1. Introduction

Contact type seals may be used in bearings for automotive transmissions to reduce the impact on bearing life caused by hard foreign objects penetrating the bearings, such as wear debris from the gears in the transmission. However, since these seals come in contact with the bearing inner ring, frictional torque during rotation may increase. Furthermore, in applications where high-speed rotation is required, such as recent EVs and HEVs, the use of contact type seals is difficult due to the restricting seal peripheral speed limits. Although further torque reduction is a requirement for bearings used in automotive transmissions, since torque reduction and sealing performance are conflicting properties, it has been difficult to achieve significant torque reduction while ensuring a high level of seal performance.

To solve this challenge, NTN has developed a low friction ball bearing seal for transmission applications that uses arc-shaped (half-cylindrical shaped) micro bumps on the sliding surface of the seal lip. This generates a lubrication oil film between the seal lip and inner ring by a wedge effect in the bump area, thereby achieving low friction. This article reports the mechanism of fluid film formation and friction resistance by soft EHL (ElastoHydrodynamic Lubrication) analysis.

2. Soft EHL analysis

2.1 Analysis model and program

Fig. 1 shows the developed seal. The figure on the left shows the cross section of the bearing with the seal and the figure on the right shows the cut out view of the seal along the circumference. Arc-shaped (half-cylindrical shaped) micro bumps in equal distance on the seal sliding surface contact the bearing inner ring. The bumps are curved in the axial direction as well, providing point contact with the inner ring sliding surface when they are installed in the bearing as

A low friction seal for use in ball bearings developed for automotive transmissions has arc-shaped (half-cylindrical shaped) micro bumps on the sliding surface of the seal lip. This generates a lubrication oil film between the seal lip and inner ring by a wedge effect in the bump area, thereby achieving low friction. This article reports the mechanism of fluid film formation and friction resistance by soft EHL (ElastoHydrodynamic Lubrication) analysis.
shown in Fig. 2. As shown in Fig. 3, at the bumps, the
distance from the sliding surface gradually decreases
in the circumferential direction, namely, direction of the
flow, which, while rotating, produces fluid pressure in
the lubricating oil on the sliding surface in a gradient,
forming an oil film between the seal sliding surface
and the inner ring. When enough oil film is formed on
the sliding surface, it creates a fluid lubrication
condition and frictional torque is expected to decrease
compared with the conventional seal without these
bumps. The thickness of the oil film between the seal
lip and bearing inner ring, and seal frictional torque
due to fluid viscosity resistance are calculated by the
soft EHL analysis coupling the oil film pressure and
seal deformation.

In general, when oil film analysis of rolling bearings
is performed, semi-infinite elastic body approximation
is used in structure analysis for calculating shapes
since the elastic deformation is small. However, when
rubber seals are analyzed, semi-infinite elastic body
approximation does not work as the elastic
deformation is large; therefore, non-linear structure
analysis is required. Changes of the lubricating oil
properties due to pressure are not considered, since
the increase of pressure in the oil is small because the
seal deforms more easily due to the increased
pressure. Seal deformation, including that of the
bumps, is calculated using general purpose non-linear
analysis software with the finite element method, and
the oil film pressure is calculated using the solution of
the following Reynolds equation that describes
viscous fluid motion on the lubricating surface.

\[
\frac{\partial}{\partial x} \left( \frac{\rho h^3}{12 \eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{\rho h^3}{12 \eta} \frac{\partial p}{\partial y} \right) = \omega_0 \frac{\partial (\rho h)}{\partial x}
\]

Where, \( x \): coordinate in the flow (circumferential)
direction, \( y \): coordinate in the direction perpendicular
to flow (rotational axis), \( \rho \): density, \( h \): oil film
thickness, \( \eta \): lubricating oil viscosity, \( p \): pressure, \( u_0 \):
velocity on the sliding surface. In addition, flow rate in
the \( x \) direction \( u \) is a function of the coordinate in the
direction of the film thickness \( z \) and can be expressed
by the following equation.

\[
u = -\frac{1}{2\eta} \frac{\partial p}{\partial x} z (h-z) \left( 1 - \frac{z}{h} \right) \omega_0
\]

The first term of the right side is the Poiseuille flow
induced by the difference of pressure and the second
term of the right side is the Couette flow induced by
the motion of the wall surface. For the cavitation
condition, Swift-Steiber’s condition was used. In
addition, a periodic boundary condition was used as
this is an axisymmetric problem.

By repeating structure analysis and fluid analysis, a
convergent solution was obtained. Since the oil film
pressure significantly changes with a small change of
oil film thickness, and the seal deformation is
significantly affected by a small change of pressure,
convergence properties may be degraded by strong
non-linearity in the coupled analysis. By appropriately
controlling the relaxation amount according to the
convergence condition in an iterative computation
when the Reynolds equation is solved with the
relaxation method, we have enabled calculation with
good convergence properties, as shown in Fig. 4. The
calculation was performed in the following two steps:

**First step:** The condition of the bearing with inner
ring, outer ring and seal is calculated with structure
analysis only. This is equivalent to the condition of the
seal while not rotating.

**Second step:** Using the calculation result of the first
step as the initial condition, coupled analysis with the fluid is performed. This is equivalent to the conditions when the seal is rotating. The initial oil film thickness is set by exploring the inner ring position where the load capacity and tension force due to oil film pressure are balanced, with the seal shape as calculated in the first step (not rotating). In the first half of this step, inner ring position is returned to the predetermined position.

In the calculation of torque, a fluid lubrication condition was assumed with the seal lip sliding surface and the inner ring well separated by the oil film and only torque due to viscosity resistance of the lubricating oil considered.

2.2 Conditions and results of calculation

The properties and operating conditions of the developed seal under review are shown in Table 1. The rubber rigidity and lubricating oil viscosity were defined as a function of temperature considering thermal expansion of the rubber. In this article, impact from surface roughness, stress relaxation of rubber, pressure to lip seal by oil flow within the transmission, shaft run-out, etc. are not considered. Fig. 5 and 6 show an example of the calculation results. Fig. 5 shows oil film pressure distribution against the seal lip sliding surface, while the diagram above shows the picture of the seal lip sliding surface viewed from the inner ring side. Oil film pressure is shown at the bump. Fig. 6 shows oil film thickness and pressure distribution along with the flow at the minimum film thickness position. Approximately 0.22 MPa of maximum pressure was observed. The minimum oil film thickness was about 1 μm. The composite roughness

<table>
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<th>Table 1 Properties and conditions</th>
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<tr>
<td>Seal material</td>
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<tr>
<td>Bearing inner ring OD (mm)</td>
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<tr>
<td>Bump radius (mm)</td>
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<tr>
<td>Bump height (μm)</td>
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<tr>
<td>Number of bumps</td>
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<td>Lubricating oil dynamic viscosity (40°C) (mm²/s)</td>
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<td>Rotational speed (min⁻¹)</td>
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of sliding surface is about Rq 0.22 \( \mu m \). Sufficient oil film is formed on the surface compared to the roughness, creating a fluid lubrication condition. **Fig. 7** shows the relationship between temperature and minimum film thickness. The oil film thickness is reduced as the temperature rises; however, the minimum film thickness is about 1 \( \mu m \) even at 150˚C, which is sufficient compared to the roughness. **Fig. 8** shows the relationship between temperature and seal torque. The indicated torque comes from two seals used in one bearing. In addition, factors making up the total torque are categorized into the following three groups and color coded: torque caused by the Poiseuille flow at the bumps, torque caused by the Couette flow at the bumps, and torque caused by the Couette flow at the non-bump area, which is the sliding surface between the bumps. Since the increase in pressure at the non-bump area is small, torque caused by Poiseuille flow at the non-bump area is also small and not shown. The torque decreases as the temperature rises, to approximately 0.007 Nm at 120˚C. The main factor of the torque is the Couette flow at the bump area. **Fig. 9** shows the relationship between the number of bumps and torque. When bumps are fewer, torque due to the Couette flow at the non-bump area increases. The reason for this is that the area of the sliding surface at the non-bump area increases and, in addition, the gap of the sliding surface of non-bump area decreases. When the sliding surface gap decreases, the shear resistance and seal torque increase. When there are fewer bumps, the distance between the bumps increases, and the sliding surface of the non-bump area undergoes elastic deformation due to the tension force, moving it closer to the inner ring sliding surface and reducing the gap between the sliding surfaces. **Fig. 10** shows a schematic diagram of the non-bump area. **Fig. 11** shows the relationship between the number of bumps and the gaps between bumps under non-operating conditions. In the figure, the gap in the area between bumps is shown as a ratio against the
height of the bumps. In this study, when the number of bumps is around 100 or less, the gap in the area between bumps becomes 0, which means that the sliding surface of the non-bump area comes in contact with the inner ring. However, during operation, an oil film is formed by the pressure gradient created when the non-bump area is close to the inner ring, resulting in no direct contact between the surfaces.

When the number of bumps increases, torque due to the Couette flow at the bump area increases. The more bumps, the lower the tension force per bump; therefore, the minimum film thickness of the bump area increases and the torque increases, because the total area of sliding surface of the bumps increases. Under the conditions of this study, the lowest torque result occurred when the number of bumps was around 100.

3. Verification test

3.1 Oil film thickness

The oil film thickness at the seal bump area was measured using the infrared microspectroscopic method. The infrared spectroscopic method is a method of structural analysis and quantification of substances that uses infrared rays applied to the test pieces and analyzes the spectrum obtained from transmitted or reflected light. When infrared rays are transmitted through substances, light in a certain frequency range is absorbed by the substance and the intensity of light through the substance is weakened. The intensity of light through the substance and the light path length are correlated. Fig. 12 shows the schematic diagram of the measurement device. Since acrylic rubber absorbs infrared rays, 0.3 μm of gold is vacuum deposited on the surface of the acrylic rubber so that the light is reflected on the rubber surface. The test piece, which simulates a seal bump, is pushed into the rotational sapphire disc and infrared rays are irradiated to the disc from the opposite side. Oil film thickness can be estimated by measuring the intensity of light reflected on the bump surface, after it is transmitted through the oil film. The pressure of the rubber bump to the disc is controlled by the Z-stage. In this experiment, the pressure and speed are set as parameters.

The measurement results and calculation results from the soft EHL analysis are compared in Fig. 13. Both results show the same trend and approximate magnitude, verifying an oil film thickness of around 1 μm. However, the measured value is larger than the analysis results in the range where the oil film is thinner. The reason for this difference may be the impact from the measuring range and vibration of the sapphire disc surface. The analysis shows the minimum film thickness; however, the measurement result is the average of a 30 x 30 μm area; therefore, the measurement value of the oil film thickness may have been larger. Also, there was approximately 0.35 μm of disc surface vibration due to the rotation, which may have resulted in a larger measurement value of oil film thickness.

3.2 Bearing torque

The bearing torque was measured by installing the developed seal on a bearing designed for use in a transmission. The specification of the seal and the test conditions were the same as shown in Table 1. The lubricating method was an oil bath with the upper surface of the lowest rolling element at the oil level. In addition, 754 N of radial load was applied to the bearing. This measurement result and measurement results of bearings using the conventional contact type seal and non-contact type seal without bumps are shown in Fig. 14.

When compared to the bearing using the conventional contact type seals (without bumps), the torque of the new seal with bumps is reduced by 80% or more. Conventional contact type seals are considered to operate in mixed lubrication or fluid
lubrication conditions, including solid contact; however, since the formed oil film is very thin due to micro EHL (elastohydrodynamic lubrication) from the rough surface, shear resistance is large even if the seal is in a fluid lubrication condition. On the other hand, the developed seal forms sufficient oil film at the bumps creating a fluid lubrication condition, which is assumed to be one of the contributing factors for the reduction of friction torque. In addition, as shown in Fig. 9, when the number of bumps is not too small, since the torque at the non-bump area due to the Couette flow is small compared to the torque due to the Couette flow at the bumps, the distance of sliding surfaces in the developed seal is increased, which should also contribute to the reduction of friction torque.

The increase of torque compared to a bearing with the conventional non-contact type seal is around 0.01 Nm. Since the seal torque of non-contact type seal is estimated to be very small, the seal torque of the developed seal is estimated as around 0.01 Nm, which is equivalent to the calculation results shown in Fig. 8 and 9.

4. Conclusion

We have developed an oil film analysis program for investigating the mechanism of oil film formation and friction resistance of a newly developed seal. This seal generates oil formation between the sliding surfaces of the seal lip and bearing inner ring due to the arc-shaped (half-cylindrical shaped) micro bumps set in equal distances on the seal lip toward the bearing inner ring sliding surface. Within the range of conditions evaluated in this article, the developed seal facilitated oil film formation due to the fluid lubrication effect of the bumps and produced lower torque than conventional contact type seals.

Although not mentioned in this article, interference, bump height, and variation of rubber properties also impact bearing torque. With the developed program, it is possible to predict torque of the newly developed seal and optimize the specification of the bumps.

References

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