

The influence of oscillation sequences on rolling bearing wear

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Abstract– Oscillating applications for rolling bearings include construction machinery like cranes and excavators and wind turbine pitch and yaw bearings. Due to the widespread use and the specific challenges of wind energy, oscillating applications have gained increased attention by the scientific community in the last years. In wind turbines, the amplitudes, their mean values and external loads of oscillations vary. In this paper, the authors examine the effect of different sequences of oscillations on rolling bearing wear by the means of tests with 7220 type bearings. Loads and kinematic parameters are downscaled from wind turbine applications; the lubricant is a fully formulated grease.

To compare different sequences of oscillations, a reference test with 40.000 cycles of 3.95° double amplitude and 2.5 GPa contact pressure was varied. Constant, small amplitudes produced clearly visible damage patches on the bearing rings. The bearing rings had no visible surface damage if larger movements of 79° each 10 cycles interrupted the small-amplitude movements. We named such large movements protection runs. Protection runs are effective, even if their movement is interrupted by pauses or small reverse movements. These results show that the sequence of varying oscillations has a significant influence on wear risk.

Keywords – wear, oscillation, rolling bearings, pitch bearings

1. Introduction

Rolling bearings rotate constantly in most industrial applications. Doing so, they are able to build up a lubricant film that separates the rolling elements from the raceways. If properly mounted, lubricated and maintained, damages of rolling bearing in rotating applications limit mostly to rolling contact fatigue. This damage mode was subject of various scientific and engineering research projects during the last decades. [1]

In some applications, the bearings do not execute full rotations. Such applications include slewing bearings for cranes and building machinery, headsets of bicycles and wind turbine bearings, namely pitch and yaw bearings. With a high number of oscillations, the large dimensions and the varying dynamic loads, wind turbine pitch bearings (see Figure 1) face the most challenging conditions [2, 3].

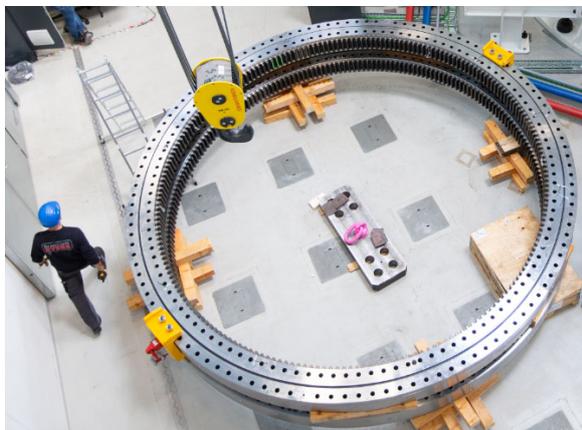


Figure 1: IMO type 12480 pitch bearings with 5m diameter

Applied research on oscillating bearings is mostly limited in two ways: The dimensions of the bearings and

the variance in operating parameters. For economic reasons, bearings with less than 100 mm outer diameter are in widespread use. For similar reasons, most test rigs for oscillating applications use constantly rotating drives and eccentric rods to convert the rotation into oscillations with constant amplitudes. This is valid for scientific works [4–6] and standardized tests like NFT and ASTM [7, 8]

First works range back to the year 1937, when ALMEN reported on wear phenomena in ostensibly motionless car hub bearings during ship transport [9]. PITROFF [10] and later BREWARD [11] focussed on similar applications.

THIEDE [12] and BOSSE [13] published test results on oscillating bearings with grease lubrication. The varied operating parameters included temperature, amplitude and frequency.

The universities of Hanover [14, 15], Magdeburg [6] and Mannheim [16, 17] (in alphabetical order) contributed to the science of oscillating bearings in the recent years. For their research they used grease-lubricated bearings. The test rigs were able to vary load, amplitudes and frequencies. In none of the publications, the amplitude varied within individual tests.

GODFREY [18], ERRICHELLO [19] and SCHWACK [14] took on the task of establishing a clear differentiation between the main damage modes False Brinelling and Fretting Corrosion.

Several works sponsored by the Forschungsvereinigung Antriebstechnik underline the influence of tribolayers on the formation and growth of wear damage. A tribolayer is a thin layer of reactive products that forms on the raceway. This layer is less prone to adhesive wear in rolling contacts and thus protects the raceway. Formation of tribolayers depends on three conditions: Presence of reactive components (among them, oxygen and additives of the lubricant), energy supply and the right time windows for the reaction. The energy supply is a rolling body passing a certain

area of the raceway under load. The time depends on how soon the next rolling body passes the same area. [20–23]

The current state of science on oscillating applications does not take into account the formation of tribolayers. This makes perfect sense, as most applications, the causes for False Brinelling and Fretting Corrosion (like standstill marks) come at constant amplitudes. Tribolayer formation relies on larger movements that uncover and cover certain areas of the raceway to allow chemical reactions. Only recently, advanced load control mechanisms of wind turbines changed this picture, as they introduced a substantial variance in amplitudes and loads at a very high number of cycles [24]. A fierce competition in the wind turbine market and the drastically falling prices for turbines make load control mechanisms a must-have for current and future wind turbines. While most issues about the application are solved, the pitch bearings, which connect rotor hub and rotor blades and allow the turning of the blades to control aerodynamic loads, still call for further investigation.

In particular, a better understanding of the influence of varying amplitudes on wear phenomena motivates the present work. Unlike previous works, the tests we describe comprised different sequences of amplitudes.

2. Methodology

2.1. Bearings

We used type 7220 angular contact ball bearings for this work. Table 1 lists their main properties. The bearings have a plastic cage and no integrated sealings. The bearings are grease-lubricated, the grease is a fully formulated grease typically used in wind-turbine pitch bearings. The grease contains solid lubricants and extreme pressure additives, the soap is lithium-based and the base oil viscosity is 50 mm²/s at 40°C.

Table 1: Type 7220 bearing properties

Property	Unit	Value
Outer diameter	mm	180
Inner diameter	mm	100
Ball diameter	mm	25.4
Contact angle	°	40
Number of balls	-	15

With an outer axial load of 90 kN, the contact pressure P equals 2470 MPa (calculations according to [1]), and the width of the contact patch $2b$ is 0.8882 mm.



Figure 2: 7220 bearing on IMO type 12480 pitch bearing

2.2. Test rig

The test rig combines static axial load application with a servo drive that can realize any movement from very small oscillation to full rotations. Figure 3 and Figure 4 give overviews of the design. A manual hydraulic pump applies pressure to the rigging cylinder (lime colour in Figure 3) which translates the pressure into axial force. The load application suffers from two parasitical load paths, one being the internal friction of the cylinder and the other one a load transfer from the outer ring of the cylinder-side bearing to the housing of the mechanical assembly. For the test purpose of understanding the influence of movement patterns, it is not mandatory to determine the absolute value of the load, the certainty to apply loads at a magnitude that causes surface wear suffices.

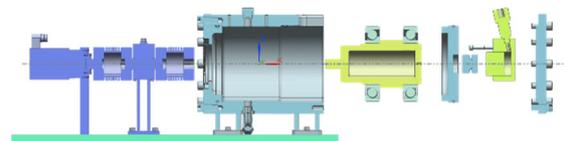


Figure 3: Test rig section view

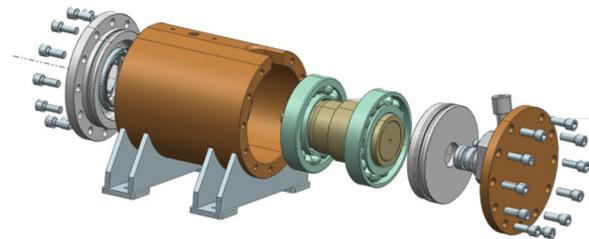


Figure 4: Mechanical assembly of test rig

The test rig has a torque and position measurement, the maximum torque is 50 Nm.

2.3. Test plan

In previous publications, the authors stated the importance of the relation between the ball travel distance x and the contact width $2b$ when describing oscillating movements of bearings. The ratio $x/2b$ is independent of bearing sizes and allows scaling of oscillatory movements. [2, 3, 25]

Based on previous experiences in pitch bearings tests, we defined a reference travel distance of 3.95° which corresponds to an $x/2b$ of 4.7. In pitch bearings, this ratio results in pitch angles between 0.4 and 1° (depending on the diameter), which are typical angles during operation. With this reference travel distance, we defined a reference test with 40.000 cycles, an axial load of 90 kN, and a speed of 4°/s (Test ID 1, see Figure 5).

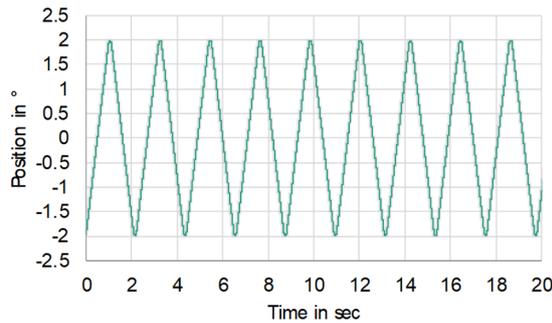


Figure 5: Motion pattern in reference test

Table 2 lists all tests. The first column contains an ID, which identifies the individual test. All tests stem from the reference test and vary the amplitude or add additional longer movements (protection runs). Test ID 4 for instance contains the same 40.000 cycles of 3.95° double amplitude, but adds an additional oscillation with 19.75° travel each 40 cycles. Table 2 lists all tests. Test ID 2 has an increased double amplitude of 8.74° (corresponding to an $x/2b$ of 12) to evaluate a possible upper limit of wear creation.

Test ID 3 attempts to vary test ID 1 and 2 in a way that is more representative for pitch bearings. Pitch bearings seldom operate with constant amplitudes, as variations in the incoming wind flow of the turbine call for different pitch angles. Thus, this test has random double amplitudes. All of these share the upper turning position, as this behaviour reflects certain types of wind turbine controllers.

Test IDs 4 to 8 include different variants of protection runs. ID 4 has a double amplitude of 19.75° . This amplitude is 5 times higher than that of the reference oscillations. To the knowledge of the authors, there are no values for effective protection runs available, as such this value was chosen arbitrarily. This test resulted in damaged raceways, and in consequence, the amplitude of the protection runs was set to a significantly higher value (79°) for test IDs 5 to 8. In addition to these tests, one test with full rotations at a speed of $4^\circ/s$ and the same load ensured that the test conditions do not cause wear if the bearing does not oscillate.

Table 2: Test plan

Test ID	Description	Visualization
1	Constant oscillations 3.95° Bearing IDs 0005, 0006	
2	Constant oscillations 8.74° Bearing IDs 0019, 0020	

Test ID	Description	Visualization
3	Random double amplitudes between 3.5 and 17.4° Bearing IDs 0025, 0026	
4	Protection runs (19.75°) each 40 cycles Bearing IDs 0007, 0008	
5	Protection runs (79°) each 30 cycles Bearing IDs 0009, 0010	
6	Protection runs (79°) each 10 cycles Bearing IDs 00021, 0022	
7	Protection runs (79°) each 100 cycles Bearing IDs 00023, 0024	
8	Interrupted protection runs (79°) each 30 cycles Bearing IDs 0017, 0018	

2.4. Inspection of bearings after test

The objective of this work is to understand if protection runs can prevent wear damage on the raceways and rolling bodies of the bearings. A visual inspection of the raceways after test provides all necessary results. An optical microscope is used for this inspection, as it allows measuring the size of damage patches and gives a good visual impression. In case of visible damages, each damage patch on the inner ring of the bearing is inspected.

3. Results and discussion

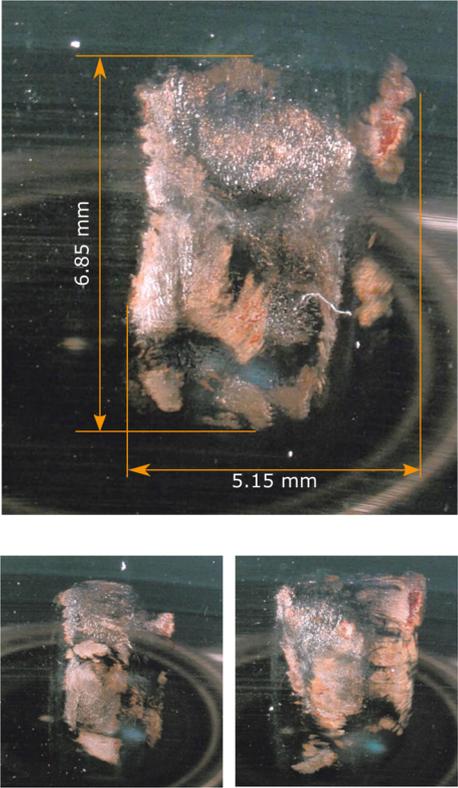
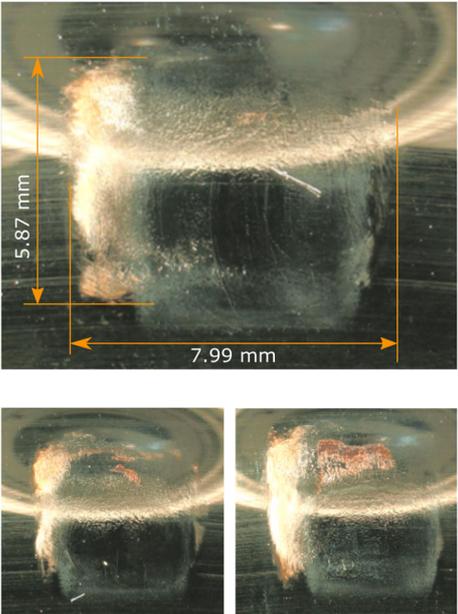
Each test ran with two bearings. The two bearings that ran with continuous rotations did not display any wear on the raceways. The following tables contain three representative results for each test. Note that the surface of the bearings shows a reflection of the microscope lens on all pictures.

Table 3 shows damage patches for tests without protection runs. These are the tests with the ID 1, 2, and 3.

Each bearing shows a clearly visible damage patch for each rolling contact. The patches contain corrosion products and have an increased surface roughness. Test ID 2 has more than twice the travel compared to

test ID 1; the severely damaged area does not increase accordingly. Test ID 3 displays the most severe damage on the raceways, with a damage concentration on the turning point (left in all pictures) shared by all cycles.

Table 3: Test results (tests without protection runs)

Test ID	Raceway condition
1	
2	

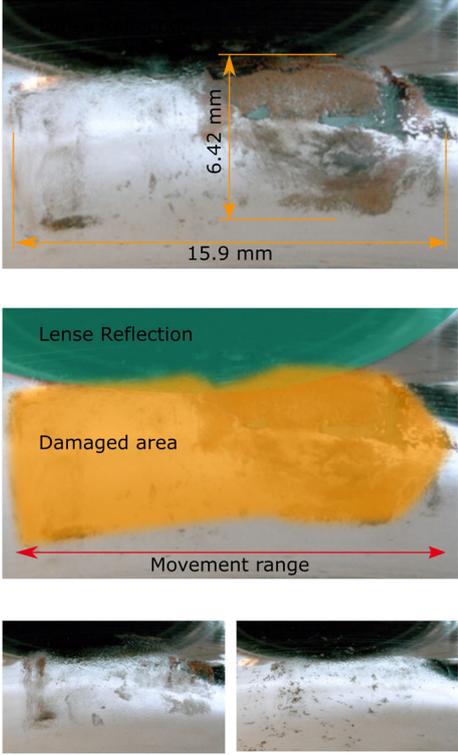
Test ID	Raceway condition
3	

Table 4 shows damage patches for tests with protection runs. These are the tests with the ID 4 to 8.

Test ID 4 resulted in clearly visible damage areas with corrosion products. In some of the areas, flakes broke out of the raceway surface.

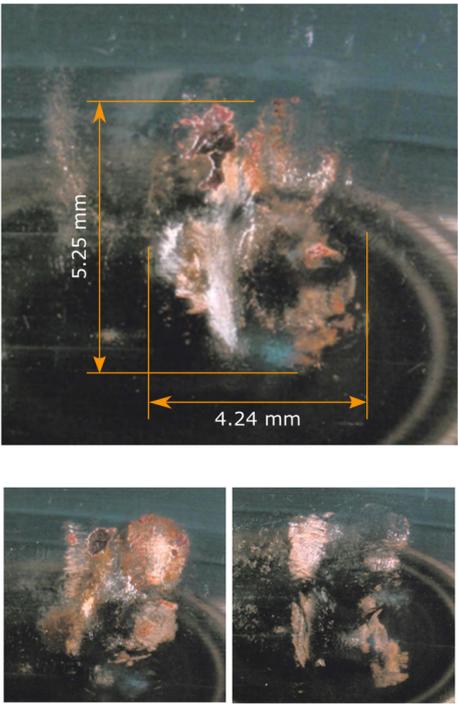
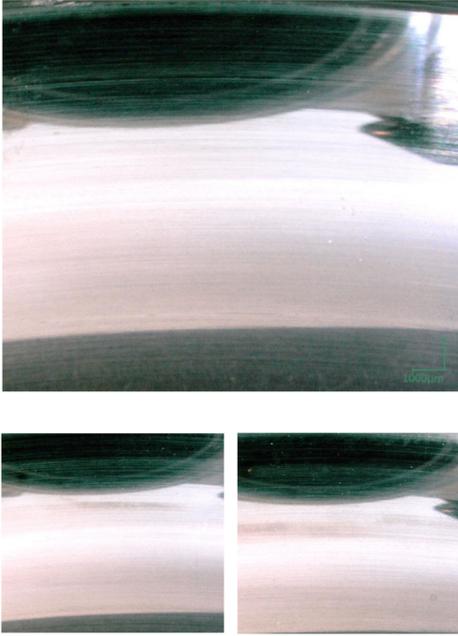
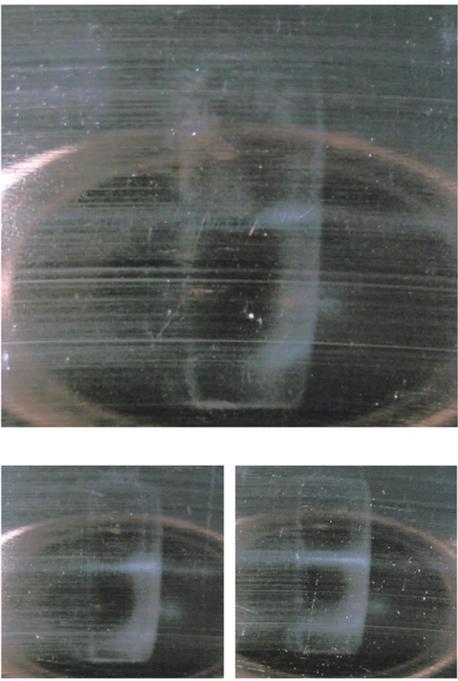
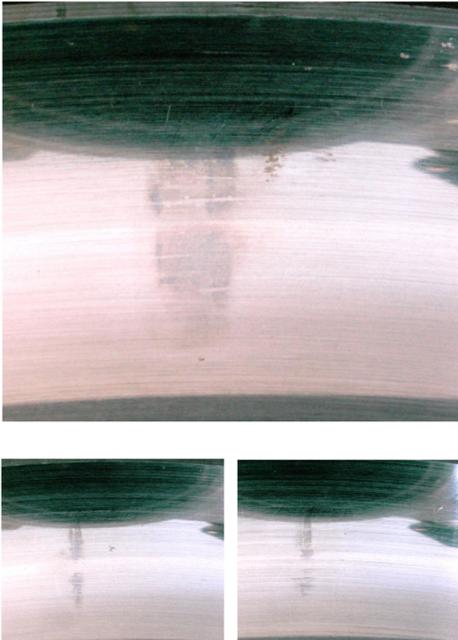
Test ID 5 displays no corrosion products. The visible change on the raceway is limited to a grey area. The shape is the outline of half an ellipse and the perpendicular lines that fade out in rolling direction. The manufactured surface structure seems unchanged.

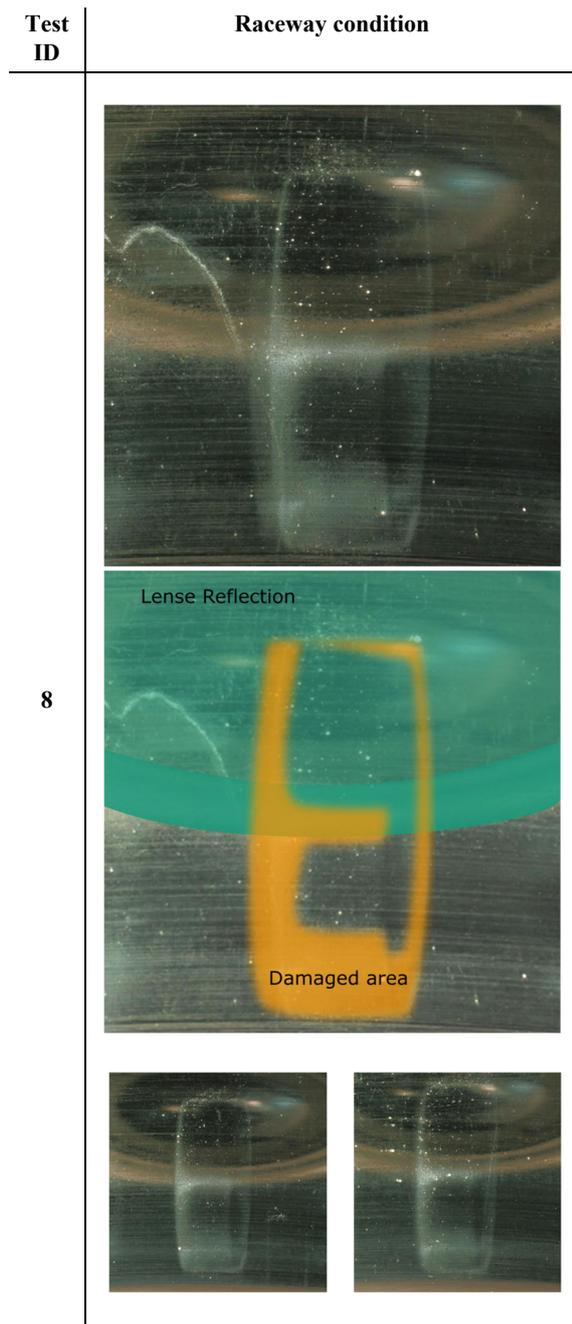
Test ID 6 does not show any changes to the raceway, similar to bearings with full rotations.

Test ID 7 shows similar results as Test ID 5. In two cases, small patches with increased roughness are visible.

Test ID 8 does not display corrosion products. The grey areas have a similar shape to Test ID 5.

Table 4: Test results (tests with protection runs)

Test ID	Raceway condition	Test ID	Raceway condition
4		6	
5		7	



All tests without protection runs resulted in clearly visible damage marks after 40000 cycles. These damage marks contain corrosion products (hematite); the damage mode is fretting corrosion. According to [14] and [19], unlubricated conditions cause fretting corrosions. In relation to the travel of the ball on the raceway, test ID 2 displays a less severe damage than test ID 1. The authors conclude that at some point during the movement with $x/2b$ of 12, lubricant enters the contact and prevents further fretting. The existence of fretting under these conditions indicates that any $x/2b$ equal or smaller than 12 does not suffice for a protection run.

Among the tests with protection runs, only the last four were able to prevent fretting corrosion. This most likely related to the travel of the protection run, which

was shortest in test ID 4. The number of cycles in between protection runs seems less relevant for damage formation, as test IDs 5,6, and 7 demonstrate.

Test ID 4 showed, in addition to the fretting corrosion, broken flakes on the raceway. For any practical application, this test result is the most critical, as the flakes can cause more damage rapidly if they get into contact areas.

Test ID 8 showed similar results to 5. The protection run is effective, even if the movement contains interruptions. The interruptions were done at travels similar to the 3.95° of the 40.000 reference cycles. This indicates that it is not necessary to build up a lubricant film to provide the protective capacity of a larger movement. Lubricant films build up with constant, long movements. In the absence of a lubricant film, it is likely that tribolayers provide the protective effect.

4. Conclusions

This work attempts to confirm the concept of protection runs for bearings in oscillatory operation. Protection runs are large movements, which are embedded in small oscillations, and protect the raceway from wear. Protection runs are particularly relevant for pitch bearings of wind turbines that operate with varying oscillation amplitudes.

To the knowledge of the authors, no publicly available test results have previously confirmed the validity of this concept. Most available research is limited to tests with constant amplitudes.

To confirm the concept, a new test rig for 7220 type angular ball contact bearings executed tests with different oscillation sequences. The test rig has a servo drive that allows execution of any desired movement. The load on the bearings was an axial load of 90 kN.

With the results of eight different tests, each of which was carried out with two bearings, we could confirm the effectiveness of protection runs. In reference tests with constant amplitudes, the raceway displayed fretting corrosion. Tests with effective protection runs completely eliminated this Fretting Corrosion. The results indicate that a minimum travel to achieve a protective effect. Runs with shorter travel are not effective, but might even add damage to the raceway. The number of cycles in between protection runs has a minor influence on the results, with more frequent protection runs resulting in less prominent grey areas on the raceway.

Protection runs do not need to be continuous; the movement may contain interruptions.

Future work in this field will cover further variation of the travel of protection runs, the frequency, and the number of cycles in between in order to further understand the influence of these parameters. We will transfer selected test sequences to tests with 800 mm and 5 m diameter slewing bearings.

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