Effect of Long-Term Inactivity on Railcar Bearings – Year 1

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16. Abstract

The performance of railroad bearings that sit idle in railyards, large industrial plants, or shipping ports has not been previously explored. Some of the bearings, with documented periods of inactivity exceeding 18 months, have been associated with major derailments. The aforementioned has led to concerns in whether the inactive periods contributed to early failure, possibly through degradation of the grease properties brought on by moisture intake or grease separation leading to uneven protection of the metal components. The work conducted in collaboration with CSX Transportation and MxV Rail, and in consultation with the National Transportation Safety Board (NTSB), aims to answer the question of whether long-term inactivity has significant effect on bearing performance and service life, and whether these are tied to changes in the lubricant.

The performed work consisted of (a) identification of installed bearings on cars that have not moved for periods of three months or longer, (b) removal of the bearings with minimum disruption to the lubricant, (c) pre-test inspection of the still-assembled bearings, (d) installation and service life testing on a laboratory test rig, with continuous performance monitoring of temperature rise, vibration spectra, and power consumption, (e) post-test inspection including disassembly, teardown, and visual inspection of all bearing components, and (f) analysis of the grease composition with specific focus on loss of oxidation inhibitors and evidence of lubricant separation.

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List of Abbreviations

ABD	Acoustic Bearing Detector
AAR	Association of American Railroads
HBD	Hot Bearing Detector
NTSB	National Transportation Safety Board
RMS	Room-Mean-Square
RPM	Revolutions Per Minute
USDOT	U.S. Department of Transportation

Disclaimer

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1. Introduction

The performance of railroad bearings that sit idle in railyards, large industrial plants, or shipping ports has not been explored. Some of the bearings, with documented periods of inactivity exceeding 18 months, have been associated with major derailments. This provides a basis for concerns that the inactive periods contributed to early failure, possibly through degradation of the grease properties brought on by moisture intake or grease separation leading to uneven protection of the metal components. Motivated by this, the University Transportation Center for Railway Safety (UTCRS) at The University of Texas Rio Grande Valley (UTRGV) in collaboration with the National Transportation Safety Board (NTSB), MxV Rail, and CSX Transportation seek to improve our understanding of the effects of long-term inactivity on bearing performance and the remaining service life.

2. Summary

CSX Transportation located railcars that sat idle for three years. Before these railcars were scrapped, the bearings were carefully removed from the inactive railcars and handled without rotating the bearings. A total of ten bearings were sent to UTRGV. Four of these bearings were pre-opened and grease collected and analyzed, and the other six bearings were still fully assembled, sealed, and marked for testing on one of the four-bearing test rigs available at the UTCRS-UTRGV bearing testing facilities. All bearings received at UTRGV were Association of American Railroads (AAR) Class F bearings. These bearings were carefully marked based on location on the railcar and testing was planned in three distinct experiments using two bearings per experiment. The results provided in this report are from the first two inactive bearings tested marked R7 and L7. This experiment started on March 25, 2024, and ended on June 24, 2024.

3. Test Setup, Instrumentation, and Measurements

3.1 Test Axle Setup

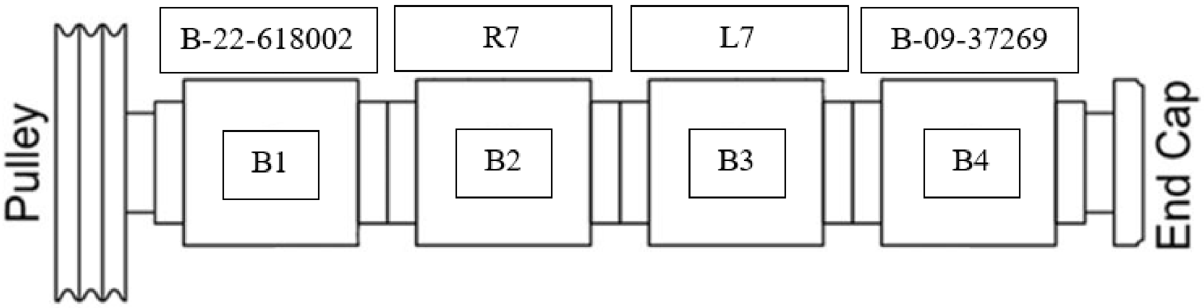

Figure 1: Schematic of the bearing setup showing their positions on the test axle

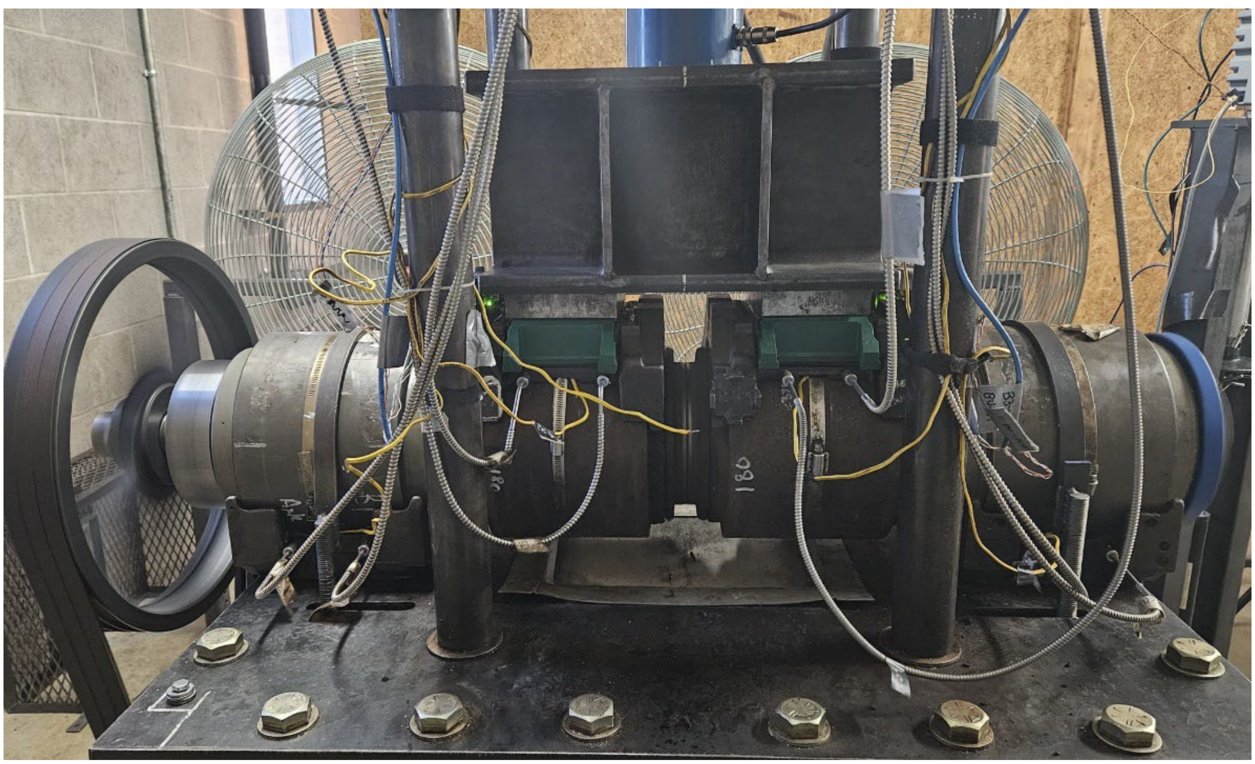

Figure 2: Picture of the UTRGV-UTCRS four bearing tester (4BT) used for this testing

3.2 Instrumentation

3.2.1 Vibration Sensors

There are two accelerometers that monitor and record the vibration levels within each test bearing to assess their condition. Both accelerometers are positioned so that they acquire radial direction acceleration. The outermost accelerometer is called the HUM Boomerang (black shell) and wirelessly records data at a frequency of 5.2 kHz for 4 seconds every 10 minutes throughout the duration of the test. The wired accelerometer behind the HUM Boomerang is called the Smart Adapter (SA) accelerometer and acquires data at a frequency of 5.12 kHz for 16 seconds every 10 minutes throughout the duration of the test.

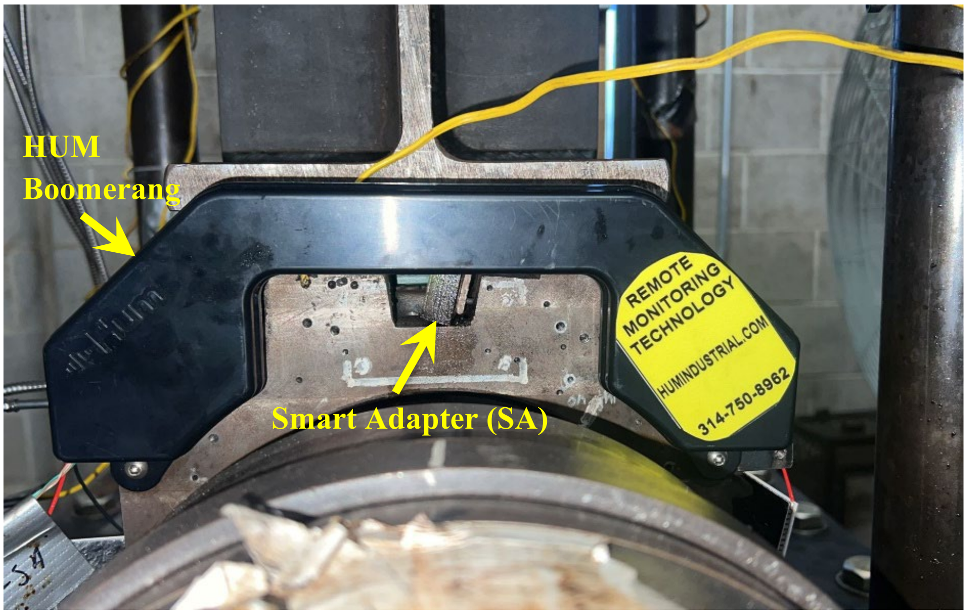

Figure 3: HUM Boomerang and UTCRS Smart Adapter accelerometer setup

3.2.2 Temperature Sensors

There are two bayonet-type (spring-loaded) thermocouples measuring the temperatures of the two cup (outer ring) raceways of each test bearing. The bayonet profiles shown in the temperature history plots are the average of the two bayonet thermocouples on each test bearing. An additional K-Type thermocouple that is held tightly in place against the cup surface using a hose clamp and is located at the same elevation as the two bayonet thermocouples measures the temperature at the middle of the cup. The temperature measurements are taken at 100 Hz every 15-second interval, and the temperature data is averaged to produce one data point at each 15-second interval.

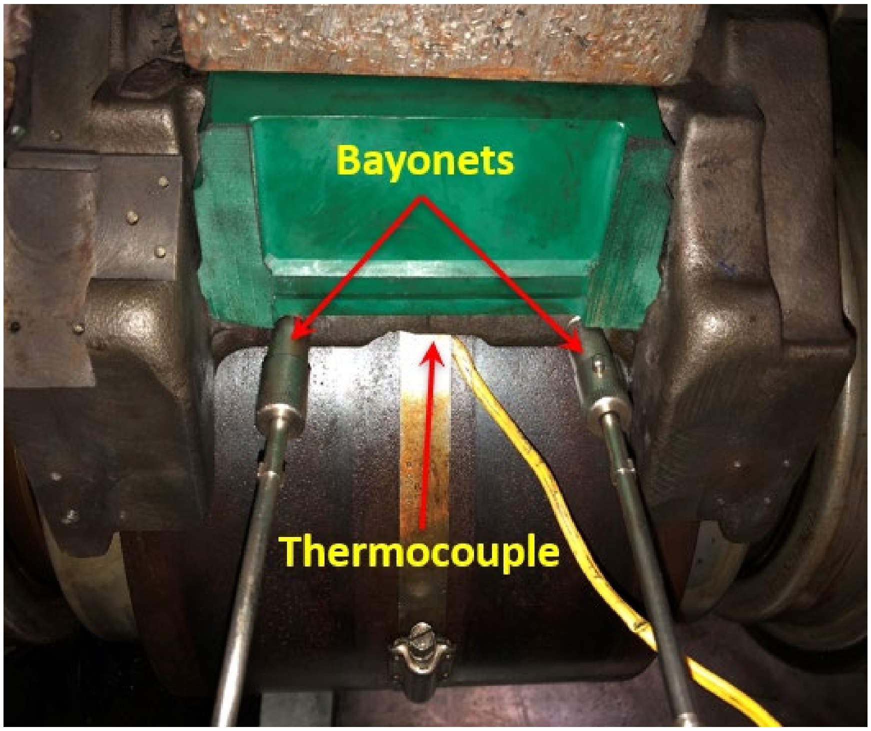

Figure 4: Temperature sensor instrumentation setup

3.3 Pre/Post Test Measurements

The bearings used for each test undergo careful documentation of all measurements pre and post testing. All measurements follow Association of American Railroads (AAR) standards and guidelines. Please refer to Figure 1 for bearing position on the test axle. In Table 1, the "MIN" and "MAX" refer to the minimum and maximum unmounted lateral bearing measurements, respectively. For mounted laterals, investigators made sure that the bearings are freely rotating if the values were zero. In Table 2, the amount of grease loss during testing in ounce (oz) and grams (g) is provided. The % loss provided is based on the total amount of grease packed in a new AAR Class F bearing, which is 22 oz (623.69 g).

Table 1: Mounted and unmounted bearing lateral spacings pre and post testing

Mounted and Unmounted Laterals Pre- and Post-Test								
Bearing	Unmounted Laterals				Mounted Laterals			
	Pre-Test		Post-Test		Pre-test		Post-Test	
	MIN	MAX	MIN	MAX	MIN	MAX	MIN	MAX
B1 - Control	0.021	0.022	0.014	0.016	0.000	0.000	0.000	0.000
B2 - R7	0.000	0.001	0.000	0.000	0.000	0.000	0.000	0.000
B3 - L7	0.015	0.017	0.010	0.011	0.000	0.000	0.000	0.000
B4 - Control	0.023	0.025	0.021	0.023	0.000	0.000	0.000	0.000

Table 2: Weights of bearings before and after testing and amount of grease loss

Bearing	Bearing Weights [lbs]		Grease Loss		
	Pre-Test	Post-Test	[oz]	[g]	% Loss
B1 - Control	79.37	79.35	0.32	9.07	1.5
B2 - R7	81.97	81.73	3.84	108.86	17.5
B3 - L7	82.25	82.19	0.96	27.22	4.4
B4 - Control	79.19	79.19	0.00	0.00	0

Table 3: Test axle press-on and press-off forces for each bearing

Bearing	Press Forces			
	Press On		Press Off	
	[kips]	[kN]	[kips]	[kN]
Cone	--	--	515.6	2,294
B4 - Control	89.5	398	364.9	1,623
B3 - L7	143.6	639	335.7	1,493
B2 - R7	160.0	712	243.4	1,083
B1 - Control	338.3	1,505	205.8	915

In Table 3, the press-on force increases with every additional bearing that is being pressed onto the test axle. Bearing B4 is the first bearing to be pressed on and B1 is the last. The press-off value provided for the cone is the initial force required to break static friction during the press-off procedure where the bearings are removed from the test axle. We utilize a loose cone (inner ring) to start the press-off process since the cone is durable and can sustain large compression forces.

4. Results

This test ran for a total of 93,554 miles (150,561 kilometers). Based on discussions with all the partners involved in this project, the target mileage for these experiments was selected as 100,000 miles (160,934 kilometers). However, this test was terminated before reaching that milestone because the defect that developed on test bearing R7 located in the B2 axle position began to cause the bearing cup (outer ring) to index under full load (34.4 kips per bearing or 153 kN), which created a hazardous operating condition. Hence, for safety reasons, and after consultation with the partners on this project, the test rig was stopped, and a full teardown and inspection of all four bearings involved in this test ensued.

4.1 Conditions at Shutdown

Applied Load: **100% load**, which is equivalent to an AAR Class F load rating of 34.4 kips (153 kN) per bearing. Note that, since the load was applied over the two middle bearings, a total of 68.8 kips (306 kN) was applied resulting in 34.4 kips (153 kN) on each test bearing.

Axle Rotational Speed: **618 RPM**

Equivalent Train Traveling Velocity: **66 mph (106 km/h)**

Average Ambient Temperature: **20.4 °C**

Total Distance Traveled: **93,554 mi (150,561 km)**

Percentage of 100k Mile Target Distance: **93.6%**

Average Air Convection Speed Over Bearings: **14 mph (23 km/h)**

4.2 Data Plots

The vibration and temperature above ambient profiles are provided in Figure 5. Figure 6 gives the absolute temperature, motor power, and axle revolutions per minute (RPM) profiles for this test, while Figure 7 presents the applied load profile for this test.

In Figure 5, the solid blue and red lines in the vibration profile plot represent, respectively, the “Prelim. Threshold” and “Maximum Threshold”. The preliminary threshold is used to differentiate between healthy bearings and possibly defective bearings. In other words, bearings with vibration levels that are lower than the preliminary threshold are healthy (defect-free) bearings, whereas bearings with vibration levels above the preliminary threshold are possibly defective. If a bearing’s vibration levels are above the maximum threshold, then that bearing is defective with a 97% confidence level. For the temperature profile plot, the solid orange line represents the maximum average bearing operating temperature above ambient for healthy bearings at the respective speed and load condition. The solid red line represents the recommended AAR Hot Bearing Detector alarm threshold, which is 170°F (94.4°C) above ambient conditions.

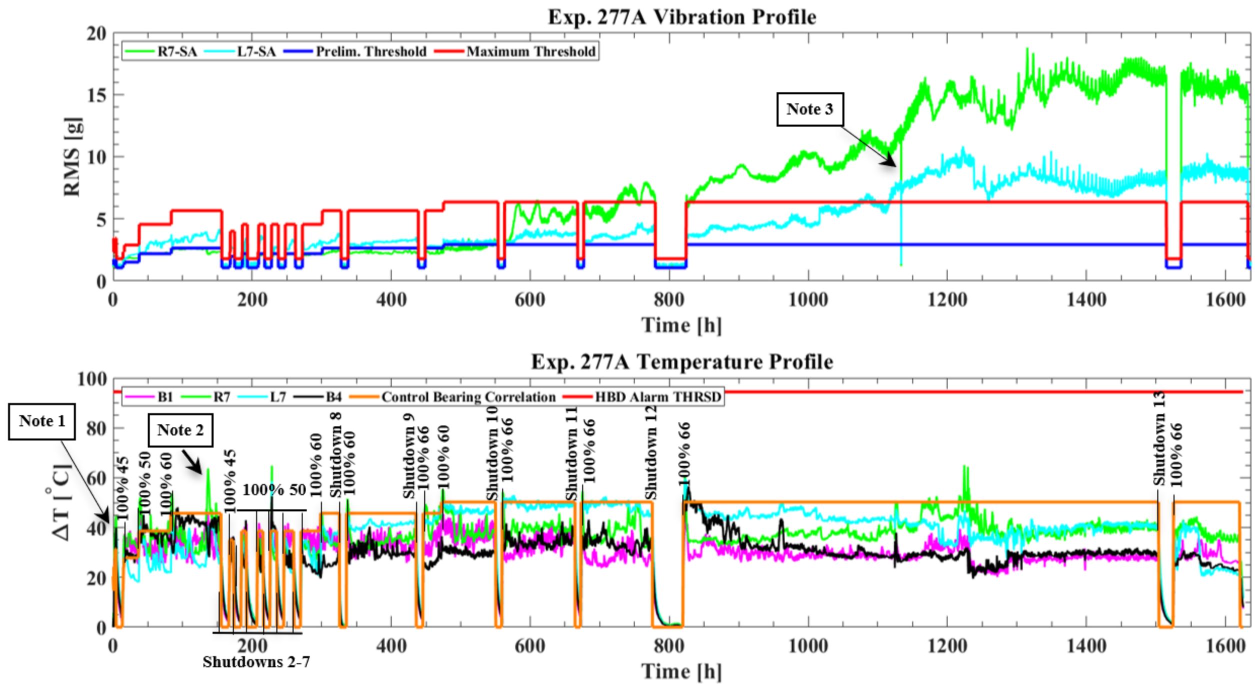

Figure 5: Vibration and temperature above ambient profiles

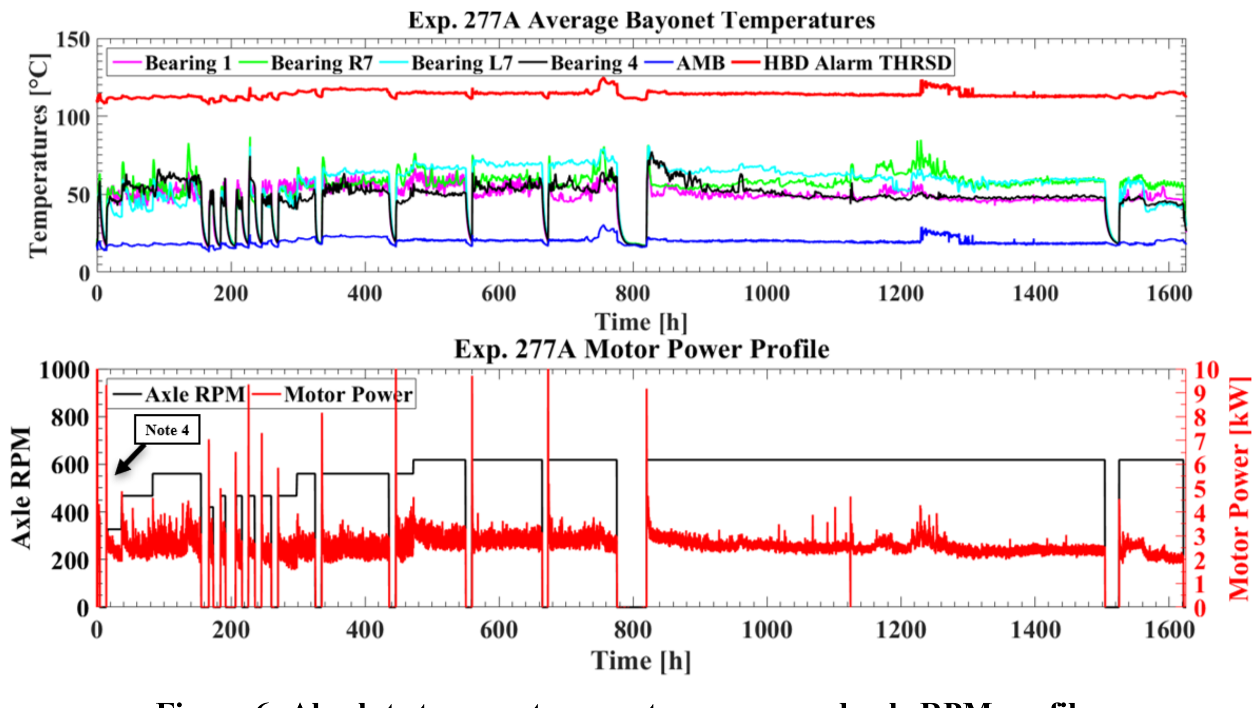

Figure 6: Absolute temperature, motor power, and axle RPM profiles

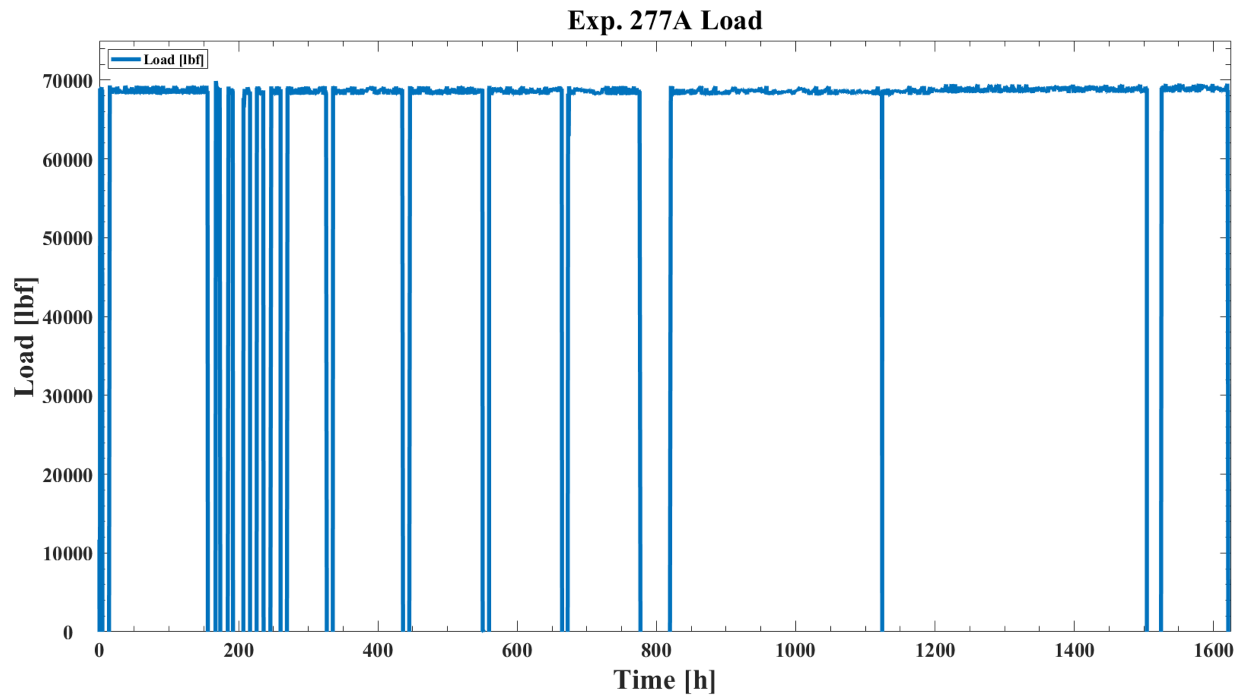

Figure 7: Applied load profile

4.3 Test Notes

Looking at Figure 5, it can be seen that the speed was incrementally increased from 25 mph (40 km/h) to 66 mph (106 km/h) over the course of one month during which the bearings ran for about 21,000 miles (33,800 km). Initially, the operating conditions were set to 25 mph at an unloaded railcar (i.e., 17% of full load or 5.85 kips per bearing) condition as we were unsure what to expect from running bearings that sat idle for three years under varying weather conditions. In fact, it took the test rig motor a few tries to initially get the test axle rotating at an equivalent train speed of 25 mph since the variable frequency drive (VFD) that controls the 30-hp (22 kW) motor kept shutting down because the allowable maximum current draw was exceeded. The spikes in motor power seen on Figure 6 at the beginning of the test occurred due to the large torque required to rotate the bearings when the axle speed was increased (Note 4). Once steady state operation was achieved, the speed and load were set to 40 mph (64 km/h) and 100% load, respectively. Detailed test notes are summarized in Table 4. In the table, the 100% load represents a fully loaded railcar which is equivalent to an applied load of 34.4 kips (153 kN) per bearing for AAR class F and K bearings. The time in hours (h) is the approximate time the event occurred during this test.

Table 4: Timeline of notable events

Notable Events Description	Bearing Location	Mileage [mi]/[km]	Load [%]	Speed [mph]/[km/h]	Time [h]
Sparks emitting from bearing on March 25, 2024, as seen in Figure 8. Tester was shut down in the evening for a cooldown period to mitigate an overnight catastrophic failure. [Note 1] on Figure 5.	Inboard Seal of R7	163 / 262	100	40 / 64	4.0
Bearing seal dislodged (see Figure 9) on March 31, 2024, which was accompanied by an abrupt rise in bearing operating temperature. For safety reasons, the tester was shut down every night from March 31 to April 10, 2024, to mitigate failure. Since no abnormal operation was observed during that period, the tester was allowed to run uninterrupted after that. [Note 2] on Figure 5.	Inboard Seal of R7	6,731 / 10,832	100	55 / 89	136.6
Bearing released grease after seal dislodged. Samples R7-10 (grease from the tester surface) and R7-11 (grease from the seal location) were collected for further analysis. Axle speed was reduced for a short period.	Inboard Seal of R7	7,986 / 12,852	100	45 / 72	173.0
Bearings released grease for a second time (see Figure 10). Samples R7-12 (grease from the tester surface on the inboard side of R7 bearing), R7-13 (grease from the plate in between R7 and L7), and L7-12 (grease from the plate on outboard side of L7) were collected for later analysis.	R7 and L7	18,997 / 30,573	100	60 / 97	436.2
Sudden spike in the vibration levels read by the accelerometer	R7	27,620 / 44,450	100	66 / 97	570.0
Bearing started emitting a loud, persistent, growling noise.	R7	32,913 / 52,968	100	66 / 97	664.4
Vibration levels exceeded the maximum RMS threshold for	R7	36,213 / 58,279	100	66 / 97	724.0

healthy bearings indicating the presence of a bearing defect.					
Tester was shut down from May 10-20, 2024, as the entire team attended the JRC Conference.	N/A	39,677 / 63,854	0	0	779.0
Level 2 vibration analysis was performed [1]. A cone defect was predicted on R7 with more than 97% certainty (See Table 5).	R7	44,603 / 71,782	100	66 / 97	894.4
The experiment was briefly shut down to install an updated safety cage around the pulley and flywheel of the test rig. This occurred over a 20-minute period, and there was a negligible drop in bearing operating temperatures since the cooling fans were turned off during this process. Standard shutdown and restart procedure was followed. [Note 3] on Figure 5.	N/A	61,885 / 99,594	0	0	1,132.5
On June 7-10, 2024, chunks of grease were released from the inboard side of R7 bearing. The accumulated grease was collected (Sample R7-14) for later analysis.	R7	72,143 / 116,103	100	66 / 97	1292.2
Metal shavings were found under the R7 bearing.	R7	76,356 / 122,883	100	66 / 97	1,339.7
The tester was shut down for one day as a precaution due to a hurricane on June 19, 2024.	N/A	87,156 / 140,264	0	0	1,514.5
Final tester shutdown on June 24, 2024, due to R7 bearing indexing under full load causing unsafe operating conditions that necessitated the test to be halted.	N/A	93,554 / 150,561	0	0	1,636.8

Table 5: Level 2 Vibration Analysis [1]

Level 2 Analysis					
R7 Bearing	Folder	71	72	73	74
	Speed [RPM] / [MPH]	618 / 66	618 / 66	618 / 66	618 / 66
	Certainty [%]	**97.71**	**97.90**	**97.30**	**99.40**
	Defective Component	Cone	Cone	Cone	Cone

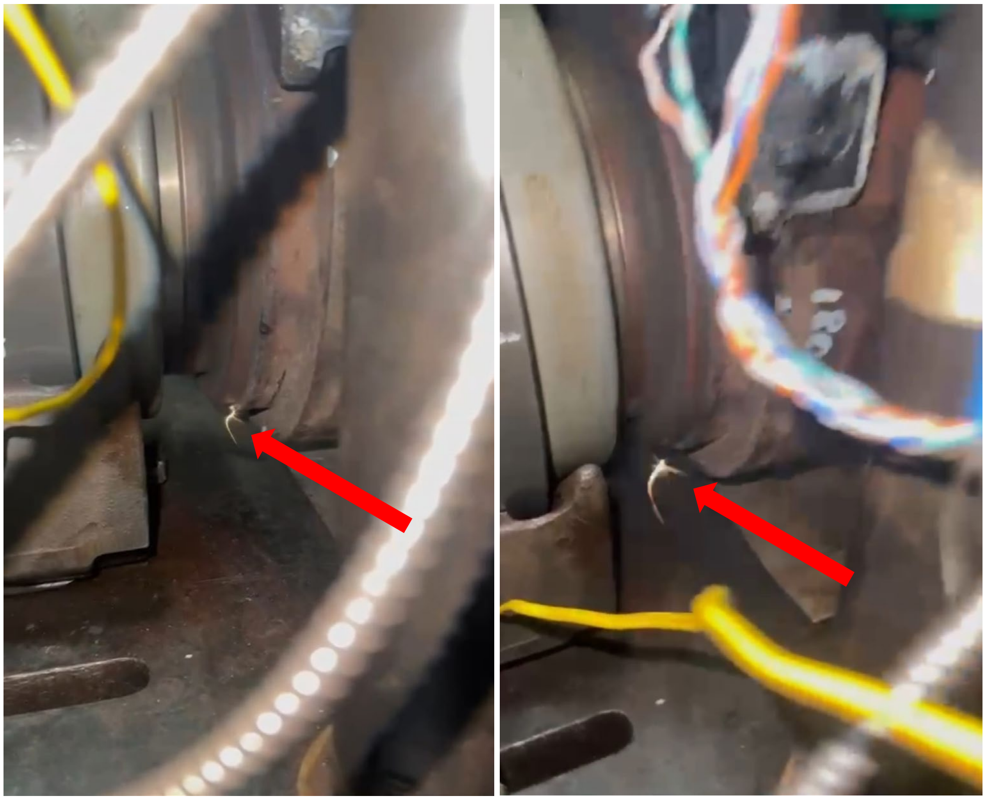

Figure 8: Pictures showing sparks emitting from the inboard seal location of R7 bearing

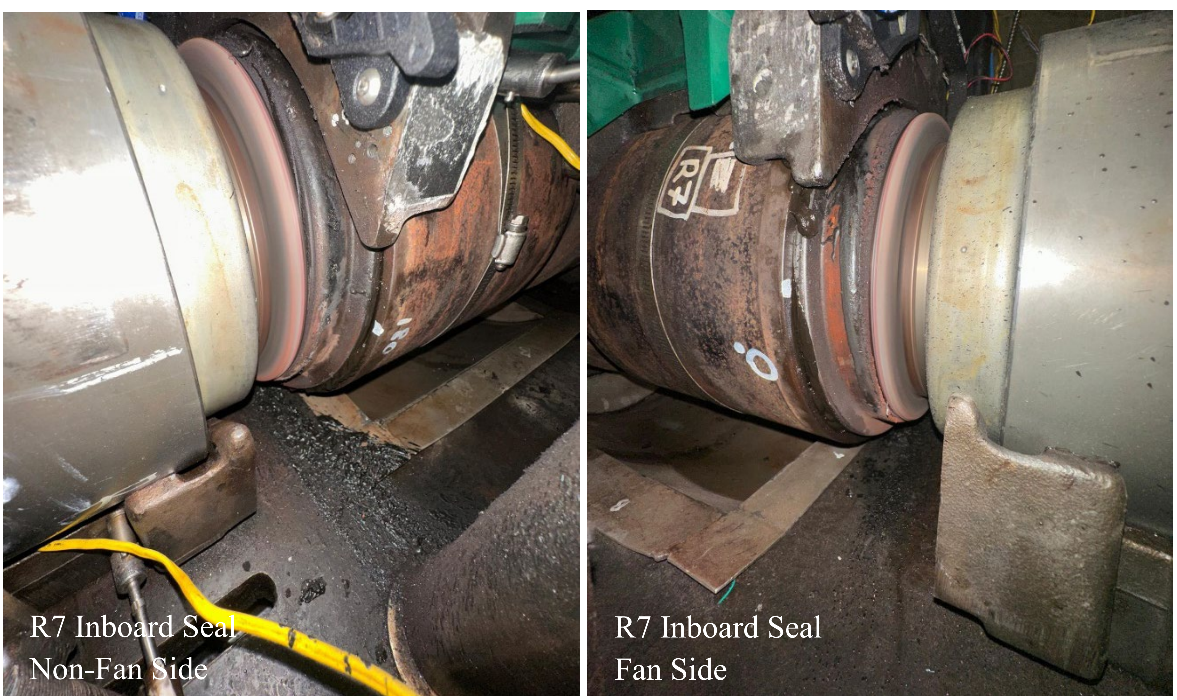

Figure 9: Picture of the inboard seal of bearing R7 that dislodged during testing. The face of the seal is no longer flush with the base of the seal that sits in the bearing cup

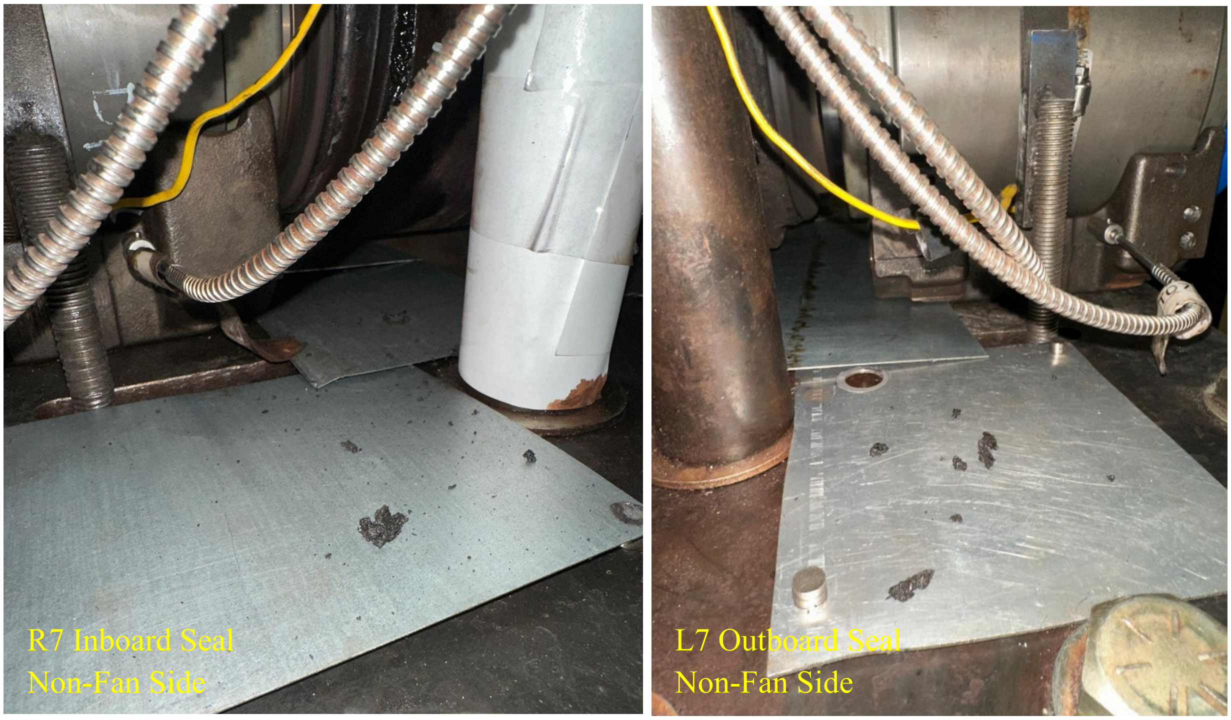

Figure 10: Pictures of the grease that was released from the R7 and L7 bearings around 19,000 miles from the start of the test

4.4 Post-Test Teardown and Inspection Pictures

After the test was terminated, all four test bearings were pressed off the test axle, disassembled, and inspected visually. Pictures were taken before and after the bearings were cleaned in the parts washer for careful documentation of the conditions of the test bearings. Select pictures from the visual inspection performed are presented hereafter.

4.4.1 R7 Bearing – Inboard Cone

Figure 11: R7 inboard cone (inner ring) raceway – Picture 1

Figure 12: R7 inboard cone (inner ring) raceway – Picture 2

Figure 13: R7 inboard cone (inner ring) raceway – Picture 3

Figure 14: R7 inboard cone (inner ring) raceway – Picture 4

Figure 15: R7 inboard cone (inner ring) raceway with crack spanning its entire width

Figure 16: R7 inboard cone (inner ring) raceway showing crack along the width

4.4.2 R7 Bearing – Outboard Cone

Figure 17: R7 outboard cone (inner ring) raceway – Picture 1

Figure 18: R7 outboard cone (inner ring) raceway – Picture 2

4.4.3 R7 Bearing – Cup

Figure 19: R7 cup (outer ring) outboard raceway damage on loaded side

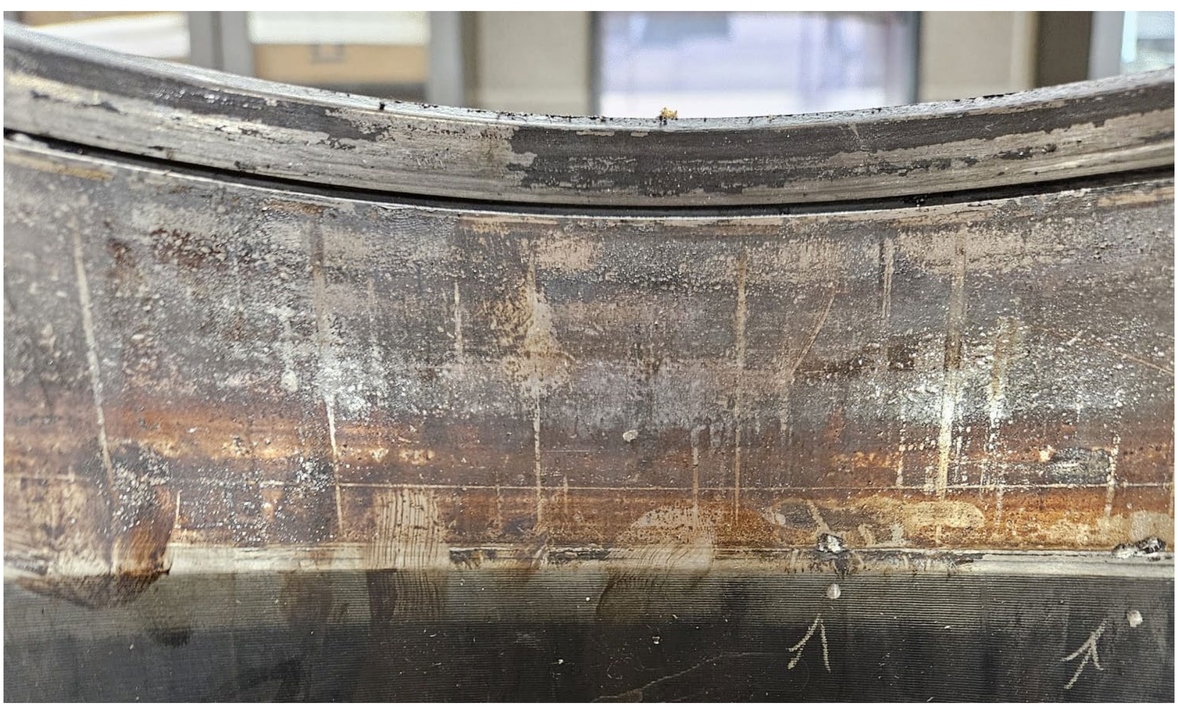

Figure 20: R7 cup (outer ring) inboard raceway on loaded side showing repaired spalls

4.4.4 R7 Bearing – Rollers

Figure 21: R7 inboard cone rollers (left) and outboard cone rollers (right)

4.4.5 L7 Bearing – Inboard Cone

Figure 22: L7 inboard cone raceway

Figure 23: L7 inboard cone raceway inner diameter surface which is pressed on test axle

4.4.6 L7 Bearing – outboard Cone

Figure 24: L7 outboard cone raceway

Figure 25: L7 outboard cone raceway inner diameter surface which is pressed on test axle

4.4.7 L7 Bearing – Cup

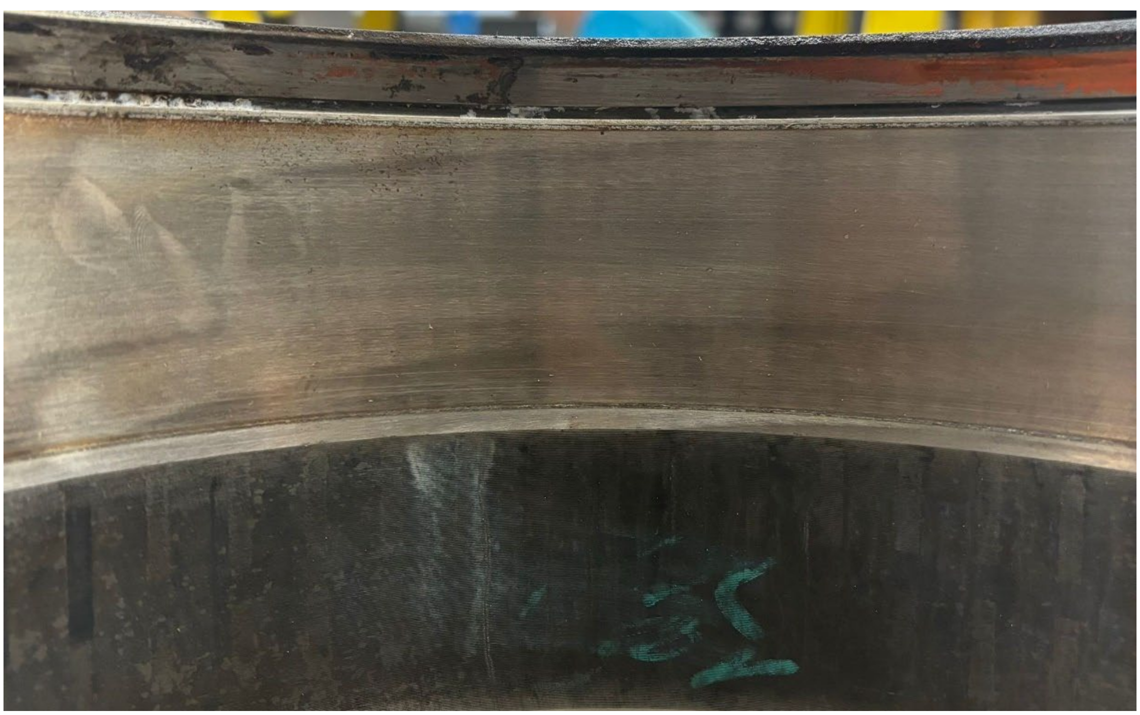

Figure 26: L7 cup (outer ring) inboard raceway on loaded side

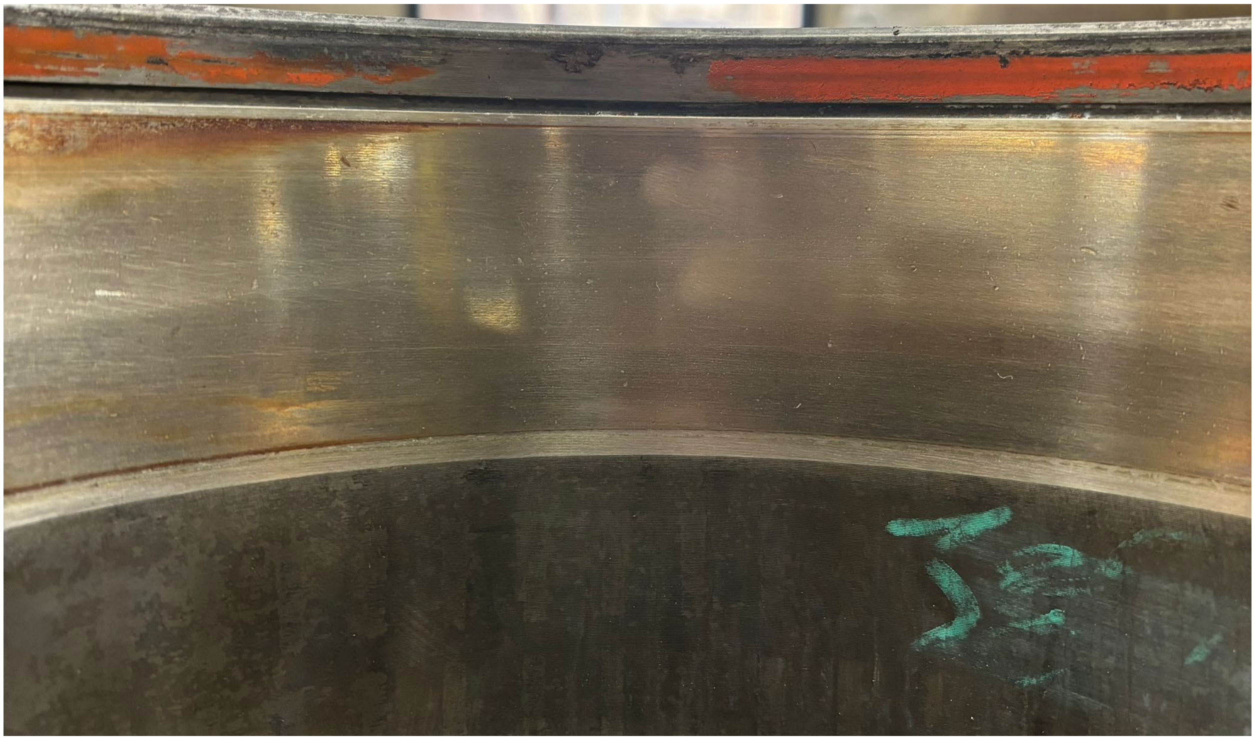

Figure 27: L7 cup (outer ring) outboard raceway on loaded side

4.4.8 L7 Bearing – Rollers

Figure 28: L7 inboard cone rollers (left) and outboard cone rollers (right)

4.4.9 Test Axle Post Press-Off

Figure 29: Test axle post press-off showing significant heat tinting in the location where the R7 inboard cone assembly was positioned

4.5 Spall Analysis

The visual inspection revealed that only components in the R7 bearing had critical damage on both cone raceways and the cup outboard raceway, with the inboard cone showing a deep crack along the entire width circumference. Careful documentation of the damage observed on the different components of the R7 bearing was done by performing detailed spall area measurements. The latter was accomplished by creating an imprint of each spall using molten bismuth, and then utilizing image processing software to accurately quantify the spall area. More information about this process can be found elsewhere [2]. The following pictures present the different bismuth imprints created to quantify the spall (damage) area. In the case of the large spall that developed on the inboard cone (inner ring) of R7 bearing, three imprints were created for better accuracy. Table 6 summarizes the spall area measurements taken and their percentage of the total raceway surface area for an AAR Class F bearing.

Figure 30: One of three parts of the R7 inboard cone spall which had a total area of 13.6 in^2 (refer to Figure 11)

Figure 31: R7 inboard cone spall of Figure 13 with a measured area of 1.82 in^2

Figure 32: R7 inboard cone spall of Figure 12 with an area of 0.45 in^2 (left) and R7 inboard cone spall of Figure 14 with an area of 0.61 in^2 (right)

Figure 33: R7 outboard cone spall of Figure 17 with a measured area of 2.47 in^2 (left) and R7 outboard cone spall of Figure 18 with a measured area of 1.43 in^2 (right)

Figure 34: R7 cup outboard raceway spall of Figure 19 with a measured area of 7.88 in^2

Table 6: R7 bearing spall (damage) area measurements

Damaged Area	Spall Size [in^2]	Total Raceway Area [in^2]	Percentage of Area Spalled [%]
R7 inboard cone – Figure 30	13.60	43.26	31.4
R7 inboard cone – Figure 31	1.82	43.26	4.2
R7 inboard cone – Figure 32 (left)	0.45	43.26	1.1
R7 inboard cone – Figure 32 (right)	0.61	43.26	1.4
R7 inboard cone – Total	16.48	43.26	38.1
R7 outboard cone – Figure 33 (left)	2.47	43.26	5.7
R7 outboard cone – Figure 33 (right)	1.43	43.26	3.3
R7 outboard cone – Total	3.90	43.26	9.0
R7 cup outboard raceway – Figure 34	7.88	56.93	13.8
R7 bearing – Total	28.26	200.38	14.1

Not surprisingly, the inboard side of R7 bearing, which is the side with the dislodged seal, exhibited the most damage, with 38.1% of the total cone raceway area being damaged. Interestingly, the cup inboard raceway showed no signs of notable damage other than some small pits and accelerated surface wear. However, the outboard raceway of the cup sustained damage on the loaded portion covering 13.8% of the total cup outboard raceway. In total, the damaged area within the R7 bearing was 28.26 in^2, which represents 14.1% of the 200.38 in^2 of total surface area for all four raceways in an AAR Class F bearing (two cone raceways and two cup raceways).

5. Test Observations

Throughout the course of this experiment, the bearings were monitored regularly, and any changes or unusual operating conditions or events were carefully documented. The bearings were not running unsupervised (overnight or over the weekends) if they were deemed to be at risk of imminent catastrophic failure.

As the test progressed, the noise levels began increasing, peaking at 123 decibels around the compromised seal area of R7 bearing. The vibration levels within the R7 bearing also reached a peak RMS g-value of 18.78. After the seal dislodged on the inboard side of R7 bearing, grease began leaking out slowly over weeks. Aluminum plates and tarps were installed to catch the released grease and make it easier for collection and analysis.

Examining the vibration profiles of Figure 5, it is apparent that the R7 bearing developed a defect around 724 hours into the test, as exhibited by the vibration levels exceeding the maximum threshold represented by the solid red line (i.e., level 1 vibration analysis described in reference [1]). In fact, the vibration profile of the R7 bearing suggests that the defect developed earlier, a little after 570 hours into the test, but the vibration levels did not exceed the maximum threshold until 724 hours into the test. As explained earlier, vibration levels that lie between the solid blue and red lines suggest that the bearing is possibly defective [1]. Interestingly, while the vibration levels clearly suggest that the R7 bearing developed a significant defect, the temperature profile was below the average operating temperature for healthy bearings running at the same speed and load conditions, and well below the HBD alarm threshold. Considering the significant damage observed on the R7 bearing components depicted in Figure 11-21 and summarized in Table 6, it is concerning that the bearing operating temperature did not exhibit any signs of distress.

A level 2 vibration analysis was also conducted, which parses through the vibration data for each bearing utilizing power spectral density (PSD) plots and determines the probability of the defect being present at a specific bearing component (cup, cone, or roller) [1]. This analysis indicated that there was a cone spall in the R7 bearing with more than 97% certainty, and a low certainty of any spalling in the L7 bearing. Upon teardown and inspection, this analysis was confirmed, as there was major damage all along the inboard cone of the R7 bearing as well as damage on the outboard cone and the outboard raceway of the cup on the loaded region; however, there was no notable damage on the L7 bearing. Moreover, significant heat tinting was also observed on the test axle in the R7 bearing inboard location (Figure 29) as well as noticeable pitting on the rollers of the R7 bearing, however, the rollers for the L7 bearing appeared normal, other than the expected wear and tear. The test was terminated when the R7 bearing cup (outer ring) began indexing on the axle under full load while running at a simulated train speed of 66 mph (106 km/h). The bearings had traveled a total of 93,554 miles (150,561 kilometers) at that point.

6. Conclusions and Future Work

This study is the first of its kind where the performance of railroad bearings with long periods of inactivity is being assessed. The study was motivated by research need statements from the railroad industry and the NTSB following the East Palestine, OH train derailment in which the bearing that catastrophically failed had two previous long periods of inactivity where the bearing sat idle for 565 days and then again for 216 days. It is important to note that this study is still in its early stages and the results presented in this report are for the initial two bearings that have been tested to this point. The authors caution against generalizing these results as the sample of bearings tested is not significant to formulate concrete conclusions.

The results thus far indicate that out of the two inactive bearings tested (R7 and L7), one of them had the seal dislodge after running 6,731 miles, lost 17.5% of its grease, developed a spall around 28,000 miles of operation, and caused the test to be terminated shortly before reaching 94,000 miles, while the other bearing ran with no notable incident other than losing 4.4% of its grease. Interestingly, other than a couple of short-lived temperature spikes, the bearing operating temperatures never exceeded thresholds for healthy bearings running at the same speed and load conditions and were well below the AAR recommended HBD alarm threshold. The latter implies that current HBD technologies deployed in the U.S. rail network would not have caught the significant damage observed in R7 bearing. However, since R7 bearing was emitting a loud noise peaking at 123 decibels, it is possible that an acoustic bearing detector (ABD) could have potentially identified this bearing as being defective or problematic.

As stated earlier, it is too soon to draw general conclusions about the performance of the inactive bearings as two other bearings from the six originally sent for testing at UTRGV just completed a 100,000 miles (161,000 km) performance test on the four-bearing tester with no vibration or temperature issues being reported. It is important to note that this study is in its initial phase, and UTRGV and its partners are working on acquiring more inactive bearings for testing to expand the scope and sample size of this study to generate statistically meaningful results.

Moreover, a comprehensive evaluation of the grease characteristics and remaining service life pre and post testing is being carried out and will be reported separately.

7. References

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