A New Form of Rolling Contact Damage in Grease-Lubricated, Deep-Groove Ball Bearings

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A new form of rolling contact damage was discovered in the fatigue tests of series 6206 deep groove ball bearings under pure radial load with grease lubrication. The appearance of the damage reveals a few cracks at right angles to the rolling direction of the ball, along with the formation of a dent about 20 µm deep in the raceway surface. This is found to occur only on the stationary outer ring raceway and the ball surface and is distributed widely within the load zone. Furthermore, under the raceway surface several cracks propagating into the substrate at an angle of about 60-80 degrees relative to the raceway surface are observed not only under the damage site but also in other nearby locations. Only the cracks at the damage position open up to the surface. The grease used for the test contained a lithium complex thickener with mineral oil as the base oil with a kinematic viscosity of 141 mm²/s at 40°C. On the test bearings two pure radial load levels of 9.14 and 12.13 kN were applied. In order to prevent the occurrence of seizure at each load, the speed of the inner ring of the test bearing was maintained at 1800-2500 min⁻¹ and 600-800 min⁻¹, respectively, to keep the outer ring circumference temperature below 65°C. It is suggested that the damage is caused by metal-to-metal contacts due to lubricant starvation under grease lubrication and to a decrease in oil film thickness due to local increases in temperature.

KEY WORDS
Rolling Contact Fatigue; Fatigue Damage; Grease Lubrication; Deep-Groove Ball Bearing; Crack; Dent

INTRODUCTION
In life tests of rolling bearings, spalling damage usually occurs on the surface of either the rolling element or the raceway as a result of fatigue of the bearing materials. The spalled surface appears as a several hundred micrometer deep pit with an uneven surface demonstrating evidence of crack propagation. On the damaged raceway surface, the uneven surface of the pit expands randomly in all directions due to repeated running of the rolling elements. On the rolling elements, in the case of balls, the surface damage spreads over the whole ball surface, whereas in the case of rollers, the progress in the pitted area is observed both in the axial as well as in the rolling direction.

Several studies on the life of ball bearings under grease lubrication have been reported in the literature. Credit for important research studies goes to Snare (1) and to Committee No. 126 on Rolling Bearing Life of the Japan Society for the Promotion of Science (2). Snare carried out life tests on 500 deep-groove ball bearings with the purpose of determining the minimum life, while Committee No. 126 had examined the effect of four types of greases on the bearing life by only changing the base oil but with the same thickener. However, as none of them had any special interest related to the rolling fatigue damage, they assumed that the appearance of spalling damage in rolling fatigue with grease lubrication is the same as for oil lubrication.

Recently, attention has been directed towards a large number of research studies carried out (Tamada, et al. (3); Murakami, et al. (4); Shibata, et al. (5); Tamada and Tanaka (6); Iso and Yokouchi (7)) that revealed that the grease-lubricated bearings in auxiliary automotive devices were damaged in less than 1/10th of the calculated rated life. After confirming the occurrence of a unique form of damage, which was never observed in conventional oil lubrication, the mechanism of occurrence of such damage has been elucidated and the method of preventing such damage is under examination. The characteristics of the damage are observed as the occurrence of spalling and the progress of cracks in random directions under the damaged surface. This occurrence of damage is found only on the stationary ring and not...
on the rotating ring and with the generation of white structure near the damage. It has been suggested that the absorption of hydrogen generated from the grease into the raceway leading to a possible hydrogen embrittlement may be an explanation of the mechanism of the occurrence of damage.

In the research related to Reliability Data Bank for Mechanical Material and Components under the banner of the Meiji University Academic Frontier Project, rolling fatigue tests have been carried out for deep-groove ball bearings in which 80% of them were lubricated with grease. This report presents the characteristics of the damage that has appeared in these tests.

### EXPERIMENTAL METHOD

In this experiment, the test bearing used was an open type #6206 series deep-groove ball bearing with inner diameter 30 mm, outer diameter 62 mm, and width 16 mm as shown in Fig. 1. Nine balls are assembled into the test bearing. The inner ring, the outer ring, and the balls are made of ASTM-A295 (52100) steel. The ball cage is made of plastic and has a snap shape. The test rig, shown in Fig. 2, has an overhang style construction consisting of a main spindle supported on two angular contact ball bearings of #7210 series with face-to-face assembly and one deep-groove ball bearing of #6210 series and the test bearing is mounted on one end. On the other end of the shaft there is a device for direct coupling of the drive motor. On the test bearings two downward radial load levels of 9.14 and 12.13 kN are applied. As a result of these load applications on the test bearings, the estimated maximum Hertzian contact stress developed and the axes of the ellipse on the inner and outer ring of the test bearings are shown in Table 1. To control the occurrence of seizure at the two loads, the test bearing’s inner ring speed was maintained at 1800-2500 min\(^{-1}\) and 600-800 min\(^{-1}\), respectively, to keep the outer ring circumference temperature below 65°C. Incidentally, the calculated result shows that the load zone is distributed between ±90 degrees around the maximum rolling element load position in the upper side of the test bearing. On the lubrication in the test bearing, it can be considered that there is no effect of gravity because grease adhered to all parts of the circumference of the outer ring shoulder after the test.

Table 2 shows the properties of the lubricating grease, which contains a special lithium complex as a thickener, and mineral oil

### Table 1—Contact Stress and Contact Ellipse in Test Bearing

<table>
<thead>
<tr>
<th>Bearing Load, kN</th>
<th>9.14</th>
<th>12.13</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum rolling element load, kN</td>
<td>4.44</td>
<td>5.89</td>
</tr>
<tr>
<td>Maximum hertzian contact stress, GPa</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inner race</td>
<td>2.80</td>
<td>3.08</td>
</tr>
<tr>
<td>Outer race</td>
<td>3.36</td>
<td>3.69</td>
</tr>
<tr>
<td>Contact ellipse, mm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inner race</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Minor axis</td>
<td>0.38</td>
<td>0.42</td>
</tr>
<tr>
<td>Major axis</td>
<td>8.16</td>
<td>8.92</td>
</tr>
<tr>
<td>Outer race</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Minor axis</td>
<td>0.66</td>
<td>0.74</td>
</tr>
<tr>
<td>Major axis</td>
<td>3.80</td>
<td>4.18</td>
</tr>
</tbody>
</table>

### Table 2—Characteristics of Grease

<table>
<thead>
<tr>
<th>Thicker</th>
<th>Special Lithium Complex</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base oil</td>
<td>Mineral oil</td>
</tr>
<tr>
<td>Kinematic viscosity</td>
<td></td>
</tr>
<tr>
<td>40°C</td>
<td>141 mm(^2)/s</td>
</tr>
<tr>
<td>100°C</td>
<td>14.76 mm(^2)/s</td>
</tr>
<tr>
<td>Dropping point</td>
<td>300°C</td>
</tr>
<tr>
<td>Consistency (60 W)</td>
<td>280</td>
</tr>
<tr>
<td>Oil separation (100°C, 24 h)</td>
<td>0.80%</td>
</tr>
</tbody>
</table>
as the base oil with kinematic viscosity 141 mm²/s at 40°C. Grease was injected equally through the cage pocket openings before the experiment into the space between the inner and outer ring of the bearing by means of a manually operated pump to fill almost 40% of the total free space.

In this experiment, the minimum oil film thickness $H_{\text{min}}$ was calculated by using an equation from Hamrock and Dowson (8) as shown below.

$$H_{\text{min}} = 3.63 U^{0.68} G^{0.49} W^{-0.073} \left[ 1 - e^{-0.7(R_y/R_x)^{0.64}} \right]$$

where

- $U$ = velocity parameter
- $G$ = material parameter
- $W$ = load parameter
- $R_x$ = equivalent radius in the rolling direction of the ball
- $R_y$ = equivalent radius in the direction of bearing axis.

To do the calculation, it is assumed that enough base oil is available on the contact surface between the ball and the ring raceway to avoid starvation, and a lubricating oil temperature of 60°C is maintained at rotational speeds of 2000 and 800 min⁻¹ for radial loads 9.14 and 12.13 kN, respectively.

The minimum oil film thickness, $H_{\text{min}}$, calculated for a radial load 9.14 kN between the ball and the inner ring contact surface is about 0.30 μm and on the contact surface of the outer ring is about 0.37 μm, whereas for radial load 12.13 kN these were about 0.16 and 0.19 μm, respectively. According to Aihara and Dowson (9) in the case of grease lubrication, it has been found that with the progress of running time, the oil film formed by the base oil gradually decreases to 70% of the initial value. In this research work, if that is the case, then for a load of 9.14 kN the film thicknesses are 0.21 and 0.26 μm, respectively, and for a load of 12.13 kN they are 0.11 and 0.13 μm, respectively. If these are expressed as a ratio of surface roughness, that is the ratio $\Lambda$, they will be 6.0, 7.3, 3.2, and 3.8, respectively.

CHARACTERISTICS OF DAMAGE

The appearance of the damage that occurred in these experimental fatigue tests is shown in scanning electron microscope photographs in Figs. 3 and 4 with the load 9.14 kN for a rotation life of $26.46 \times 10^6$ cycles and with the load 12.13 kN for a rotation life of $73.58 \times 10^6$ cycles, respectively. In Figs. 3 and 4, the ball is rolling in the lengthwise direction of the photograph. A large number of cracks appeared in the rolling track, and it was found that they propagated into the raceway surface almost perpendicularly to the rolling direction of the ball. In Fig. 3, it can be seen that one crack leads to the occurrence of a small shallow spall, but in Fig. 4 no spalling is evident.

Thirteen rolling fatigue tests were conducted on bearings with a radial load of 9.14 kN. Among these, three had ball damage and two were suspended, and on the remaining eight, the damage occurred on the outer ring raceway. After the experimental tests, out of eight samples, six (75%) revealed the occurrence of cracks on the raceway similar to that shown in Fig 3. In the case of the bearing load of 12.13 kN, out of 17 tests, 6 had ball damage and one was suspended. In the remaining 10 the damage occurred on the outer ring raceway, and out of these, 9 (90%) revealed only the cracks. On the other hand, the occurrence of spalling was found on two bearings with a load of 9.14 kN and one with a load of 12.13 kN. These results suggest that this newly identified rolling contact damage occurs frequently under grease lubrication.

This form of damage occurs at a number of load cycles greater than the calculated catalogue rated life for both load conditions. For example, for a bearing load of 9.14 kN the minimum life was $10.13 \times 10^6$ rotations as against the calculated rated life of $9.7 \times 10^6$ rotations; that is, 1.04 times the calculated rated life. In the case of a bearing load of 12.13 kN, compared with a calculated rated life of $4.16 \times 10^6$ rotations, the test life was found to be $6.05 \times 10^6$ rotations; i.e., 1.45 times of the calculated rated life.

The shape of the damaged portion in Fig. 3 was measured using a Taylor-Hobson form shape measuring system and by using the software that transforms the shape to a plane surface, a three-dimensional image is shown in Fig. 5. Figure 6 shows the...
cross section profile of the damaged portion in which the depth of the dent is found to be around 19 $\mu$m. In other measured test bearings the depth was also about 20 $\mu$m. When such damage does occur, the vibration acceleration is increased, which is detected by the vibration sensor mounted on the test bearing housing, and the test stops. With the test rig stopped, in order to confirm the existence of the damage, when the shaft of the test rig is rotated by hand, the ball will fall in the position of the damaged portion of the test bearing, which will not permit smooth rotation of the shaft. On the other hand, when the shaft is rotated by the motor, a noise caused by the bearing failure is generated in the test bearing.

In fact this damage is not spalling. There is a continuation of the fine scratches superfinishing process between the damaged zone and the undamaged portion on the outer ring raceway surface, as confirmed from the observation of Figs. 3 and 4.

In order to confirm whether the damage has occurred only at one place on the raceway surface in this test, the form in the direction of the circumference of the outer ring raceway surface was measured by means of a Talyrond roundness measuring system. This result is shown in Fig. 7. In the figure, the solid line shows the form of the outer ring raceway surface and the dotted line expresses the true round. According to the figure, one portion of the solid line goes outside the round, while the rest of the large portion is drawn exactly as an arc of a circle. This clearly indicates that there is only one damaged portion on the outer ring raceway surface, and only that portion has a dent, which agrees with the result shown in Figs. 5 and 6. Furthermore, the figure represents that the position of the damage is about 40 degrees away from the maximum rolling element load position in the exit side within load zone.

The occurrence of damage positions in the load zones of 11 test bearing samples at the two load levels is shown in Fig. 8. The X-axis indicates the occurrence position of the damage in degrees within the load zone; a negative sign (−) means the entrance side relative to the maximum rolling element load position, which is set at 0 degrees, and a positive sign (+) means the exit side. The Y-axis gives the frequency of the damage. As evident from Fig. 8, such positions have a wide range of distribution from the maximum rolling element load position relative to the direction of the rolling of the ball; i.e., from −50 degrees of the load zone entrance side up to +40 degrees of the exit side.

Figure 9 presents the observation of the existence of damage and its progress by measuring the vibration acceleration. The X-axis shows the cumulative rotation number and the Y-axis indicates the rms value of the vibration acceleration. The test
condition for this case was set for a load of 9.14 kN and a rotation speed of 2000 min\(^{-1}\). As evident from Fig. 9, after reaching a cumulative number of rotations of 8.505 \(\times 10^6\), the vibration level increases approximately constantly. Even for other tests with the same load, and again in the case of a load 12.13 kN for the entire 12 test bearings observed, an increasing trend was observed similar to Fig. 9. Although it is not easy to see in Fig. 9, the initial vibration acceleration increased approximately uniformly with the cumulative number of rotations, with a slope of approximately 4 \(\times 10^{-9}\) m/s\(^2\)/rev. In Fig. 9, as the slope in the progress of damage beyond the cumulative number of rotations 8.505 \(\times 10^6\) is about 4 \(\times 10^{-9}\) m/s\(^2\)/rev., it is clear that the slope at the period of damage development is three orders larger than that of the initial running.

In an example of a test bearing with a load 9.14 kN, at a rate of rotation 2000 min\(^{-1}\), and a life with a cumulative number of rotations of 8.539 \(\times 10^6\), Fig. 10 shows the condition of the subsurface under the raceway surface in the neighborhood of the damage. Many cracks have developed at random intervals in the rolling direction of the rolling element on the damaged surface, (ii) no occurrence of cracks at a right angle to the rolling direction of the rolling element on the damaged surface, (iii) the damage occurs only in the neighborhood of the spalling (Yoshioka (10), (11)).

Comparison with Damage in Oil Lubrication
The damage found in this experiment with grease lubrication was different from damage seen with oil lubrication until now, and a discussion of the difference follows.

The fatigue damage that occurs in oil lubrication is spalling, and not the damage reported and observed in Figs. 3 and 4. Except for peeling, spalling generally initiates in the subsurface due to the maximum shear stress. If the maximum shear stress depth for the outer ring in this test is estimated from the minor diameter of the contact ellipse in Table 1, based on the load they would be 170 and 190 \(\mu\)m, respectively. Thus according to these values, it is estimated that in oil lubrication, the damage reaches a position somewhat deeper into the subsurface.

The occurrence position of the spalling in case of oil lubrication is generally found either at or close to the maximum rolling element load position. No one has reported finding a position several tens of degrees away from the maximum rolling element load position. In addition, nothing like this has been experienced by the author. In fact, the cracks under the surface are only found in the neighborhood of the spalling (Yoshioka (10), (11)).

Subsequently when trying to compare the process of the occurrence of damage from the change in the vibration acceleration, in the case of oil lubrication the acceleration at the moment of damage suddenly increases, generally leading to the stoppage of the test rig (Mano, et al. (12)).

When the damage in oil lubrication is compared with the observations in the previous sections on the damage in bearing tests with grease, there is a clear difference. It is therefore considered that the damage found with grease is different from the damage in oil lubrication.

Comparison of Damage in Auxiliary Automotive Device Bearings
An examination of the problematic damage in auxiliary automotive device bearings lubricated by grease is carried out. The common characteristics of damage in the current bearing fatigue tests and the auxiliary automobile device bearings are as follows: (i) the occurrence of cracks at a right angle to the rolling direction of the rolling element on the damaged surface, (ii) no occurrence of damage on the running ring, and (iii) the damage occurs only on the stationary ring and the rolling element.

However, three differences in the nature of the damage are seen. In the case of the automotive bearings, a large number of cracks are observed progressing in various directions on the circumference of the spalling (Tamada, et al. (3)). Compared to this, in the current experimental test as shown in Fig. 10, the direction of progress of the cracks is at an angle to the raceway surface, and is occurring not only on the damaged portion but also at positions away from this. The bearing life in the case of automotive device bearings is found to be 1/10 to 1/20 of the calculated rated life (Iso and Yokouchi (7)), but in our research, such a large reduction in life versus the calculated rated life has not been found. Although the characteristics of the damage seem similar, spalling is taking place in the automotive bearings, but not in our tested bearings, where there is formation of cracks.
a dent, and several cracks progress there at right angles in the rolling track. When these three points are taken together, there is a clear difference between the damage of the automobile device bearings compared to the damage found in these experiments.

**OTHER CHARACTERISTICS OF DAMAGE IN THIS TEST**

A discussion on the formation of the dents in these fatigue tests is needed. One possibility is that this might have occurred due to the application of excessive force at the time of mounting the test bearings or due to an external shock force or vibration during the test. In the case of the test of Fig. 9, if an indentation of about 20 µm deep is formed at the time of mounting of the bearing, an increase in the vibration acceleration of around 1 m/s² should have been observed at the beginning of the test, which is not observed here. In addition, as no sudden change in the vibration acceleration during the test could be seen, and the dent was discovered only after the stoppage of the test rig due to the condition of the increased vibration acceleration, it is unlikely that the dent occurred due to a shock load during the test.

Furthermore, if the dent is formed at the time of mounting of the test bearings or due to a deformation during the test, then there should have been indentations at each point of the ball interval on the outer ring. However, as such marks cannot be found in Fig. 7, it is confirmed from a different point of view that the above-mentioned dent-forming possibilities are unlikely.

In addition to this, on the running inner ring in this test, no damage was found. An explanation for this is that, as shown in Table 1, the maximum Hertzian contact stress developed on the outer ring in this experiment is 1.15 times that on the inner ring. However, some of the damage occurred at lower Hertz stress on the outer raceway than the maximum Hertz stress at the inner raceway as shown in Fig. 8. On the other hand, the oil film thickness on the outer ring becomes about 1.1-1.2 times of that on the inner ring. But if one looks from the λ value, the ratio of the minimum oil film thickness to the surface roughness value, for both the inner and the outer rings, the value is λ ≥ 3.2, and hence in this region (Takata (13)), it is considered to have no effect on the life. Therefore, it is difficult to consider that the contact stress alone is enough to explain the cause of damage on the outer ring.

**MECHANISM OF OCCURRENCE OF DAMAGE IN THIS TEST**

Although the mechanism of the occurrence of the damage is not clear at present, it is considered to be due to lubricant starvation.

It cannot be considered that during the test the base oil separated from the thickener always forms a lubrication film on the contact surface, as assumed by Aihara and Dowson (9). In the initial stage of the experiment, such a condition may exist, but with the progress of the test for a long period the lubrication film may get thinner. At some point a lump of the grease, composed of the base oil and the thickener, may enter the contact. If this happens during the test, the bearing temperature may drop by several degrees due to the temporarily enhanced lubrication.

Furthermore, as evident from Fig. 10, the cracks under the surface are not only below the damaged portion, but they have developed at random intervals at positions beyond this region. Thus, the occurrence of metal-to-metal contact due to lubricant starvation under grease lubrication and a decrease of oil film thickness due to local temperature increases at various places under the load zone and is likely to be the cause of the occurrence of cracks. As the damage occurs at positions with the most severe metal-to-metal contacts, the occurrence of damage in a wide range of the load zone can be explained as shown in Fig. 8.

In addition, in looking for reasons for this damage not reaching the state of spalling, it is supposed that even if a few open cracks parallel to the raceway surface under the surface occur as shown in Fig. 10 and the base oil enters, the promotion of the crack to spalling does not occur since there is not enough base oil on the contact surface.

Therefore, it is considered that the reason for this damage is the lack of lubricant in rolling fatigue testing process with grease.

**CONCLUSION**

In fatigue tests of rolling bearings using grease composed of a special lithium complex as the thickener and mineral oil as the base oil, a new form of damage appeared whose features are as follows:

1. Surprisingly, without the occurrence of spalling, a dent is formed and several cracks in the direction almost normal to the rolling direction of the ball develop under and in the neighboring zones of the dent.
2. The damage occurs only on the stationary ring. The locations of the occurrence of the damage are distributed over a wide range in the load zone. A large number of such cracks progressing at angles ranging from about 60-80 degrees to the raceway surface are observed under the raceway surface, including the zone of damage.

The cause of the occurrence of this damage is suggested to be metal-to-metal contact due to lubricant starvation under grease lubrication and to the decrease of oil film thickness originated due to local temperature increases during the rolling fatigue test under grease lubrication. In the future, there is a plan to clarify whether this damage is due to the use of this particular grease or if the cause of occurrence is indeed due to the lack of lubricant leading to metal-to-metal contacts.

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