/j.ijmecsci.2018.01.021, 2018.
- Khan, Muhammad Ishaq; Maccioni, Lorenzo; Concli, Franco: Investigating Grease Behaviour in Tilted Double-Row Tapered Roller Bearing Installed in Wind Turbine by Developing a Full Scale Multi-Phase CFD Model. DOI: 10.5194/wes-2025-97, 2025.
- 355 Popko, W.; Thomas, P.: IWES Wind Turbine IWT-7.5-164 Rev 4. In: *Fraunhofer Inst. for Wind Energy Syst.* DOI: 10.24406/IWES-N-518562, 2018.
- Stammler, M.: Endurance Runs for Pitch Bearings of Wind Turbines. In: *Fraunhofer Verl.*, 2020.
- 360 Stammler, M.; Baust, S.; Reuter, A.; Poll, G.: Load distribution in a roller-type rotor blade bearing. In: *J. Phys.: Conf. Ser.* 1037. DOI: 10.1088/1742-6596/1037/4/042016, 2018.
- Stammler, Matthias: Wear test programs for roller-type pitch bearings of wind turbines. In: *Wind Energ. Sci.* 8 (12), S. 1821–1837. DOI: 10.5194/wes-8-1821-2023, 2023.
- Stammler, Matthias; Schleich, Florian: Load case selection for finite-element simulations of wind turbine pitch bearings and hubs. In: *Wind Energ. Sci.* 10 (4), S. 813–826. DOI: 10.5194/wes-10-813-2025, 2025.
- 365 Wang, H.; He, P.; Pang, B.; Gao, X.: A new computational model of large three-row roller slewing bearings using nonlinear springs. In: *J. Mech. Eng. Sci.* 231 (20), S. 3831–3839. DOI: 10.1177/0954406217704223, 2017.
- Zhang, Zekun; Gao, Xuehai; Ren, Sizhe; Ma, Cheng: Effect of Flexible Support Structures on Sealing and Raceway Damage of Pitch Bearings in Wind Turbines. In: *Machines* 14 (2), S. 186. DOI: 10.3390/machines14020186, 2026.