 NTN has developed a new grease lubrication system for machine tools main spindles which features heat-induced base oil delivery from a grease chamber installed next to the bearing. The lubrication system does not require any external additional lubrication device. Experimental result demonstrates high-speed rotation of an angular contact ball bearing with a 100 mm bore diameter, that has a 1.8 million $d_{mn}$ value for definite position preloading and 2.0 million $d_{mn}$ value for constant pressure preloading. NTN's new grease lubrication system can thus be an alternative to the air-oil lubrication technique while keeping the maintenance-free advantage of the conventional grease lubrication method, in addition, it will support higher performance machine tools as well as improve customer’s work environment and reduce environmental burden.

1. Introduction

The use of higher speed main spindles has been increased used on machine tools, including machining centers, in order to achieve better machining efficiency and machining accuracy. To cope with the challenges associated with high-speed machine tool main spindles, NTN has been developing and improving various main spindle bearings and currently markets these bearings under the “ULTAGE Series” brand name. In recent years, people around the globe have been becoming increasingly concerned about the environment, so manufacturers worldwide must remain committed to eco-friendly technologies. In response to the technological challenges related to this, NTN has been developing unique bearing products through investigations into bearing lubrication techniques. The resultant bearing product lines include "eco-friendly jet lubrication angular contact ball bearings," "eco-friendly air-oil lubricated angular contact ball bearings (HSL type)" and "grease lubricated sealed angular contact ball bearings (BNS type)." Grease lubrication systems, in particular, have smaller impacts on the environment and can be maintenance-free. As a result, demand is growing for grease lubrication systems for machine tool main spindles.

One typical example of the authors’ achievement is that the above-mentioned “grease lubricated sealed angular contact ball bearing (BNS type)” has achieved a bearing life of more than 20,000 hours at a running speed equivalent to $d_{mn}$ of 1.4 million.

The authors developed a “new grease lubrication system for machine tool main spindles” that is capable of greater bearing speed and longer bearing life with grease lubrication that does not need maintenance. This novel grease lubrication system is comprised of a conventional internal design but has a grease chamber in the front side of the bearing from which only the base oil in the grease is extracted to lubricate the bearing. This novel grease lubrication system was incorporated into an angular contact ball bearing with a bore diameter of 100 mm, making the bearing capable of high-speed operation and a long bearing life. The $d_{mn}$ value reached 2 million with constant pressure preloading and 1.8 million with definite position preloading. In this paper, the authors report the achievements of their research into a novel grease lubrication system that is completely maintenance-free.
2. Structure of the New Grease Lubrication System

To achieve high speed and long life with a bearing by using grease lubrication, it is necessary to not only provide a sufficient amount of lubricant for a bearing while running at a high speed but also to maintain the supply of grease base oil from a limited amount of prefilled grease. With the authors’ newly developed grease lubrication system, the base oil in the grease is actively separated while the bearing is running, and this separated base oil is steadily supplied to the raceway surface of the bearing. The bearing structure of the authors’ new grease lubrication system is illustrated schematically in Fig. 1. As a part of the outer ring spacer, a cavity is formed in front of the bearing and the grease prefills this space. At the same time, this provides a shouldered surface near the contact point between the rolling elements and the outer ring raceway. In addition, a thin axial flow path is formed between the shouldered surface and the end point of the grease chamber, thus connecting the grease chamber to the raceway surface.

For this structure, the authors considered a base oil delivery mechanism that works as follows. The running bearing generates heat, causing the temperature of the grease in the grease chamber to increase and the base oil in the grease to separate from the thickener. As the temperature increases, the pressure in the grease chamber also increases, causing the base oil from the above-mentioned axial flow path to flow through the clearance on the shouldered surface, resulting in delivery to the outer ring raceway surface. As a result, the bearing is actively lubricated by the base oil delivered to the outer ring raceway surface.

3. Experiment Using a Model Test Rig

To investigate the characteristics of base oil delivery from the grease chamber and verify the base oil delivery mechanism described in Sec. 2, the authors used a model in which this base oil delivery mechanism was applied to an angular contact ball bearing with a 100 mm bore diameter. The findings from the experiment are provided below.

3.1 Model test rig and test method

The authors’ model test rig for base oil delivery is schematically illustrated in Fig. 2. The appearance of the test rig is shown in Photo 1. The radial clearance in Fig. 2 is indicated on a radius basis. The properties of the two types of grease developed for machine tool main spindle bearings that were used are summarized in Table 1. Grease was filled into the grease chamber via four grease prefilling holes provided on the outer circumferential surface until the entire space of the grease chamber was filled. The amount of prefilled grease was approximately 47 g.

The following characteristics were tested since they were considered to affect base oil delivery:

- Temperature-dependent variation in the grease chamber
- Size of radial clearance at the end point
- Type of grease used

![Fig. 1 Schematic construction of new lubrication system](image1)

![Fig. 2 Sketch of model test rig for base oil delivery](image2)

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Grease</th>
<th>Grease-B</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thickener</td>
<td>Urea</td>
<td>Urea</td>
</tr>
<tr>
<td>Base oil</td>
<td>PAO</td>
<td>PAO</td>
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<tr>
<td></td>
<td>Polyol ester</td>
<td>Diester</td>
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<td>Amount of thickener wt%</td>
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<td>11.7</td>
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<tr>
<td>Base oil kinetic viscosity mm²/s 40°C</td>
<td>40.6</td>
<td>40.6</td>
</tr>
<tr>
<td>Worked penetration, 60 strokes 60W 25°C</td>
<td>243</td>
<td>280</td>
</tr>
<tr>
<td>Spreading consistency</td>
<td>230</td>
<td>260</td>
</tr>
</tbody>
</table>
3.1.1 Variation in the amount of base oil delivery depending on the heat cycle

Grease-A in Table 1 was used to investigate the base oil delivery due to heat increase. The amount of base oil delivery in a model test rig subjected to repeated temperature increase and decrease (hereinafter referred to as the “heat cycle”) was compared with a model test rig in which the temperature was increased and then maintained that increased. The temperature was automatically controlled by a constant temperature chamber and the pattern of the heat cycle was as illustrated in Fig. 3(a). The temperature variation in each heat cycle (15°C) was equivalent to the difference between the temperature in the grease chamber when the bearing was running (41°C) and when the bearing was at a standstill (room temperature: 26°C).

The results of the amounts of base oil delivery relative to the radial clearance of 0.15 mm are summarized in Fig. 3(b). The model test rig (Photo 1) was set so that the flow path composed of the radial clearance was situated vertically, and the oil content delivered to the clearance end point was removed with a piece of blotting paper once every 50 to 100 hours. The increases in the weight of the pieces of blotting paper were totaled and the resultant sum was taken as the amount of oil delivery.

Immediately after grease prefilling, delivery of not only the base oil but also of the grease itself was observed. This portion was wiped away and was not included in the amount of delivery. The state of base oil delivery at the end point of clearance is shown in Photo 2. When the model test rig was not subject to the heat cycle, no delivery was observed after 100 hours elapsed though a minor delivery occurred at the initial stage of testing. In contrast, when the model test rig was subject to the heat cycle, the delivery continued and the amount of delivered base oil was proportional to the time elapsed. In other words, when the heat cycle is applied to the grease chamber, the base oil in the grease is repeatedly separated from the grease and delivered through the clearance to function as a lubricant.

Next, the effect of temperature differences in each heat cycle on the amount of base oil delivery was studied using the combination of Grease-A and a radial clearance of 0.15 mm. For comparison, the test was performed with temperature differences in each heat cycle of 15°C and 3°C. The heat cycle pattern used for this test is shown in Fig. 4(a), and the result...
of this test is graphically plotted in Fig. 4(b). The amount of base oil delivery is greater with a greater temperature difference in each heat cycle. However, base oil delivery is still possible with a temperature difference of about 3°C.

Thus, the minimum temperature difference necessary for base oil delivery through a clearance seems to be 3°C. Assuming that a machine tool main spindle is continuously run for an extended period virtually without stoppage, the authors’ lubrication system can be applied even though, for example, only periodic temperature variation due to control over the jacket cooling temperature is available.

3.1.2 Size of radial clearance and associated amount of base oil delivery

The authors studied the interrelation between the size of radial clearance at the end point through which the base oil is delivered and the amount of base oil delivery. The sizes of the radial clearances adopted were 0.05 mm, 0.15 mm, 0.2 mm and 0.45 mm. The differences in temperature in the heat cycles, relative to the temperatures on the outer rings of the running bearings (41°C) were 3°C (between 41°C and 38°C) and 15°C (between 41°C and 26°C). The heat cycle patterns shown in Fig. 4(a) are the same as mentioned previously. After 250 test hours, the measured amounts of base oil delivery were graphically plotted in Fig. 5. Grease-B was used in this test and the properties, are summarized in Table 1.

The absolute value of the amount of base oil delivery with a temperature difference of 3°C differs from that with a temperature difference of 15°C. Nevertheless, under each of these different temperature conditions, a smaller radial bearing clearance led to a greater amount of base oil delivery.

3.1.3 Effects of different grease types on the amount of base oil delivery

The authors studied the effect of the type of grease prefilled in the grease chamber on the amount of base oil delivery by using the two types of greases summarized in Table 1. The temperature difference in each adopted heat cycle was 15°C, and the size of the radial clearance used was 0.05 mm. The heat cycle pattern adopted was the same as that described.

![Fig. 4. Relation between temperature difference and base oil delivery](image1)

![Fig. 5. Base oil delivery versus radial gap](image2)

![Fig. 6. Base oil delivery versus grease property](image3)
previously and illustrated in Fig. 3(a). After 250 test hours, the measured amounts of base oil delivered are graphically plotted in Fig. 6.

It can be understood that the amount of base oil delivery varies greatly depending on the type of grease used. Compared with Grease-A, the amount of base oil delivery with Grease-B was approximately 2 times more than Grease-A. In other words, compared with Grease-A, Grease-B is more prone to base oil separation. The choice of a grease that is prone to base oil separation is one important consideration for the authors’ new grease lubrication system.

3.2 Base oil delivery mechanism

As mentioned above, the factors that allow the authors’ novel grease lubrication system to be capable of base oil delivery include the heat cycle, optimized radial clearance and the choice of grease. The authors studied how these factors affect base oil delivery by measuring the pressure in the grease chamber in the base oil delivery mechanism. For the test, the base oil delivery model illustrated in Fig. 2 was used, the pressure in the grease chamber was measured with a distortion gage pressure sensor and the grease temperature was determined with a thermocouple. The sizes of radial clearances adopted were 0.05 mm, 0.15 mm, 0.2 mm and 0.45 mm. Grease-B was prefilled in the grease chamber.

The results of the grease chamber pressure measurements are summarized in Fig. 7(a). In addition, the temperature readings in the grease chamber obtained at the same time using the model with a radial clearance of 0.05 mm are also indicated. In the heat cycle pattern adopted, the lowest temperature was 38°C and the highest temperature was 41°C (temperature difference of 3°C), and the cycle was repeated once every 2 hours. The pressure in the grease chamber kept at 38°C at the start of the test was taken as 0 kPa. Fig. 7(b) is a partial view of the measurement record shown in Fig. 7(a), with the time axis enlarged.

From these diagrams, the following observations, which fulfill expectations, can be made about the grease chamber. First, the pressure increases as the temperature increases and the pressure drops as the temperature drops. Second, the pressure during temperature increase also correlates to the size of the radial clearance at the end point of the grease chamber. The pressure was greatest when the clearance was the smallest at 0.05 mm and decreased with greater clearance. Furthermore, the maximum pressure relative to the clearance of 0.05 mm is approximately 10 kPa. The authors think that the pressure in the grease chamber readily increases with the smaller radial clearance due to better sealing of the interior of the grease chamber. They also conclude that the grease in the grease chamber expands as the temperature increases, causing the base oil and thickener in the grease to separate. Since the fluidity of the separated base oil is higher than the thickener, as the pressure in the grease chamber increases, the separated base oil is pushed out through the clearance. It appears that when delivery of the separated base oil through the clearance begins, the pressure in the grease chamber gradually decreases as shown in Fig. 7(b).

Next, when the temperature of the grease drops, the grease begins to shrink, and as a result, the pressure in the grease chamber drops. While the pressure is dropping, the rate of base oil delivery from the clearance gradually decreases and a portion of the delivered base oil is drawn back into the clearance. Then, the interior of the grease chamber seems to be sealed due to the surface tension of the oil. This theory is further supported by the fact that the minimum pressure value is somewhat negative. 
authors believes that slight increases in the pressure that follow are due to the gradual shrinking of the model itself. This pattern is further repeated when the temperature increases and decreases again due to the repeated heat cycle.

Thus, existence of the base oil delivery mechanism described above has been confirmed.

4. Spindle Test

The authors applied their new grease lubrication system to an angular contact ball bearing that carries a spindle and executed a running test, simulating operation with a machine tool. The bearing used was an angular ball contact bearing with a 100 mm bore diameter. The technical data for the bearing is summarized in Table 2, and a cross-sectional view of the bearing is shown in Fig. 8.

Incidentally, the lubricant feed rate adjustment function provided by the “radial clearance (see Fig. 2)” in Sec. 3 “Experiment Using a Model Test Rig” is borne by the “shouldered surface clearance (see Fig. 1)” with actual bearings such as those used in the authors’ spindle test.

Table 2 Bearing specifications for spindle test

<table>
<thead>
<tr>
<th>Tested bearing</th>
<th>$\phi 100 \times \phi 150 \times 24$ mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contact angle</td>
<td>$20^\circ$</td>
</tr>
<tr>
<td>Rolling element material</td>
<td>Si3N4</td>
</tr>
<tr>
<td>Cage material</td>
<td>Laminated phenol resin</td>
</tr>
<tr>
<td>Prefilled grease in bearing</td>
<td>Prefilled Grease-A, 6 g</td>
</tr>
</tbody>
</table>

4.1 Temperature increase confirmation test

To investigate the operating characteristics of the authors’ new grease lubrication system, they measured the temperature increase on both constant pressure preloaded bearings and definite position preloaded bearings. A critical requirement in order for the grease lubrication system to be used industrially is the prevention of a significant increase in bearing temperature. Fig. 9(a) and 9(b) illustrate spindle test rigs for both preloading systems. In each test rig, two bearings being tested were arranged in a back-to-back duplex configuration (DB set). The test conditions used are summarized in Table 3. Among the test conditions, the axial load value was determined by considering the maximum allowable load for the contact surface pressure of the rolling elements. For the definite position preloading test, the preload was set to the minimum level necessary for practical machine tool operation (0 initial clearance on mounted bearing).

![Fig. 8 Section view of test bearing](image)

Fig. 8 Section view of test bearing

![Fig. 9 Spindle test rig](image)

Fig. 9 Spindle test rig
The results of bearing temperature measurements with the constant pressure preloading system are graphically plotted in Fig. 10(a), and the results with the definite pressure preloading system are provided in Fig. 10(b). With the definite preloading system that keeps the axial load to the bearings at a near-constant level, the test rig was run at \( d_{mn} \) of 2 million (15000 min\(^{-1}\)). As a result, the authors confirmed that the temperature on the bearings at various running speeds were stable and the spindle can operate without any problems.

At the same time, as can be understood from Fig. 10, with the definite position preloading system, the temperatures on the inner ring and outer ring at \( d_{\text{atan}} \) of 1.8 million (14000 min\(^{-1}\)) were 46˚C and 37˚C, respectively, which are roughly the same as in the case of the constant pressure preloading system that runs at the same speed. However, with the definite position preloading system at \( d_{\text{atan}} \) of 2 million (15000 min\(^{-1}\)), the temperatures on the inner ring and outer ring were 53˚C and 43˚C, respectively. These values are higher than those of the constant pressure preloading system. Since the running bearings were overloaded (preloading) under this situation, the authors judged that the bearings were not capable of operating normally under these conditions.

Assuming that the limiting speed at a particular bearing temperature in the definite position preloading system is the same as the bearing speed of the same bearing with the constant pressure preloading system, and based on the test result described above, the limiting speed of the definite preloading system matches \( d_{\text{atan}} \) of 1.8 million (14000 min\(^{-1}\)).

Thus, by using the authors’ new grease lubrication system, it is possible to operate machine tool spindles at higher speed ranges than had been previously possible with air-oil lubrication systems.

### 4.2 Endurance test
The durability of bearings lubricated with the authors’ new grease lubrication system was evaluated. Durability tests were performed with both a definite position preloading system and a constant pressure preloading system. The test rigs used are illustrated in Figs. 9(a) and 9(b). The constitution of the grease chamber and the operating conditions used are summarized in Table 4. The bearing speed of both the constant pressure preloading system and the definite position preloading system were at \( d_{\text{atan}} \) of 2 million (15000 min\(^{-1}\)) to emphasize the running speed. With the definite position preloading system, overloading would occur as judged from the results obtained in the previous Sec. 4.1. Therefore, the initial clearance of the bearing was adjusted so that an appropriate preload (axial load 2.2 kN) was attained at the speed of 2 million \( d_{\text{atan}} \) (15000 min\(^{-1}\)) in the adopted heat cycle pattern, as illustrated in Fig. 11(a), each repeated cycle consists of 50 hours of operation and 1 hour of rest. The temperature difference in each cycle was approximately 15˚C. With the definite position preloading system, the spindle was also tested in both vertical and horizontal positions to...
examine the effect of the attitude of the main spindle on the bearing life. The test results are graphically illustrated in Fig. 11(b). The endurance test results of bearings lubricated with the authors’ new grease lubrication system are presented along with the results for a conventional grease lubricated angular contact ball bearing with seals on both sides that was tested under the same conditions. When operated at \( d_{mn} \) of 2 million, the conventional grease lubricated bearing seized after approximately 500 hours had elapsed.

It is clear that the authors’ new grease lubrication system is effective at providing a sufficient amount of lubricant and an uninterrupted lubricant supply. A certain bearing lubricated with the authors’ new lubrication system has successfully achieved 18000 hours of continuous operation and is still running.

Incidentally, the endurance test results reported here are all obtained from bearings lubricated with Grease-A. Currently, a bearing endurance test is in progress with the definite position preloading system using Grease-B. The bearings being tested in both horizontal and vertical attitudes have already achieved 2000 operating hours and are still operating smoothly.

5. Conclusion

To cope with the mounting needs for higher-speed machine tool main spindles, the authors have developed a unique grease lubrication system that allows the bearings being lubricated to be completely maintenance-free. By using this lubrication system that features heat cycle-driven base oil delivery from the grease chamber, an angular contact ball bearing with a 100 mm bore diameter can run at \( d_{mn} \) of 1.8 million with a definite position preloading system and at \( d_{mn} \) of 2 million with a constant pressure preloading system. In summary, the authors’ new grease lubrication system is a unique lubrication scheme that helps achieve better machine tool performance and mitigate environmental impacts because it possesses a completely maintenance-free operation of the grease lubrication system and is also capable of higher speed operation not possible with any conventional grease lubrication system.

References

1) NTN Precision Rolling Bearings, Cat. No. 2260/E

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