4 点接触玉軸受の滑りと動力損失の解析

Analysis of Sliding and Power Losses in Four-point Contact Ball Bearing

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1. Introduction

Due to requirements for higher electricity/fuel efficiency, electric powertrains in BEVs and HEVs need to be more efficient, compact and lightweight. These electric powertrains reduce the components compared to conventional MTs, ATs and CVTs, and rolling bearing losses also have a greater impact. Therefore, rolling bearings are also required to propose smaller and lower-loss designs. This trend has led to the use of 4-point contact ball bearings (single / double row) in some powertrains, aiming to reduce size by replacing deep groove ball bearing and to reduce losses by replacing tapered roller bearing.[1)]

Generally, 4-point contact ball bearing is recommended in the catalogues of rolling bearing manufacturers to be used under conditions where axial load is dominant (2-point contact condition), but in electric powertrains there are often conditions where radial load is dominant. However, there are almost no reports showing sliding and loss predictions under such conditions. In this report, the sliding and losses of 4-point contact ball bearing are analysed using a model with 6 DOFs on balls. It also presents a comparison with a common deep groove ball bearing with 2 point contact.

2. Overview of Calculation Method

Generally, displacements and loads of each ball in a ball bearing are calculated by giving only 2 DOFs of in-plane displacements x and y. In this method, displacements and stiffness of balls can be solved accurately, but it is difficult to know the conditions at the contact area between balls and raceways because the rotation and orbital speed of balls are calculated using the outer/inner ring control assumption. Therefore, L. Houpert et al. developed a ball bearing model in which balls are given 6 DOFs (ball displacements in the plane x, y, ball rotation axis α_r, ball rotation speed ω_r, gyroscopic speed ω_{gyro} and ball orbital speed ω_{orb}) as shown in Fig. 1.[2)] In this model, the forces between balls and outer/inner rings are considered as sliding friction at contact ellipses, hydrodynamic rolling force/pressure, elastic rolling resistance, centrifugal forces, gyroscopic moments and contact normal forces.

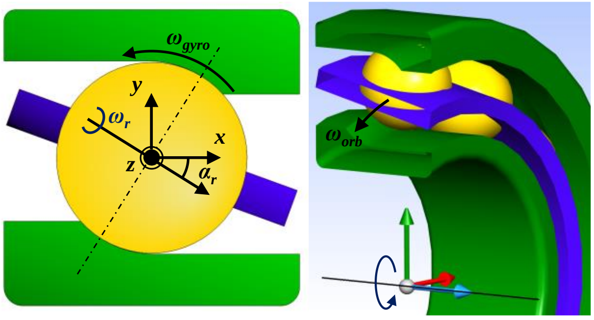

Fig. 1　6 DOFs of Ball

In this report, a dynamic analysis study of 4-point contact ball bearing is conducted using this ball bearing model, focusing on sliding and power losses on the contact ellipse between balls and outer/inner rings. (Note . L.Houpert et al.'s ball bearing model also includes a cage, which is not included here.)

3. Power Loss Formulas for Ball Bearing

Without inclusion of a cage, total power loss P^T of a ball bearing is obtained by summing up the components occurring between inner ring-balls and outer ring-balls using equation (1).

$$P^T = \sum(P^S + P^{MER} + P^{FR}) \tag{1}$$

Loss P^S due to ball sliding is obtained by integrating ball sliding velocity Δu_z and force dF^S on each slice along contact ellipse a and using equation (2).

$$P^S = a_i\left(\int_{-1}^{1}\Delta u_{z_i} dF^S_{z_i} + \int_{-1}^{1}\Delta u_{x_i} dF^S_{x_i}\right) + a_o\left(\int_{-1}^{1}\Delta u_{z_o} dF^S_{z_o} + \int_{-1}^{1}\Delta u_{x_o} dF^S_{x_o}\right) \tag{2}$$

Loss P^{MER} is calculated by elastic rolling resistance M^{ER} (hysteresis) using equation (3). (where $\omega_{x,y}$: ball rotation velocity)

$$P^{MER} = \left(M^{ER}_{x_i} + M^{ER}_{x_o}\right)\omega_x + \left(M^{ER}_{y_i} + M^{ER}_{y_o}\right)\omega_y \tag{3}$$

Loss P^{FR} due to hydrodynamic rolling force F^R (due to oil film at inlet) is obtained from equation (4). (where $\bar{u}_z$: mean entrainment speed)

$$P^{FR} = F^R_i \bar{u}_{z_i} + F^R_o \bar{u}_{z_o} \tag{4}$$

Table 1　Internal Geometry of Analysis Bearing

Type		4 Point Contact	Deep Groove (6206)
PCD		46.0mm	
Ball Dia.		3/8 inch (9.5250mm)	
Contact Angle		20°	0°
Raceway Curvature	I.R.	52%	51%
	O.R.	53%	52.25%
Radial Clearance		12.5 μm	

4. Bearing Specifications and Calculation Conditions

The design of analytical bearings is shown in Table 1. The deep groove ball is 6206 and 4-point contact ball is designed with the same dimensions and the same number of balls/PCD. The loads and speeds to be analysed are shown in Table 2, with a temperature of 60 °C (lubricant kinematic viscosity 14.97 cSt). Case 1 applied only pure radial load, Case 2 added an axial load of 0.25 times of radial load, Case 3 added an axial load of 0.5 times of radial load, and Case 4 also added an axial load of 0.75 times of radial load. In addition, the relative tilt of outer/inner rings was set to 0. The calculations were performed using software MASTA from SMART MANUFACTURING TECHNOLOGY, which uses the calculation model developed by L. Houpert et al.

Table 2　Load and Speed for Analysis

	Radial Load	Axial Load	Speed
Case 1	2000N	0N	2000rpm
Case 2		500N	
Case 3		1000N	
Case 4		1500N	

5. Power Loss Results from Dynamic Analysis

5.1 Total Power Loss Results

Comparison of the total power loss of deep groove ball and 4-point contact ball of the same size is shown in Fig. 2. The loss of the 4-point contact ball is more than twice as large as the deep groove ball under conditions with high radial load (Cases 1 and 2), but the loss becomes smaller as the axial load ratio increases, and the loss is smaller than the deep groove ball in conditions such as Case 4, with 2-points contact. Also, it was observed that the loss fluctuations of 4-point contact ball tended to increase in conditions where 2-point and 4-point contacts transitioned during rolling, as in Case 2 and Case 3, compared to Case 1 and Case 4.

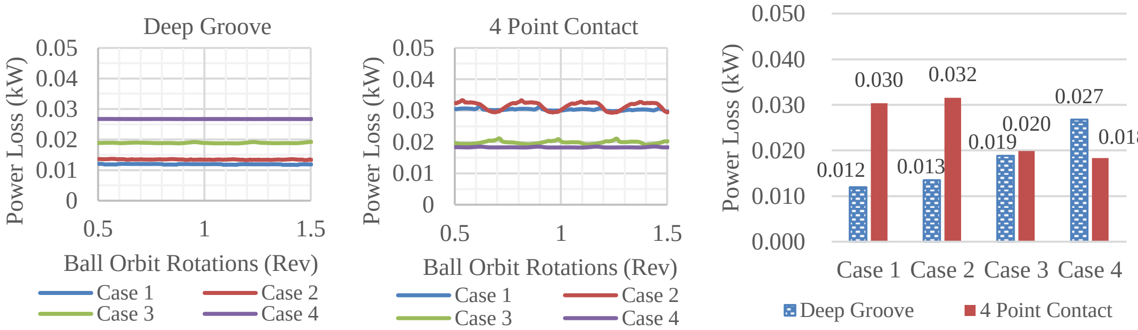

Fig. 2　Power Loss Comparison between DGBB and 4pt Ball

5.2 Results for Loss components (per Ball)

The loss components per ball at maximum are shown in Fig. 3, focusing on Case 2, where the loss of 4-point contact ball is the highest and the loss fluctuation is also large. The loss P_{fz} due to sliding due to the rolling direction accounts for a major part of the loss for both deep groove ball and 4-point contact ball, and this difference is almost directly equivalent to the difference in the total loss between the two. Fig. 4 shows the variation of each loss component for one ball orbital rotation. The 4-point contact ball was found to have a larger P_{fz} than the deep-groove ball, even though the ball load was smaller than the deep-groove ball, which was caused by a higher sliding velocity on the contact ellipse. The contact with the smaller contact load showed unstable behavior in the P_{fz} during ball entry. (Where, P_{fz} is the component due to sliding of P^S in rolling direction and P_{fx} is the component due to sliding of P^S in axial direction.)

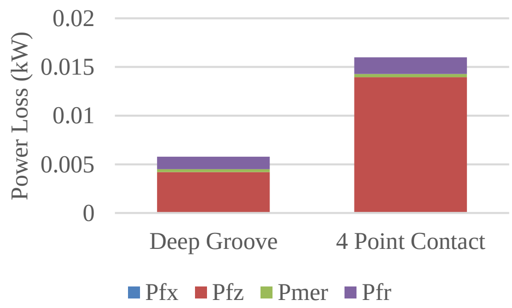

Fig. 3　Comparison of Loss Components at Max (Case 2)

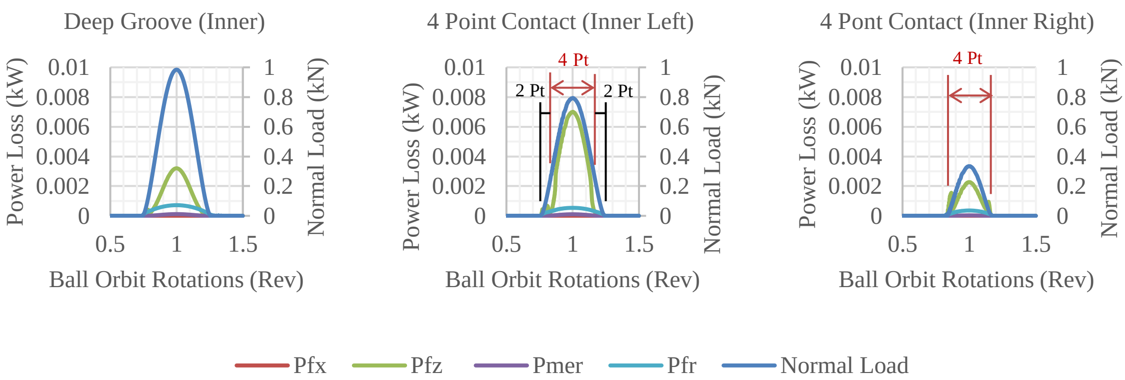

Fig. 4　Variation of Loss Components for 1 Ball Orbital Revolution (Case 2)

6. Conclusion

Dynamic analysis using the ball bearing model with 6 DOFs showed that, in conditions of relatively low rotational speed, the power loss of 4-point contact ball bearing is larger than deep groove ball bearing due to the higher sliding speed on the contact ellipse, including the variation over rotation. This kind of analysis will be important in the future to capture phenomena that are difficult to monitor, and to use it to determine whether or not a bearing can be used.

References

1) Kawai：Double Row Four Point Contact Ball Bearing, NACHI TECHNICAL REPORT vol.34B4, (2018)
2) L. Houpert, J. Clarke & C. Penny : Tribological Models for Advanced Ball Bearing Simulation, Tribology Transactions, (2023), DOI: 10.1080/10402004.2023.2213470