Physical and Numerical Investigation of the Friction Behavior of Graphite Lubricated Axial Ball Bearings

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1. Introduction

Even though graphite is known for centuries as lubricant, not much information about its applicability in technical systems is available in regular mechanical construction literature (cf. [1]). Especially at high temperatures, graphite has high potential for the lubrication due to the carbon bond energy.

Rolling bearings are being used in many mechanical systems. Temperature limits of regular (considered here to be without additives) used oils or greases are at approx. 100°C. In context of the carbon bond energy, graphite has potential for the lubrication of rolling bearings in high temperature environments. However, before analyzing rolling bearings at high temperature, the friction behavior at regular conditions has to be investigated.

For this reason, the first aim of this investigation is the experimental analysis of the friction behavior of graphite lubricated rolling bearings. For the transfer of these results to simulation models, a first approach for describing graphite lubrication in numeric simulation models is presented. The further aim of the project is the preparation of a demonstrator bearing, which includes a relubrication unit.

2. Physical testing of axial ball bearings

2.1 Method

Axial ball bearings of type 51208 have been chosen for testing. The lubrication has been done by graphite dispersion, which has been sprayed on the bearings before and during the test runs. For the initial lubrication, the bearings were disassembled and the rings and cage (with balls) have been coated with graphite dispersion with an airbrush spray gun. The graphite forms a matt black surface on the rings and balls, the thinner immediately evaporates.

The tests have been conducted on an IPEK rotational test bench, which was extended with a relubrication unit. This unit consists of mechanical clamps for its fixation in the test chamber and a commercial airbrush gun.

Within this test setup, the inner bearing ring is driven. Above the bearing arranged sensors have measured the axial force and the torque, which has been transferred from the inner (driven) to the outer ring. The axial force has been applied by a pneumatic cylinder on top of the measure chamber.

For testing, the bearings have been driven with rotational velocities of 250 rpm ($\hat{=}\ 6.75\ m/s$), 375 rpm ($\hat{=}\ 10.125\ m/s$) and 500 rpm ($\hat{=}\ 13.5\ m/s$). Axial loads for the comparable Hertzian pressures between ring and one ball of 1 GPa ($C/P = 47$), 1.5 GPa ($C/P = 14$) and 2 GPa ($C/P = 6$) have been applied. Eight specimens have been tested, each one at a constant velocity but with cyclic changing loads. This means, that with one bearing the first applied pressure has been 1 GPa, followed by 1.5 GPa and 2 GPa. This cycle is repeated two times for gaining nine test runs each 15 min per bearing. The evaluation has been done by the computed coefficient of friction $\mu = M_{measured}/(F_{axial}\ r_{Bearing})$

2.2 Results

Before placing the bearings in the test chamber, the rings and cage (including balls) have been reassembled. Immediately, much of the graphite spalls from the surfaces, which contact the mutual parts. Thus, not much of the graphite stays on the surfaces but still small amounts of graphite are kept in the roughness valleys of the race. On the flat ring areas next to the races, the thickness measuring of the graphite coating reveals a layer thickness of approx. 75 µm. The spalling of the graphite shows the necessity of relubrication and that the thickness can be neglected.

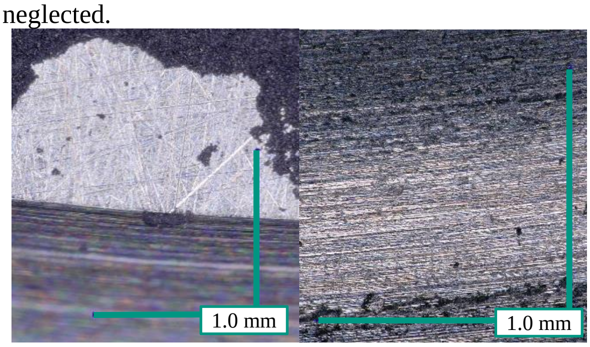

Figure 1.The left picture shows the spalled graphite on the ring-shaped area next to the race of the inner ring. The race is focused on the right picture with scattered graphite spots.

The test runs have been conducted for eight bearings with each nine tests. So, in total 72 time series are available for the evaluation. For each test run, the average coefficient of friction is computed and plotted by its pressure as box plot in Figure *2*. The median value (red line) decreases with the pressure. So, by increasing the pressure, less rotational resistance occurs in the bearings. The values show, that for graphite lubrication in axial ball bearings coefficients of friction of approx. 0.005 can be expected. This is about three times more than for regular lubrication.

So, graphite is capable of lubricating ball bearings, but not at the same reduction of the resistance than regular oil and grease lubrication. Regular lubricated bearings have coefficients of friction of approx. 0.0015 for $C/P = 10$ [2].

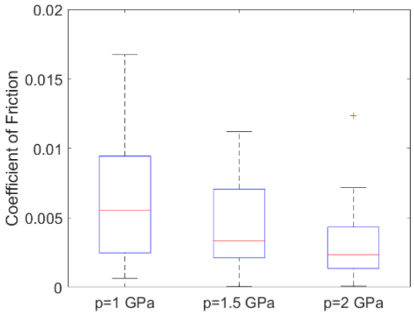

Figure 2. The three boxes show the resulting coefficients of friction per pressure velocity.

3. Computational approach for the depiction of graphite lubrication

3.1 Computational approach

Multi body simulations (MBS) are being used in bearing development processes because e. g. the system geometry can be easily replaced. Hereby, a modelling and computational approach for the depiction of a contact is required. For the relation between the normal and shear forces in contacts, the coefficient of friction is a suitable parameter. However, being a system dependent variable, the determination of it as single value is not suitable. Further approaches are necessary to depict the tribological behavior. For regular oil and grease lubrication, computation formulas are available for the determination of the frictional torque in bearings, which split the torque to a load dependent and load independent term [2]. But for the description of solid graphite lubrication in rolling bearings, no approach for multi-body simulations is available.

In contrast to oil and grease, graphite has no viscous material properties. Therefore, the split of the computation of the bearing resistance torque with regarding the viscosity (cf. [2]) seems unsuitable. However, since graphite sliding experiments show [3] that graphite is shifted out by sliding motions, an approach, which splits sliding and rolling friction is presented.

The application of the friction computation is applied in a multi-body model of the axial ball bearing 51208. For reducing the computational time, the model consists of the two rings and three balls. The cage is modelled as spring-damper systems between the balls.

The frictional force in each of the six contacts is computed with the Amontons-Coulomb friction law

$$F_{fric} = \mu_{MBS} F_{normal}$$

The coefficient of friction is split in a sliding, rolling and rotating term

$$\mu_{MBS} = \mu_{slide} + \mu_{roll} + \mu_{rot}$$

The value for μ_{slide} calculates as a share of the literature values [3] of sliding graphite friction. The share is computed by the slide-to-roll ratio (SRR).

$$\mu_{slide} = \mu_{slide,lit} \cdot k_{SRR}$$

The hereby presented simulation, conducted in MSC Adams, results are conducted without rolling or rotating resistance ($\mu_{roll} = \mu_{rot} = 0$). Since the SRR is hereby the main parameter, three initial simulations with Hertzian pressures of 1 GPa and SRRs of 1 %, 2 % and 10 % are considered.

3.2 Numerical results

Table 1. In the MBS, the coefficient of friction rises with an increasing SRR.

SRR	1 %	2 %	10 %
μ_{MBS}	0.0023	0.0024	0.0056

In Table 1, the results of the MBS illustrate that the results are in the same scale as the physically measured coefficients. Since the SRR are assumed values and the rolling and rotating resistance are neglected, further analysis is necessary to create the entire computational model for the depiction of graphite lubrication in MB software.

4. Conclusion and Outlook

The results from the physical experiments have indicated a relation of the coefficient of friction and the applied pressure. A clear relation between the rotational velocity and the coefficient of friction has not been found. The coefficients of friction have shown that graphite has potential for the fluid free lubrication of rolling bearings. However, the lubrication is dependent on the relubrication since the graphite does not get to the contact on its own or due to system behavior. A relubrication mechanism is necessary to secure the lubricant supply.

The results from the early stage computational model for the description of graphite lubrication in rolling bearings on base of a sliding coefficient of friction have illustrated right scale values, which requires extensive elaboration.

Since especially the SRR is estimated to strongly influence the coefficient of friction in solid lubricant systems, solutions for measuring it will be analyzed with the aim of extending the test bench. Further, the variation of the amount of graphite for lubrication will show the durability of a lubrication and what amounts of graphite would be necessary for a lifetime lubrication.

5. Acknowledgment

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