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President’s Podium

Direct Observation of Replenishment of Lubricants at Contact Inlet in Ball Bearings
M. Hokao and K. Sonoda, Core Technology R & D Center, NSK Ltd., Japan
J. Sugimura, International Institute for Carbon-Neutral Energy Research, Kyushu University, Japan

Bearing Corrosion Analysis Using Machine Vision and Computational Algorithms
Jason Galary, Director of Research and Development, Nye Lubricants, Inc., USA

In Memoriam

2018 Calendar of Events

Advertiser’s Index
2018 has been another wonderful year for NLGI. Our membership has grown and we continue to see attention from other industry companies interested in joining our organization. Thanks to each of you for helping us to maintain the high standards that our members have come to expect. We believe that this organization offers a high level of value and is an outstanding experience for both current and potential members. At least, that’s what WE believe. What we really would like to know is what YOU think.

Over the next few months, you may receive a personal phone call from one of our board members asking you some questions about your membership including:

- Verifying membership type and company contact information
- What other employees at your company should receive NLGI e-mails about our annual meeting, issues of The Spokesman, etc.?
- What do you enjoy most from your NLGI membership?
- Is your company interested in getting more involved by participating on NLGI committees?
- If there’s one area of improvement you’d like NLGI to focus on, what would it be and why?

The reason for these calls is simple. We want to give back to YOU – the member. Obtaining feedback from these calls will allow the board to plan for the future strategically while offering the best benefits and services possible for our members. For us to add value to the organizations, we need to know what you value.

Please take a few minutes and share candid feedback with the designated board member and thank you in advance for your participation.

If you don’t receive a phone call but would still like to share your thoughts and ideas about NLGI, please contact myself, Crystal or any of the Board members. We really do want to hear from you!

You might remember from my Podium last month that “Enhancing Membership Growth and Outreach” is one of the goals I’ve outlined during my Presidency. Establishing more communication with our members and understanding what is valuable to them is one step toward achieving that goal. We have a great team of leaders and members and, together, I’m confident we will achieve this goal and continue to move to achieve our vision to be recognized as the premier source of support to key stakeholders of the worldwide lubricating grease industry.

I look forward to learning what’s valuable to each of you.

Joe Kaperick
Afton Chemical
NLGI President 2018-2020
Application: Industrial Gear Lubricant Formulation

Recommendation: IRGALUBE® 353

Monson, an Azelis company, is the best partner to solve your formulation needs. We recommend IRGALUBE® 353, an easy to handle liquid EP/antiwear additive providing excellent gear protection. Acts as a FZG “enhancer” with commercial gear oils. Good compatibility with water, Ca detergents and ZnDTPs. No antagonism towards iron and yellow metal protection. Leverage our full range of technical resources and quality products by contacting customer sales and service at 1-800-235-0957, or via email to csr@monsonco.com for your local sales representative.
Direct Observation of Replenishment of Lubricants at Contact Inlet in Ball Bearings

M. Hokao and K. Sonoda
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Japan

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Japan

Introduction
Lubricating grease has characteristic rheological properties such as yield stress, shear thinning, and time dependence [1,2], and it is believed that these properties depend on the structure formed by thickener dispersed in the grease [3-6]. Since the rheological characteristics prevent its leakage from a bearing, grease is suitable for reducing or omitting maintenance of rolling bearings. Owing to this practical advantage, grease lubrication is widely applied in rolling bearings. However, from the viewpoint of bearing performance, the rheological characteristics of grease may also cause some disadvantages for lubricant replenishment at contact inlets [1] and agitating resistance in rolling. Therefore, it is important to understand the influence of grease rheology on actual bearing lubrication.

The lubricating condition in EHL contacts is actively investigated using optical evaluation with ball on disk tests. It is shown that the lubricating condition is greatly affected by replenishment of lubricants at contact inlets, since the cavitation caused by the reduction of pressure at the outlet of a contact area makes a drastic decrease in lubricant volume at contacting surfaces. Poor replenishment induces starved lubrication and causes a breakdown of the oil film. Owing to its poor fluidity, grease lubrication tends to undergo starvation more often than oil lubrication. Oil bleeding, structural breakdown of grease, capillary forces [7, 8, 10], surface tension, van der Waals attractive forces [9], centrifugal forces [1], ball spin, and bearing cage gap [10-12] are reported as factors that affect replenishment.

In addition to these factors, replenishment in a rolling bearing may be affected by the bearing geometry such as constant passage of rolling elements and curved surfaces of rings. An experimental study using a sapphire bearing shows that starved lubrication occurs due to the increase in bearing speed and oil viscosity in oil lubrication [13]. However, there are few studies in which the replenishment at a contact inlet is observed in a rolling bearing with grease lubrication.

Grease within the bearing cavity and the contact area cause the generation of resistant forces against bearing rotation. The resistant forces in a bearing can be divided into three forces, namely the rolling friction between rolling elements and ring raceways, the sliding friction, and the viscous drag force imposed by the lubricant within the bearing cavity [14]. The rolling friction includes both elastic hysteresis of bearing steel and viscous rolling resistance of lubricants at the EHL contacts. Starvation causes the increase of both of the rolling friction and the sliding friction, and excess lubricant increases the drag force.

The distribution of grease changes greatly during the initial running stage. The grease quickly flows to the bearing shoulders, seals/shields, and the cage [15] from the original position. The grease distribution is affected by both the running conditions, including bearing type, rotating speed and temperature, and grease rheological properties. It was reported that the distribution of grease in a bearing was affected...
by the yield stress of grease. Greases with lower yield stress showed better grease circulation in a bearing and tended to keep their torque levels during running tests. On the contrary, greases with higher yield stress showed larger reduction of starting torque after running tests [16]. Although grease distribution seems to influence bearing torque, the details of the relationship between the grease distribution and the bearing torque is not understood sufficiently.

This study investigated the relationship between replenishment of lubricant at the contact inlet of a ball bearing and bearing torque by using four types of lithium soap grease with different rheological properties at low speed conditions. Replenishment at the inlet of the EHL conjunction was evaluated by the inlet distance of lubricant, which was determined by direct optical observation around the contact region using a bearing with a transparent outer ring made of glass. Bearing torque was measured simultaneously while direct observations were made. As a comparison with grease lubrication, base oil was also studied. In order to investigate the influence of the inlet distance on the bearing torque, small quantities of lubricant were put in a bearing to minimize the influence of the viscous drag force. Rheological properties were evaluated with the yield stress and the hysteresis loops [2].

**Experimental**

**Measurement of the inlet distance and the bearing torque**

Figure 1 shows the bearing test apparatus. To observe a contact area, BK7 optical glass was employed as the outer ring of a deep groove ball bearing 6306 (inner diameter: 30 mm, outer diameter: 72 mm, width: 19 mm) with a plastic cage. The diametrical clearance between a ball and a pocket of the cage was 0.34 mm. The bearing had 16 EHL contacts and 8 balls. The raceway of outer ring was coated with semi-reflective chromium film for the observation of interference images at the contact ellipse. Seals were applied to the bearing.

The inner ring was rotated and radial load was applied through the outer ring. Interference images at the constant area of the outer ring were taken by a high-speed camera at a frame rate of 8,000 fps. The rotating speed, the radial load, and temperature in the tests were 100 min⁻¹, 29.4 N, and 25 °C, respectively. 85 frames of a contact area were taken during the passage of a ball at this rotating speed. One of the 85 frames was selected for the observation, since the inlet distance did not substantially change during the passage of a ball.

The amount of lubricant in a bearing was 0.5 g, which corresponded to 3.5% of the volume of space of a 6306 bearing, in order to reduce the influence of the viscous drag force on bearing torque.

The rotating tests were started after running-in for one min. The high-speed camera was mounted above the top of the outer ring, and images around the contact region were taken five times (at 3, 900, 1800, 3600, and 7200 sec after the start of rotation).

The inlet distance was measured from the image of the contact area and was defined as the distance from the edge of the inlet side of the contact ellipse to the interface of the lubricant and air. In this study, the inlet distance at the top of the outer ring was measured, since the radial load on the bearing was the highest at this position, and there was less influence of gravity on lubricant replenishment at this position. A pretest with one of the greases showed that for seven successive ball contacts, measurements of the inlet distance were scattered from 390 to 530 mm after 300 sec. Therefore, bearing tests were conducted two or three times for each of the lubricant samples, and the averages of the inlet distance were obtained.

Bearing torque was measured using a torque meter connected to a rotating axis. This configuration enables the simultaneous observation of the interference image and the bearing torque.

**Test greases**

Table 1 shows the properties of test greases. Two types of lithium soap, i.e. lithium 12-hydroxystearate and lithium stearate, were
used for the thickeners. All the greases contain ester oil as the base oil with no additives. The greases were prepared by rapidly cooling the mixture of base oil and dissolved thickener powder, and then processing with a three roll mill. The concentration of the thickener was selected to obtain greases with two different ranges of penetration. The yield stress and the hysteresis loops were measured in rheology tests. In addition to the greases, the base oil was also tested.

Measurements of rheological properties

The yield stress and the hysteresis loops were measured at 30 °C with a rheometer. To determine the yield stress, storage modulus G' and loss modulus G'' were measured while the strain was increased from 0.01 to 1,000% in oscillating tests at 10 Hz. A parallel plate was used for oscillating tests, since sample fracture occurred [1, 2], i.e., cracks formed in the gap, when a cone was used under these conditions. The yield stress was defined as the shear stress when loss tangent G'' / G' became unity [3, 16].

The hysteresis loops were obtained by continuously scanning, first ascending and then descending, the shear rate between 0 and 1,000 s\(^{-1}\). The duration of each scan (ascending or descending) was 100 sec. A cone with an angle of 0.5° was used for the hysteresis loop tests. Before the ascending scan, the sample was sheared at 10 s\(^{-1}\) for 5 min in order to reduce the influence of the shearing history on grease [2]. The maximum shear rate of 1,000 s\(^{-1}\) was intended to simulate the maximum shear rate at the inlet conjunction in practical bearings in a measurable range, although that is assumed that actual maximum shear rates are higher than this by two orders of magnitude or more [1, 16]. Sample fracture was not observed after hysteresis loop measurements.

A flow curve was plotted as shear stress against shear rate. The hysteresis loop area was defined as the area between the curves. Figure 2 shows the result of the loop measurement of Grease A. The flow curves showed shear thinning and time-dependency, in other words, thixotropy. The thixotropic property accompanies a structural change of thickeners brought about by shear. The gap between the ascending curve and the descending curve is large for thixotropic grease, and the hysteresis loop area corresponds to the extent of the structural changes that occur in a shear rate range in a practical bearing. In addition, the shear stresses at 0 s\(^{-1}\) in Figure 2 indicate the decrease in the yield stress after the measurement.

The yield stress and the hysteresis loop area of the greases are shown in Table 1. Lithium 12-hydroxystearate grease had higher yield stress than lithium stearate grease with the same penetration level. The hysteresis loop area tended to correspond to worked penetration and had no correlation with the yield stress.

AFM measurements

Thickeners on untreated surfaces of the greases were observed using an atomic force microscope (AFM) in non-contact mode in ambient air [4-6]. Sample grease was coated on a flat glass plate, 1 mm thickness, at the shear rate of 5 s\(^{-1}\), and then a cantilever made of Si\(_3\)N\(_4\) was oscillated with a scan speed of 0.5 Hz. Viscoelastic character of the surface affects the response of the oscillation of the cantilever.

Figure 3 shows AFM images of test greases obtained by analyzing this phase difference [6]. Images were taken on two scales, 10 mm and 3 mm square. Dark parts represent thickeners. This method does not include a drying pretreatment of the grease, and can show more realistic, dispersed state of thickeners in the grease than the conventional methods in which a dried thickener image is obtained with an electron microscope.

The 10 mm square image is adequate to observe the thickener structure in grease. From the 10 mm square images of the softer (NLGI grade 2), high penetration Grease B and Grease D, it can be seen that lithium stearate thickener in Grease D tends to disperse more evenly than lithium 12-hydroxystearate thickener in Grease B.

Comparing the same thickener type grease, harder (NLGI grade 3), low penetration grease showed denser thickeners than softer NLGI
grade 2 greases. The 3 mm square images show the details of the thickener shape. The lithium stearate thickeners in Greases C and D are larger and have a more rod-like shape than those in the lithium 12-hydroxystearate greases. Oil separation from the greases was not observed visibly in ambient air, though the thickeners showed uneven dispersion.

**Results**

**Observations of contact inlet**

Figure 4 shows the images of the contact region after 3 and 7,200 sec. Bearing balls have moved from left to right in the images, and the inlet conjunction is at the right edge of the contact ellipse. A horseshoe-shaped film is shown at the contact ellipse at this test condition. Interference images of the grease are similar during the test period. The images show part of the contact ellipse and the interface between lubricant and air at the outer ring surface. The upper and lower parts of the image are obscured because light intensity is insufficient due to the curved surface of the outer ring groove. Interference images suggested the difference of oil film thickness of greases, though the analysis of oil film thickness was not attempted in this study.

All test lubricants formed a meniscus at the contact inlet, and replenishment of the lubricants was observed. The inlet distance became shorter with rotating time, and the rate of the decrease depended on the test lubricant.

Figure 5 shows changes in the inlet distance with time. All the lubricants showed an initial drop of the inlet distance, and after this drop, it was almost level. The initial drop was larger for the base oil and Grease A than the other greases. The decrease in the inlet distance of Grease B was the smallest among all the tested greases. The greases kept larger inlet distances than the base oil.

Tested bearings were observed visually. Little grease was attached to the seals, and most of the grease was seen on the side of the raceway of rings and on part of the cage. In tests of the base oil, very little oil was seen on the side of the raceway.

**Bearing torque**

Figure 6 shows the relationship between the inlet distance and the bearing torque. As mentioned above, the inlet distance and bearing torque were measured simultaneously. The plots in Figure 6 are the data taken in all the tests. The figure shows that the bearing torque increased with the inlet distance. Grease A showed an extensive range of both the inlet distance and the bearing torque. The base oil showed short distance and low torque.

**Discussion**

The influence of grease type on replenishment

The inlet distance represents replenishment at the contact inlet, and a large inlet distance means that a large amount of lubricant is supplied to the inlet conjunction. Figure 5 demonstrates that the inlet distance of the base oil decreased in the initial period and reached the minimum value among these test lubricants after 7,200 sec. This implies that lubricating oil is not necessarily suitable for supplying more lubricant than these greases, at least during this test period with the lubricant amount used in the present tests.

Due to its excellent fluidity, oil may easily drop and move away from the contact surface, and the reduction of oil volume around the contact area may cause the decrease in the inlet distance. On the contrary, grease attached around the contact area seems to contribute to keep replenishment at this low speed condition. The harder grease with lower penetration showed large initial decrease of replenishment and shorter inlet distance than the softer, high penetration grease with the same type of thickener. The lithium 12-hydroxystearate grease showed a tendency for large decrease of the inlet distance as compared with the lithium stearate grease with the same penetration level.

The AFM images shown in Figure 3 indicated a difference in the thickener structure, and this structural difference may affect the rheological characteristics shown in Table 1. The lithium stearate greases with rod-like thickener, Grease C and Grease D, tended to have lower yield stress than Grease A and Grease B with lithium 12-hydroxystearate thickener.
As for the hysteresis loop area, both the grease with dense thickeners and the grease with even dispersion of thickener tended to show larger loop area. This suggests that the yield stress is affected more by thickener shape than distribution, and the thixotropic property is influenced by both the density and the distribution of thickeners.

The thickener structure has several features such as thickener shape, size, volume, and distribution [6]. The yield stress reflects the structural change in the transition from the elastic state to the viscous state, and the thixotropic property is affected by the change after the transition. Therefore, influential features for the yield stress could be different from those for the thixotropic property. The shape of thickeners may be an influential factor especially for the yield stress, and thickener with a rod-like shape may move due to weaker shear force. On the contrary, the thickener volume and distribution seem to have greater influence on the thickener structure and may increase the thixotropic property.

Figure 7 and Figure 8 show the relationship between the rheological properties and the inlet distance after 2 h of rotation. The inlet distance showed a better correlation with the hysteresis loop area than with the yield stress, and the inlet distance was larger for grease with smaller hysteresis loop area. This implies that the thixotropic property prevents replenishment. The significant change of flow properties of the thixotropic grease may cause channeling [1] at the sides of the rolling track.

The bearing tests show that oil lubrication does not necessarily lead to better replenishment in a bearing, and grease was attached to the side of the raceway and the cage in the grease lubrication. This implies that lubricant replenishment in grease lubrication is driven by both local capillary action [7, 8, 10] and some other external force on the grease around the contact area [10].

To replenish continuously by the capillary force generated between a ball and a ring, the fluidity of the lubricant needs to be low enough to flow while balls pass through it. Although it seems possible that shear degraded grease or its base oil is forced to reflow by the capillary force, this effect was not clear in these tests. One possible mechanism of an external force is that applied by the cage [10-12]. It is likely that the grease on the sides of the contact area attaches to a ball, and the cage forces the adhered grease to flow to the sliding surface of the ball. This may allow grease to reflow to the contact area as long as sufficient volume of grease is attached on a ball. The thixotropic grease does not seem to be suitable for supplying grease attached on a ball due to the generation of channeling in the early stage of rotating. In addition, it seems that thixotropic grease is scraped easily by the cage. Centrifugal force [1] is also possible as the external force, although this test was conducted with the inner ring rotation at low speed.

The influence of the inlet distance on bearing torque.
The factors that affect bearing torque generation can be divided into the rolling friction, the sliding friction, and the viscous drag force. Since viscous dissipation within the lubricant at the inlet conjunction influences the rolling friction [14], the inlet distance affects the bearing torque.

In this study, the viscous drag force was minimized by putting a small quantity of lubricant in a bearing. The bearing observation results suggest that circulation of grease in the bearing did not occur, and the viscous drag force was limited. The sliding friction was indicated to be almost constant in the tests since the direct observations of the contact area showed that all test lubricants were in fully flood condition. Figure 6 showed the correlation between the inlet distance and the bearing torque; the bearing torque increased with the inlet distance. This, in turn, suggests that the inlet distance has a good correlation with the rolling friction. The bearing torque can be reduced by decreasing the inlet distance of lubricants in fully flood conditions.
Conclusions
This study investigated the replenishment of grease at a contact inlet of a ball bearing by direct observation of the inlet distance. In addition, correlations between rheological properties, the inlet distance, and the bearing torque were also investigated with four types of lithium soap grease and an ester base oil at a low speed condition.

The results are summarized as follows.

1. Lubricating oil is not necessarily more suitable than grease for replenishing larger amounts of lubricant at contact inlets.

2. The bearing torque increased with the inlet distance. This suggested that the inlet distance had a good correlation with the rolling friction.

3. The harder NLGI 3 grease with lower penetration showed large initial decrease of replenishment and shorter inlet distance than the softer NLGI 2 higher penetration grease with the same thickener type.

4. The lithium 12-hydroxystearate grease showed a tendency for larger decrease of the inlet distance compared with the lithium stearate grease with the same penetration level.

5. The inlet distance showed a better correlation with the hysteresis loop area than with the yield stress, and the inlet distance was larger for grease with smaller hysteresis loop area.

References
Table 1 Properties of greases

<table>
<thead>
<tr>
<th>Grease</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base oil type</td>
<td></td>
<td></td>
<td>Ester oil</td>
<td></td>
</tr>
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<td>Base oil viscosity at 40 °C, mm²/s</td>
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<td>Lithium stearate</td>
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<td>3</td>
<td>2</td>
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<td>Hysteresis loop area, kPa/s</td>
<td>682</td>
<td>114</td>
<td>776</td>
<td>141</td>
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Figure 1 Bearing test apparatus

Figure 2 The hysteresis loop of Grease A
<table>
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<th>Grease A</th>
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<table>
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<td><img src="image4.png" alt="Image of Grease B" /></td>
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</table>

**Figure 3** AFM images of test greases
Figure 4 The inlet distance of the greases at different times

Figure 5 Changes in the inlet distance with time
Figure 6  Relationship between the inlet distance and bearing torque

Figure 7  Relationship between yield stress and the inlet distance

Figure 8  Relationship between hysteresis loop area and the inlet distance
Bearing Corrosion Analysis Using Machine Vision and Computational Algorithms

Jason Galary
Director of Research and Development
Nye Lubricants, Inc., Fairhaven, MA 02719 USA

Abstract

This paper presents a new corrosion detection system to identify, accurately measure and categorize corrosion on bearings. The new corrosion detection system uses machine vision technology hardware and computer software to identify and classify corrosion and accurately determine the percentage of affected surface area on test specimens. This new methodology reduces human error, improves accuracy and increases repeatability of consecutive measurements relative to the traditional visual evaluation of corrosion.

At present, there are several test methods that use bearings to determine the corrosion prevention properties of lubricating greases. They include the EMCOR method (ASTM D6138) and two other standard corrosion tests (ASTM D1743 and D5969). The downside of these methods is that they rely on subjective visual evaluation of corrosion and an approximate rating system.

This paper also presents results from a study to compare the new corrosion detection system with the subjective visual analysis technique. Lithium greases (NLGI grade 2) based on PAO (polyalphaolefin) base oil were tested according to D6138. Corrosion on bearing races was evaluated with both techniques by means of an experimental design (DOE). Statistical analysis of percent corroded surface area from the new system showed that there was a statistically significant difference between greases from two manufacturers. This important result was not obvious from visual ratings. Actual corrosion percentages provide finer resolution than traditional visual analysis.
1 Introduction

Corrosion is defined as the deterioration and break down of metal due to chemical reactions with moisture, salts and other materials. Left unprotected, machine components can suffer severe damage from corrosion of their surfaces. Mild steel alloys such as 100Cr6 and AISI 52100, used commonly in rolling element bearings, are vulnerable to corrosion. Since bearings are exposed to water in many applications, many specifications require greases to protect steel from corrosion.

1.1 Current Corrosion Tests

Several test methods are used widely for measuring corrosion prevention properties of lubricating greases in rolling element bearings [1-6]. These include ASTM D1743 [3], ASTM D5969 [1,4] and the EMCOR test, ASTM D6138 [2]. A good comparison of the history of these test methods can be found in the paper by Kaperick, Aguilar, and Lennon [6].

ASTM D1743, “Standard Test Method for Determining Corrosion Preventive Properties of Lubricating Greases”, evaluates the ability of grease to protect bearings stored under wet conditions. A tapered roller bearing is immersed in distilled water for 48 hours at 52⁰C. ASTM D5969 is similar to D1743, although bearings are immersed for only 24 hours in various concentrations of salt water. At the end of these two tests, the bearings are disassembled and evaluated on a pass-fail basis (Table 1). For a grease to pass either test, two out of three bearings must receive a pass rating (no corrosion spots greater than 1.0 mm diameter). D1743 and D5969 are static tests because the bearings are motionless during these procedures.

ASTM D6138 is a dynamic bearing corrosion test called “Standard Test Method for Determination of Corrosion-Preventive Properties of Lubricating Greases Under Dynamic Wet Conditions (EMCOR Test)”. In this test, double row self-aligning ball bearings (1306K/236725) are immersed partially in salt water or distilled water while they undergo a sequence of running and stopping during 168 hours. At the end of the test, the bearings are disassembled, and corrosion is evaluated on a scale of 0-5 (Figure 1).

<table>
<thead>
<tr>
<th></th>
<th>ASTM D-1743</th>
<th>ASTM D-5969</th>
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<tr>
<td>Pass</td>
<td>No Spots &gt; 1.0mm</td>
<td>No Spots &gt; 1.0mm</td>
</tr>
<tr>
<td>Fail</td>
<td>Spot &gt; 1.0mm</td>
<td>Spot &gt; 1.0mm</td>
</tr>
</tbody>
</table>

Table 1: ASTM D1743 and D5969 Corrosion Test Criteria [1,3]
1.2 Limitations

The static and dynamic ASTM bearing corrosion tests give an indication of the efficacy of a lubricating grease to protect metal surfaces from corrosion. However, those methods are subjective because they rely on the eyesight and judgement of the person performing the test. The reproducibility of results from EMCOR tests run in salt water is two grades [2]. This means that one laboratory may report no corrosion (rating 0) while another laboratory may report up to 1% of surface area corrosion (rating 2, light corrosion) for the same grease and standard test method.

The limited reproducibility is due partly to the subjectivity and human error that reside in the grading process. If observations of corrosion are more accurate and precise, then it becomes possible to use a rating scale with finer resolution. These points motivated this project to develop a corrosion detection system that utilizes a computational method to determine and analyze corrosion on bearings.
1 Materials and Methods

1.1 Corrosion Detection System

The new corrosion detection system consists of a light source, a camera and computer analysis software (Figure 2). The custom analysis software system was developed in house. It calculates the actual percentage of surface corrosion on a bearing race after a corrosion test.

The corrosion detection system utilizes a frequency-controlled light source calibrated to a proprietary frequency to illuminate a bearing race. Light reflected from the bearing race is filtered with a polarizing filter before it reaches a high-resolution camera (patent pending). This hardware was designed specifically to reduce noise and other unwanted effects, and to allow stronger discrimination of corrosion (edge detection). The camera creates digital images of bearing surfaces. The entire system is contained in a blacked-out sealed box (to control the amount of light samples are exposed to) and placed on an anti-vibration table.

Proprietary software (a series of computational algorithms) is used to analyze images of the bearing surface (Appendix). This software was written in an embedded language with various elements in C++ and LabVIEW (National Instruments).

The new corrosion detection system was designed to match the accuracy of the human eye for direct comparison with ASTM corrosion tests. The accuracy of human vision, also known as visual acuity, was calculated [7] and found to be 30 μm. It is vital to understand human visual acuity as magnification devices cannot be used with the ASTM methods discussed above. Differences in visual acuity of individuals can affect corrosion test results.

For the statistical analysis and DOE calculations in this study, Minitab 17 (Minitab, Inc.) was used.
1.2 Lubricating Grease

The lubricating grease used in this study satisfied MIL-PRF-32014, “Performance Specification: Grease, Aircraft and Instrument” [5]. This grease is formulated from lithium soap thickener and polyalphaolefin base oil. It is used as an aircraft grease for bearings, gears, and other mechanisms. As this is a military specified lubricant, there are multiple suppliers. This study compared products from two different suppliers. The typical physical properties of MIL-PRF-32014 greases are listed below (Table 2).

<table>
<thead>
<tr>
<th>Physical Property</th>
<th>Test Limits</th>
<th>Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Viscosity 40°C</td>
<td>140cSt, max</td>
<td>ASTM D-445</td>
</tr>
<tr>
<td>Viscosity 100°C</td>
<td>16cSt, max</td>
<td>ASTM D-445</td>
</tr>
<tr>
<td>Pour Point</td>
<td>−35°C</td>
<td>ASTM D-97</td>
</tr>
<tr>
<td>Cone Penetration, P60</td>
<td>265-305</td>
<td>ASTM D-217</td>
</tr>
<tr>
<td>Cone Penetration, P100K</td>
<td>350 max</td>
<td>ASTM D-217</td>
</tr>
<tr>
<td>Copper Corrosion</td>
<td>1b max</td>
<td>ASTM D-4048</td>
</tr>
<tr>
<td>Corrosion Prevention 5% SSW</td>
<td>Pass</td>
<td>ASTM D-5969</td>
</tr>
<tr>
<td>Cleanliness (25-74 micrometers)</td>
<td>1,000 max</td>
<td>FED-STD-791-3005.4</td>
</tr>
<tr>
<td>Cleanliness (&gt;75 micrometers)</td>
<td>0</td>
<td>FED-STD-791-3005.4</td>
</tr>
<tr>
<td>Dropping Point</td>
<td>200°C min</td>
<td>ASTM D-2265</td>
</tr>
<tr>
<td>Evaporation (22 hours, 177°C)</td>
<td>10% max</td>
<td>ASTM D-2595</td>
</tr>
<tr>
<td>Extreme Pressure Wear Properties</td>
<td>0.80 max</td>
<td>ASTM D-5706</td>
</tr>
<tr>
<td>Ball Bearing Lubrication Life (160°C)</td>
<td>400 hours min</td>
<td>ASTM D-3336</td>
</tr>
<tr>
<td>Load Wear Index</td>
<td>30kgf min</td>
<td>ASTM D-2596</td>
</tr>
<tr>
<td>Low Temperature Torque (−54°C)</td>
<td>1.4Nm max starting</td>
<td>ASTM D-1478</td>
</tr>
<tr>
<td>Low Temperature Torque (−54°C)</td>
<td>0.5Nm max 1 hour running</td>
<td>ASTM D-1478</td>
</tr>
<tr>
<td>Oil Separation</td>
<td>8% max</td>
<td>ASTM D-6184</td>
</tr>
<tr>
<td>Oxidation Stability, 500 hours</td>
<td>35psi max</td>
<td>ASTM D-942</td>
</tr>
<tr>
<td>Water Washout (41°C)</td>
<td>15% max</td>
<td>ASTM D-1264</td>
</tr>
<tr>
<td>4-Ball Wear</td>
<td>0.65mm max</td>
<td>ASTM D-2266</td>
</tr>
</tbody>
</table>

Table 2: MIL-PRF-32014 Grease Physical Properties [5]

1.3 Experimental Design

A three-factor, two-level full factorial DOE was created to evaluate the new corrosion detection system in a test of variability between two manufacturers of MIL-PRF-32014 grease, two lots from each manufacturer, and test results from two independent labs (Table 3). The corrosion preventive properties of the lubricants were evaluated for bearings tested in a solution of 3% synthetic sea water per the ASTM method. Tests were run in accordance with ASTM D6138 (EMCOR).
The goals of this study were:

1. Determine the repeatability of results from the new corrosion detection system.

2. Compare the results and statistical spread of the results for percent corroded surface area measured with the new corrosion detection system versus the standard visual EMCOR corrosion ratings.

3. Investigate three factors (supplier, lot, test lab) that may have a statistically significant effect on the corrosion of bearings lubricated with MIL-PRF-32014 grease.

![Full Factorial DOE Design](image)

**Figure 3: Full Factorial DOE Design**

2 Results

2.1 Visual Versus Computational Methods

Thirty-two bearings were tested, disassembled and evaluated with the standard visual rating system as well as the new corrosion detection system (Table 3). There were two replicates for each of 16 cases.
Table 3: Corrosion test results from standard visual evaluation and the new computational corrosion detection system

<table>
<thead>
<tr>
<th>Material</th>
<th>Lab</th>
<th>Lot</th>
<th>EMCOR Rating</th>
<th>EMCOR Corrosion</th>
<th>Computational Result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manufacturer 1</td>
<td>Lab 2</td>
<td>Lot 1</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.417%, 0.027%</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>Lab 1</td>
<td>Lot 1</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.488%, 0.107%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 1</td>
<td>Lot 2</td>
<td>1.2</td>
<td>&lt; 1%</td>
<td>0.004%, 0.008%</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>Lab 1</td>
<td>Lot 2</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.108%, 0.158%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 1</td>
<td>Lot 2</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.024%, 0.109%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 1</td>
<td>Lot 1</td>
<td>0.0</td>
<td>&lt; 1%</td>
<td>0.000%, 0.000%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 2</td>
<td>Lot 2</td>
<td>1.2</td>
<td>&lt; 1%</td>
<td>0.003%, 0.057%</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>Lab 1</td>
<td>Lot 2</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.133%, 0.066%</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>Lab 2</td>
<td>Lot 2</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.299%, 0.166%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 1</td>
<td>Lot 2</td>
<td>1.2</td>
<td>&lt; 1%</td>
<td>0.003%, 0.009%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 2</td>
<td>Lot 1</td>
<td>0.1</td>
<td>&lt; 1%</td>
<td>0.000%, 0.002%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 2</td>
<td>Lot 2</td>
<td>1.2</td>
<td>&lt; 1%</td>
<td>0.003%, 0.010%</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>Lab 2</td>
<td>Lot 2</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.121%, 0.186%</td>
</tr>
<tr>
<td>Manufacturer 1</td>
<td>Lab 1</td>
<td>Lot 1</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.222%, 0.042%</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>Lab 2</td>
<td>Lot 1</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.203%, 0.095%</td>
</tr>
<tr>
<td>Manufacturer 2</td>
<td>Lab 1</td>
<td>Lot 1</td>
<td>2.2</td>
<td>&lt; 1%</td>
<td>0.393%, 0.120%</td>
</tr>
</tbody>
</table>

Every bearing had less than 1% corrosion using the standard EMCOR rating system. However, their categories ranged from 0 to 2, with the majority of these bearings category 2.

Results from the new corrosion detection system were all less than 0.5% corrosion of the total surface area of each bearing. All of these results were plotted in a histogram, and a normal statistical distribution was fitted over it (Figure 4). Most of the results fell within the normal distribution, and only three samples fell outside the +/-3σ level or more than three standard deviations from the average.

The results from the full factorial DOE were analyzed using Minitab software. The standardized effects of the factors were plotted on a Pareto Chart (Figure 5). Only one factor had a statistically significant effect on the percent corroded surface area: the manufacturer of the grease. The choice of independent test lab and grease lot did not have a statistically significant correlation with percent corrosion and can be ignored as variance factors in this analysis.

While both manufacturers of MIL-PRF-32014 grease claimed that their products met the specifications, the new corrosion detection system found a statistically measurable difference between the two, possibly due to a difference in the additive formulations for the MIL-PRF-32014 greases, processing methods, etc.
Figure 4: Histogram of MIL-PRF-32014 Samples with Normal Distribution Curve

Figure 5: Pareto Chart of the DOE Standardized Effects of Factors on Percent Corrosion
A 2-sample t-test was performed to take a closer look at the distribution of data (Figure 6) for greases from the two manufacturers. From this plot, it is apparent that the average surface corrosion differs significantly. There is almost no overlap of the 95% confidence intervals for the two means. This confirms that greases from these two suppliers are significantly different with respect to corrosion protection.

![Distribution of Data](image)

Figure 6: Distribution of Results (Percent Corrosion) by Grease Manufacturer

2.2 Additional Materials Tested

To further illustrate the capability of this new corrosion detection system, several perfluoropolyether (PFPE) greases were tested (Table 4). These PFPE greases did not perform well in the EMCOR test, and all of the bearings received a visual rating of 3 (1 – 5% corroded surface area). The computational results for the three PFPE grease ranged from 1.12 to 4.42% corroded surface area. This higher precision of the results from the new corrosion detection system illustrates the value of this computational method.
3 Conclusions

This research illustrated the use and value of a corrosion detection system for measuring corrosion on bearing races. Statistical analysis and surrogate modeling through a designed experiment showed that this computational approach is sensitive enough to detect subtle differences in the formulation or manufacturing process of MIL-PRF-32014 grease. There was good repeatability of results for bearings that were tested multiple times and by different operators.

A wide range of percent corrosion was observed within a specific rating category as currently used in the ASTM D6138 EMCOR method. With the new corrosion detection system, it is feasible to accurately measure percent surface corrosion, which will allow specifications to be written with more specific requirements and research and development to be carried out with a higher level of accuracy.

This new corrosion detection system is accurate enough to detect small but statistically significant differences between two manufacturers of MIL-PRF-32014 grease. This presents the opportunity to use this testing methodology for more accurate screening experiments and differentiation of materials.

The bearing corrosion tests outlined in the ASTM D1743, D5969, and D6138 are excellent. This research shows an opportunity to improve these tests by using more accurate technology to measure surface corrosion on bearings instead of subjective human visual evaluation.
4 Acknowledgements

I would like to thank my colleagues in the Application Development and Validation Testing (ADVT) Lab at Nye Lubricants Inc. who conducted the EMCOR testing, specifically Mason Wood and Kevin McBarron. I would also like to thank Brendan Smith and Richard Bellizzi, also from Nye Lubricants, for their support with the development of the Corrosion Analysis system.

Appendix

The visual acuity of the average sighted person is 1/60 of a degree, also known as 1 minute of arc. Using the trigonometric sketch of the human eye in Figure 7, the smallest size visible in microns can be calculated as shown below.

![Figure 7: Trigonometric Sketch of the Human Eye](image)

First we start with the basic trigonometric formula with \( d \) being the viewing distance

\[
2d = \arctan \frac{\alpha}{2}
\]

Where \( \alpha \) represents the visual acuity expressed in radians.

\[
\alpha = \pi \frac{\theta}{180}
\]

Under the assumptions that the closest object an adult can focus on is approximately at 100 mm and the average acuity is 1 minute of arc, this would mean that the smallest visible size would be around 30 microns [7].
Using this calculated value as the minimum size for detection on the bearing race, anything smaller would be ignored for rating but used in the calculation of overall surface corrosion. In order for the computational bearing analyzer to handle the shape of the bearing race, the field of view (FOV) of the lens and the area of the race were used to determine how many images to take without getting any fringe effects from the curvature of the race. After the computational bearing analyzer takes a series of high resolution images, software stitches them together to form one single image.

This stitched image now goes through a series of calculations called edge detection. The purpose of applying an edge detection algorithm is to find the transition areas on the bearing surface or to identify one object from another [8]. The process of edge detection is used to find the boundaries of an image where there is significant change in gray-level intensity. In this case, software finds the edges where the bearing surface transitions from clean metal to an area of corrosion.

The edge detection on this surface requires the gradient and magnitude to be calculated. In order to determine the gradient of the surface, we need to measure the first derivative which will give us the positive or negative spike at the boundary point. As the image of the bearing surface is a two-dimensional discretized gray-level function \( f(x,y) \), the gradient (magnitude) of this function \( \nabla f \) is given by the following equation.

\[
\nabla f = \sqrt{\left(\frac{\partial f}{\partial x}\right)^2 + \left(\frac{\partial f}{\partial y}\right)^2}
\]

The gradient direction is given as an angle measured by

\[
\phi(x,y) = \tan^{-1}\left(\frac{\partial f}{\partial y}\right)\frac{\partial f}{\partial x}
\]

where \( \phi \) is the angle with respect to the y-axis. The gradient can then be approximated to the following equation for simpler computation.

\[
\nabla f \approx \left|\frac{\partial f}{\partial x}\right| + \left|\frac{\partial f}{\partial y}\right|
\]

The function for these bearing surfaces is a discrete function and the derivatives can be approximated by difference equations. These derivatives are calculated using masks where the term \( \frac{\partial f}{\partial y} \) is the difference in the x-direction and is the difference in the y-direction. From this, the difference between pixels in the rows and columns of a mask can be found. Using an example of a 3 X 3 mask of pixel values (Figure 8), the approximate gradient can be calculated using \( x5 \) and \( x3 \) to get the derivative in the x-direction and \( x7 \) and \( x1 \) for the derivative in the y-direction.
There are many different types of edge detection algorithms like Prewitt, Sobel, and Laplace operators. Some of the algorithms will increase noise as they are first-order derivative-based operators, while second-order derivative-based operators will typically reduce noise and smooth out the edges. Using the Sobel operator as an example of an edge detection operator will balance the ease of using a first derivative computation and also provide a smoothing action on the edge detection [9]. This will allow the gradient $\nabla f$ and gradient direction $\phi$ to be calculated easily and provide a smoothed response on the edge detection with this operator:

$$
\frac{\partial f}{\partial x} \approx (x_6 + 2x_7 + x_8) - (x_0 + 2x_1 + x_2)
$$

$$
\frac{\partial f}{\partial y} \approx (x_2 + 2x_5 + x_8) - (x_0 + 2x_3 + x_6)
$$

$$
\nabla f \approx |x_6 + 2x_7 + x_8 - x_0 - 2x_1 - x_2| + |x_2 + 2x_5 + x_8 - x_0 - 2x_3 - x_6|
$$

After the edge detection operator is applied to the entire image, the image is transformed based on the areas that are not within the acceptable gray scale. (The acceptable gray scale was determined through the measurement of a very large sample of pristine surfaces for bearings that had been cleaned recently and dissembled, but not exposed to a corrosive material. The results were used to construct a histogram of the distribution of normal gray scale intensities for clean metal surfaces without corrosion.) After this transformation, the corroded surface area is calculated, and the percentage of corroded surface area is reported.
References


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Edward J. Palecki, 91, originally from Pittsburgh, Pennsylvania, passed away on August 2, 2018. Ed resided at Casey’s Pond Senior Living Community in Steamboat Springs, Colorado. He is survived by his sister, Helyne McMullen, who lives at Casey’s Pond, and by 4 daughters and 2 sons: Ann Fierstos (Ed) of Hendersonville, North Carolina; Michael Palecki (Emma) of Tallahassee, Florida; Jean Ray (Jim Chubrilo) of Steamboat Springs, Colorado; Paul Palecki (Susan) of Royse City, Texas; Mary Palecki of Littleton, Colorado; and, Carol Palecki of Oakland, California. He is also survived by 10 grandchildren and 3 great-grandchildren. He is preceded in death by his son, Thomas Palecki and his beloved wife of 60 years, June Palecki.

Ed was born on May 5, 1927 in Pittsburgh, Pennsylvania. He graduated from St. Francis de Sales High School in McKees Rocks, Pennsylvania, where he began dating his future wife, June. He earned his Bachelor’s Degree at Duquesne University in Pittsburgh, Pennsylvania and pursued post-graduate studies later in life in Kansas City, Missouri. Ed served in the Navy on the USS Oglethorpe in the South Pacific during the conclusion and aftermath of WW II.

Ed worked with Trans World Airlines in various management capacities for 34 years. While employed with TWA, he moved his family from Pittsburgh to Baltimore, Maryland, to Freeport, New York, and finally to Kansas City, Missouri. After a short retirement, Ed became General Manager of the National Lubricating Grease Institute in Kansas City, where he worked for an additional six years

Ed loved to travel and circumnavigated the globe numerous times, often with his wife, June, and his sons and daughters. He visited family and hosted family reunions overseas in Greece, India, Japan, Peru, and Lithuania, the birthplace of his beloved parents, Adolph and Anna. He was an excellent photographer and won several awards for his work. He especially enjoyed sailing his small boat on Weatherby Lake near the family’s home in Kansas City. Ed was passionate about music, literature, nature and outdoor adventures with his family. He had an enthusiastic spirit and described the afterlife as “the next great adventure”.

Above all, Ed was a wonderful father to his large and loving family and a devoted husband to June. He lived a full life and will always be remembered as a good, honest and kind man with a warm and wonderful smile. He will be greatly missed by his family and all those who had an opportunity to know him.

A funeral mass will be held in the chapel at Holy Name Catholic Church, 524 Oak Street, Steamboat Springs, Colorado on Monday, August 13 at 3:00 PM. A reception will be held afterwards at 4:30 PM at Casey’s Pond.

His ashes will be placed with those of his wife, June and son, Tom at Resurrection Cemetery in Kansas City, Missouri, at a later date. At the family’s request, in lieu of flowers, donations may be made to the Casey’s Pond Employee Appreciation Fund, Casey’s Pond Senior Living, 2855 Owl Hoot Trail, Steamboat Springs, CO 80487.
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