High-Speed Tapered Roller Bearing for Machine Tool Main Spindles

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Tapered roller bearings feature high stiffness and are used to support the main spindles of machine tools, but require new geometrical and air-oil lubrication systems to overcome their difficulty in high-speed operation. Power loss minimization of the bearings are attained by fully lubricating the rib-roller end contacts as well as by reducing lubricant supply to the roller-race contacts to enable high-speed rotation. The mechanism is specified by a ribbed cup, multiple nozzle holes penetrating the rib for air-oil supply and a resin cage riding on outer-ring spacers. Based on this concept NTN has experimentally produced an air-oil lubricated 100 mm bore tapered roller bearing which successfully operates with a maximum \( d_{\text{max}} \) value of 1.25 million which is beyond previously reported records within the category of air-oil lubrication.

1. Introduction

High-speed running performance and rigidity, which are generally required for machine tool main spindles, are factors that greatly depend on the performance of the bearings that support the main spindles. The types of main spindle bearings used include rolling bearings, dynamic pressure and hydrostatic bearings lubricated by oil and air, and magnetic bearings. Rolling bearings are often used for this purpose because they feature an excellent total balance of all the important considerations, including maintainability and cost-effectiveness. When rolling bearings are incorporated into machine tools such as machining centers that are required to run at higher speeds, four rows of high-speed angular contact ball bearings are provided on the main spindle front side. In this arrangement, high-speed operation is possible with a definite position preloading system. However, the rigidity of this configuration is not always high enough, as the main spindles in combined multifunctional machine tools, such as lathe main spindles, need higher rigidity. Therefore, the front-side bearing arrangement on combined multifunctional machine tools is often comprised of a double row cylindrical roller bearing and an axial load carrying angular contact ball bearing.

However, with this spindle-bearing configuration, the contact angle of the axial load carrying angular contact ball bearing exceeds 30˚ and slipping occurs on the bearing due to a gyro-moment when it runs at a higher speed. Thus, the high speed running performance of this configuration is not satisfactory.

Modern machine tools are required to have enhanced machining efficiency which can be achieved by using. Tapered roller bearings which feature very high rigidity in both radial and axial directions. Due to their greater rigidity, tapered roller bearings are employed on certain large lathes and turning centers. However, since they do not excel in the high-speed running performance required of recent machine tools, they are used less frequently. Therefore, NTN attempted to develop a unique tapered roller bearing that not only features the high rigidity associated with tapered roller bearings but also boasts a high-speed performance that is better than a combination of a double row cylindrical roller bearing and an axial load carrying angular contact ball bearing.

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There have been efforts to develop high-speed tapered roller bearings for machine tool main spindles, and an example\(^1\) has been reported in which \(1.25\) million \(d_{mn}\) was achieved with a tapered roller bearing equivalent to the \(32020\) model. In this example, however, a forced circulation lubrication system was employed with a lubricating oil flow rate that exceeds \(4\) liters per minute. Unfortunately, this example is not practical as a replacement for a grease- or air-oil-lubricated combination of a double row cylindrical roller bearing and an axial load carrying angular contact ball bearing because of a greater lubrication cost and a greater power loss from the entire bearing system that includes this lubricating oil circulation system. The authors have conducted research, prototyping and experiments on high-speed application with an air-oil lubricated tapered roller bearing and have achieved an unprecedented \(1.25\) million \(d_{mn}\). This paper reports on this achievement.

### 2. Requirements for Higher Bearing Speed

A challenge that must be addressed to achieve higher speed with a rolling bearing is limiting the temperature increase that results from higher bearing speed. If the temperature on a rolling bearing is excessively high, the preload varies greatly due to thermal expansion, and this in turn further promotes temperature increase. This problem is particularly apparent with definite position preloading systems, which are often used on bearing systems that must have very high rigidity.

Arrangements for inhibiting temperature increases on a rolling bearing include a forced circulation lubrication system and a jet lubrication system in which a large amount of lubricant is fed to a rolling bearing to remove heat from the bearing in an amount greater than the heat generated on the bearing that results from the viscosity of the lubricating oil. However, the authors did not employ either of these arrangements. In the case of air-oil lubrication systems in which small amounts of lubricating oil are fed to the rolling bearing with compressed air, the key to high-speed operation is minimization of the heat occurring from the running rolling bearing.

Now, let us consider the heat generating factor, that is, the torque generation factor for the tapered roller bearing illustrated in Fig. 1. On a tapered roller bearing, the generatrix of the rollers intersects the prolonged line from the inner-outer ring raceway at a point on the rotation axis. Therefore, sliding friction from spinning between rollers and raceway does not occur.\(^2\) Also, since the mode of roller-bearing ring contact is linear contact, differential sliding does not take place and the elastic hysteresis loss is usually small.\(^3\) Given this, typical factors to be considered for a high-speed tapered roller bearing include:

1. **EHL (elasto-hydrostatic lubrication) rolling viscous friction between rollers and the bearing ring**
2. **Sliding friction between the roller large-end face and the rib face**
3. **Sliding friction between the rollers and cage pockets**
4. **Stirring friction with oil and air in the bearing**

First, the EHL (elasto-hydrostatic lubrication) rolling viscous friction between rollers and the bearing ring in \(1\) can be determined with the following expression:\(^4\):

\[
f_{\text{EHL}} = \phi_{\text{EHL}} \cdot \frac{29.2R(GU)^{0.64W^{0.24}}}{a} l \quad (1)
\]

where, \(\phi_{\text{EHL}}\) is an EHL starvation coefficient that represents the degree that the amount of oil is insufficient, and can take a value in the range from 0 to 1; \(\phi\) represents a correction coefficient derived from shear heating generation occurring at the EHL inlet and is determined based on the load, bearing speed and lubricating oil physical properties; \(a\) is the viscosity-pressure coefficient of the lubricating oil used; \(l\) stands for the contact length of the rollers; \(G\), \(U\) and \(W\) are non-dimensional material, velocity and load parameters, respectively, and are uniquely determined by the geometrical shape and running speed of the bearing, the load, and the physical properties of the lubricating oil. Therefore, if the geometrical shape of the bearing, operating conditions and the lubricating oil are given, the rolling viscous friction can be decreased by gradually decreasing \(\phi_{\text{EHL}}\) to zero by allowing starvation to occur whenever possible by
decreasing the oil fed between the rollers and bearing ring while maintaining the EHL oil film. Consequently, at the same time, the shearing heat generation at the inlet decreases and $\phi_s$ becomes smaller.

Next, let us consider the sliding friction occurring between the roller large-end face and the rib face mentioned in 2) above. The contact surface pressure at this area is low, and the fluid lubrication mode occurring there is an equivalent viscosity-rigid body or changing viscosity-elastic body (EHL) mode.5) Thus, the authors executed numerical analysis in the equivalent viscosity-rigid body mode in order to evaluate quantitatively the effect of starvation on the frictional resistance between the roller large-end face and rib face.

More specifically, the authors determined the reaction force on the oil film and the pressure center position from the oil film pressure distribution, which was obtained by solving the Reynolds equation for the two-dimensional flow on a polar coordinate system in accordance with a relaxation method. They repeated calculation with the Newton-Raphson formula until the conditional expression, where the reaction force on the oil film was equal to the applied load and the pressure center position passed the roller center position, was satisfied. The cavitation boundary conditions applied to the Reynolds equation were the Reynolds conditions. From the floatation and skew angle on the roller large-end face finally obtained, the viscosity-friction torque on the rib can be calculated. Accordingly, in Fig. 2, which simulates the interference region in terms of fluid lubrication between the roller large-end face and inner-ring rib, the hatched area is the region subjected to fluid lubrication analysis for a case where no starvation is present ($\theta = \theta_{\text{min}}$). Assuming that the oil film pressure generation starts at the angle $\theta$, the rib starvation coefficient can be defined with the following expression:

$$\phi_{\text{rib}} = \frac{\theta - \theta_{\text{min}}}{\theta_{\text{max}} - \theta_{\text{min}}} \quad (2)$$

When this rib starvation coefficient is used and the rib viscosity-friction torques for all the 29 rollers are totaled, the result illustrated in Fig. 3 is obtained. Since most of the pressure occurs when $\theta$ is $\pi/2$ or lower, the authors assumed that $\phi_{\text{rib}}$ was less than 50%. From Fig. 3, it should be understood that when $\phi_{\text{rib}}$ is at 45%, the torque is approximately twice as great. This is because, despite the narrower pressure generation region (that is, frictional torque generation region), the rib needs to carry the same load, and as a result, the oil film thickness on the rib is smaller and the increase in the torque is greater than the effect of reduction in the pressure generation region. Assuming that in reality the starvation occurs not only in the circumferential direction but also in the radial direction, more increase in the torque can be expected.

In summary, to reduce the rolling viscosity resistance, it is necessary to minimize the amount of lubricating oil. On the other hand, to reduce the frictional resistance on the rib, it is necessary to provide a sufficient amount of lubricating oil. The authors believe that by reducing the amount of oil on the raceway surface, the torque resulting from 3) and 4) described above can also be reduced.

To determine the actual relation between amounts of air-oil lubrication and torques, the authors used the standard 32020 tapered roller bearing, which has specifications as summarized in Table 1, and measured the total torque and rib torque. As usual, lubricating oil was supplied from the back side (inner-ring smaller rib side) by using eccentric pump action.
The resultant measurements are graphically plotted in Fig. 4.

In Fig. 4 the difference between the total torque and rib torque equals the sum of 1), 3) and 4) described above. Since the effect of 4) is sufficiently small in the relatively low speed range of less than 4000 min⁻¹ and 1) is predominant over 3) with roller bearings, the difference between the total torque and rib torque is essentially equivalent to 1), that is, the EHL rolling viscosity friction on the raceway surface. As can be understood from expression (1), when there is no effect of starvation and heat generation, the rolling viscosity resistance increases in proportion of the 0.65th power of the rotation speed. However, the total torque starts to decrease at around 300 min⁻¹ when the effect of heat generation seems to be minimal. This appears to be due to the significant effect of starvation.

Also, it will be understood that as the amount of oil is reduced, the effect of starvation is more apparent and the rolling viscosity resistance decreases. At the same time, because the rib is under fluid lubrication conditions, the rib torque is proportional to the bearing speed, and increases with a decreased amount of lubricating oil. This fact further supports the analysis results described above.

In summary, in order for a tapered roller bearing to be able to run at a higher speed, it is necessary to invent a mechanism that minimizes the supply of lubricating oil to the raceway surface while supplying plenty of lubricating oil to the rib.

3. Structure and Performance of the Authors’ Bearing

3.1 Basic structure

Based on the approach described above, the authors developed and evaluated various prototype bearings with unique lubrication schemes and internal constructions. Consequently, the authors finally developed the unique tapered roller bearing with the construction illustrated in Fig. 5. Now, referring to Fig. 5, let us describe its features. First, the authors’ tapered roller bearing lacks an ordinary inner-ring rib, and instead, the rib is situated on the outer-ring side. This arrangement prevents the lubricating oil from flying away due to the centrifugal force of the rotating inner-ring as well as resultant insufficiency in the amount of lubricating oil. At the same time, to effectively lubricate the rib, multiple direct-injection nozzles are provided on the rib surface that supply air-oil lubricating oil. Incidentally, the usual oil supply from the back side is not provided. Instead, the lubricating oil from the direct-injection nozzles is directed to the raceway surface through the spaces between the rollers. Because the pockets and rollers interfere with each other at higher speeds on roller guiding cages such as the standard type, the authors have adopted

![Fig. 5 Section view of developed tapered roller bearing](image)
a guide arrangement on which the outer-ring rib guides the rollers. In addition, the authors have also adopted a lightweight oil-retaining phenol resin as the cage material. The rollers used are made of a lightweight ceramic material of a lower thermal expansion coefficient.

3.2 Test and bearing performance

The construction of the test spindle is illustrated in Fig. 6. The spindle was driven by an external inverter-regulated motor that was connected to the spindle by a coupling. Among the bearings used, only the authors’ bearing was arranged in a two-row back-to-back duplex configuration. The boundary dimensions of the authors’ bearing were the same as those of the 32020, with a bore diameter of 100 mm. Since the width of the pocket bars is greater with the authors’ bearing than the pressed steel cages on standard bearings, the number of rollers on the authors’ bearing is 23 which is less than standard bearings, which have 29. The housing incorporated an arrangement for jacket cooling. The preloading system used was a definite position preloading system with zero initial clearance. The major test conditions used are as summarized in Table 2.

In testing the authors’ bearing, the temperatures and torques on the inner and outer rings relative to various spindle (inner ring) speeds were measured. For this purpose, the spindle running speed was set to each intended speed setting by controlling the inverter-regulated motor, and the temperatures and torques on the inner and outer rings were measured and recorded only when they became stable.

The torque was measured with a torque meter mounted to the spindle. Each torque measurement includes the torques on the two pieces of the authors’ bearing and the windage on the rotating sections. The air-oil feed rate to each piece of the authors’ bearing was 1.8 cc/h.

The measured temperatures on the inner and outer rings are plotted in Fig. 7(a), and the obtained torques in Fig. 7(b). Note that the torque values in Fig. 7(b) are those converted for one bearing. As can be understood from these diagrams, the authors’ bearing does not exhibit a rapid increase in temperature and torque up to an inner ring speed of 1000 min⁻¹, and can attain 1.25 million daN with air-oil lubrication.

In addition, the authors checked the running speed of the cage at various bearing speeds using a stroboscope. As a result, no slipping was found,

![Fig. 6 Schematic construction of test spindle](image)

![Table 2 Specifications of spindle test bearing and test conditions](image)

### Table 2 Specifications of spindle test bearing and test conditions

<table>
<thead>
<tr>
<th>Bearing</th>
<th>Description: Equivalent to 32020XUP4 (modified rib construction)</th>
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<tbody>
<tr>
<td>Size: ID 100 x OD 150 x 34 mm total width</td>
<td>Contact angle: 17°</td>
</tr>
<tr>
<td>Rollers: 23 Si3N4 rollers</td>
<td>Cage: Phenol resin machined cage, outer-ring guide</td>
</tr>
<tr>
<td>Bearing clearance: Definite position preloading with a clearance of zero as assembled</td>
<td>Bearing lubrication: Air-oil</td>
</tr>
<tr>
<td>Oil supply only to the rib surface</td>
<td>Nozzle dia. φ0.6 x 12 (one bearing row)</td>
</tr>
<tr>
<td>ISO VG32 oil used</td>
<td>0.01 cc/20 s oil feed rate (one bearing row)</td>
</tr>
<tr>
<td>Jacket cooling oil temperature: Room temperature ±1°C</td>
<td></td>
</tr>
</tbody>
</table>

![Fig. 7 Test results of developed tapered roller bearing](image)

![Fig. 7(a) Measured temperature of inner and outer ring](image)

![Fig. 7(b) Measured torque per bearing](image)
thereby it was confirmed that the rollers were rolling correctly. The authors also inspected the bearing elements that underwent the test and detected no damage such as discoloration and wear. Incidentally, the authors did not attempt to use a low viscosity lubricating oil for the purpose of preventing disruption of the oil film on the raceway surface that could result from a decrease in the amount of oil. Instead, they adopted ISO VG 32, a standard lubricating oil type commonly used for machine tools. However, the authors think that use of a lower-viscosity lubricating oil could further limit the increase in torque and temperature.

The authors also tested the standard 32020 bearing, and learned that even when a maximum bearing speed was attained by varying the viscosity and feed rate of lubricating oil, the bearing temperature dramatically increased at 6000 min⁻¹, and further operation was impossible.

As previously mentioned in the first section of this report, the authors’ bearing is expected to satisfy requirements for not only high-speed performance but also rigidity. Therefore, the authors prepared two bearing arrangements, wherein one bearing arrangement was comprised of the authors’ bearing type and the other bearing arrangement consisted of a double row cylindrical roller bearing (which was used as a front-side bearing on a conventional bearing arrangement for machine tool spindles) and an axial-load carrying angular contact ball bearing. Then, assuming that these bearing arrangements were each incorporated into a machine tool spindle, the authors calculated the deflections of the spindles that carried the loads, and compared the calculation results. The arrangement of the authors’ bearing is illustrated in Fig. 8(a), and the arrangement consisting of a double row cylindrical roller bearing and an axial-load carrying angular contact ball bearing is shown in Fig. 8(b). The specifications for the authors’ bearing are identical to those summarized in Table 2.

The double row cylindrical roller bearing used was a standard NN3020, the axial-load carrying angular
contact ball bearing used was a standard HTA020ADB, and the bearing used on the rear side was the N1016 single row cylindrical roller bearing.

Deflections occurring on the front end of the spindle when pure axial loads and pure radial loads are applied to the front end are plotted in Figs. 9(a) and (b). For example, when an axial load of 4 kN is applied to the front end, the axial deflection on the authors’ bearing is approximately 50% less than with the conventional bearing arrangement. When a radial load of 4kN is applied to the front end, the radial deflection on the authors’ bearing is approximately 80% less than the conventional bearing arrangement even though the front side bearing span with the authors’ tapered roller bearing is approximately 30% smaller than the conventional bearing arrangement. More specifically, it has been confirmed that the authors’ bearing has high rigidity against both axial and radial loads.

As mentioned in the first section of this report, the high-speed rotation performance of the axial-load carrying angular contact ball bearing within the conventional bearing arrangement shown in Fig. 8(b) is insufficient, and this bearing arrangement is capable of only 1 million \( d_{\text{MN}} \) with air-oil lubrication.

In contrast, in terms of both high-speed performance and rigidity, a spindle supported by the authors’ tapered roller bearing excels over spindles supported by a conventional bearing arrangement that consists of a double row cylindrical roller bearing and an axial-load carrying angular contact ball bearing.

4. Conclusion

By using their unique tapered roller bearing that boasts greater rigidity for machine tool main spindles, the authors have invented a novel bearing construction and a novel air-oil lubrication mechanism to address the issue of unsatisfactory high-speed performance, which is a drawback of conventional tapered roller bearings. As a result of testing the authors’ tapered roller bearing with an inner bore diameter of 100 mm, this bearing achieved unprecedentedly high \( d_{\text{MN}} \) of 1.25 million with air-oil lubrication. For bearings that support machine tool main spindles, higher running speed and greater rigidity are challenges to be addressed in achieving better surface finish quality for workpieces and improved machining efficiency. NTN will remain committed to addressing these technical challenges by pursuing further advances in its bearing technology.

References

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