Graphite Lubrication in Axial Ball Bearings and a Description Approach for Lubrication Mechanisms

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Abstract: Oil and grease are the two most used lubricants in mechanical engineering. Besides these two, solid lubricants are known as good lubricants, even though they are hardly used in industrial applications. To gather knowledge about the frictional behavior of graphite in axial ball bearings and to transfer this knowledge to different fields of application, the authors present an description approach for lubrication regimes and experimental results of graphite as lubricant in axial ball bearings.
The first part of this publication considers C&C²-Models for the description of the friction regimes known from the Stribeck-Curve. With these models, solid lubrication is described and different system states e. g. running-dry can be closely analyzed. The use of this enables engineers to analyze their systems and to assess the effects of a considered lubrication.
In the experimental part of this publication, axial ball bearings (type 51208) are tested in an IPEK system tribometer for testing under practice-oriented conditions. During the runs, the bearings are lubricated with graphite dispersion with the aim of analyzing the influence of different amounts of the lubricant on their frictional behavior. The frictional behavior of the bearings is evaluated with the coefficient of friction, which is computed as the quotient of the measured torque and measured axial force multiplied with the bearing radius. The results of the investigations are coefficient of friction curves, which correlate with the amount of sprayed in graphite dispersion and the frequency of the relubrication.

Introduction

In general, oil and grease are the most used lubricants for rolling bearings. These two lubricants are well investigated and much knowledge and experience are available in regular literature [1–3]. Furthermore, bearings are thoroughly understood as well as the surroundings. The latter are e. g. sealings, which cope well with these two lubricants. However, in high temperature environments, the function of oil or grease lubricated bearings is jeopardized. Other environments can e. g. require to conduct or insulate electric current as in train collectors or electric motor brushes. For the function fulfillment, lubricants require special additives. Besides just adding further additives to lubricants, the change of the base lubricant itself is possible, but today hardly considered by engineers. Due to missing knowledge and experience with non-oil or -grease lubricants, solid lubricants are rarely used. Further, a look into regular design literature [1, 3, 4] reveals that hardly any information about solid lubricants is available. Hence, this publication aims for the analyzation of graphite as lubricant for ball bearings.

State of the Art

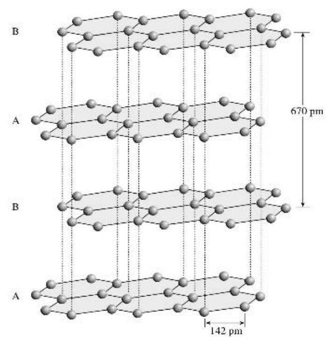

Figure 1. Graphite structure [6]

The most known application for graphite is the pencil. In this system, the most important function is the transfer of graphite as wear to the paper for creating a black or dark gray line. This requires that the bonded graphite can be easily abraded and sticks to the paper surface. The amount of transferred graphite is determined by the hardness of the pencil, which is the result of the graphite-clay relation in the graphite lead. Besides the added clay, the manufacturing process and further additives can change the hardness [5]. Next to the visible line on the paper, the haptic feeling of the writing is an important factor. This haptic feeling is created by the ratio between clay and graphite, which also determines the abrasive wear. Hence, the graphite is first hold together in the lead, then with the shear load continuously abraded and in the end sticks to the paper. The use of graphite in pencils depict, that its lubrication properties are known for decades.

Bragg [6] has been one of the first researcher, who has analyzed graphite as chemical material. Graphite comprises out of layers of ring-shaped carbon atoms (Figure 1). Bragg has explained the lubrication effect by the sliding of the graphite layers against each other. However, the lubrication effect is not only justified by this arrangement since the bonding forces between the layers are large [7]. Besides the layered structure of graphite, the lubrication effect requires further molecules in-between to reduce the bonding energy and effect the low friction [7, 8].

After the basics of graphite have been reported, the analysis of the friction behavior in laboratory tests is the next step in the order of tribological testing [1]. Morstein, Dienwiebel present in [9] the results of micro-tribometer friction tests with graphite coated plates. They present that with a low layer thickness, a coefficient of friction of approx. 0.1 are achieved and with increasing layer thickness the coefficient of friction increases to about 0.5 due to ploughing effects. A further result shows, that the friction is reduced with a larger normal force in the contact. Therefore, the known frictional effect of graphite is verified for sliding contacts.

Graphite is one representative out of the group of solid lubricants. PTFE, better known as Teflon, has been analyzed by e. g. Nguyen, Kamga, Gedan-Smolka, Sauer, Emrich, Kopnarski, Voit in [10, 11] . In a block-on-ring test rig, coefficients of friction of approx. 0.28 – 0.57 are measured in dependence to the exact PTFE-compound mixture. MoS_x as solid lubricant has been investigated by Tillmann, Wittig, Stangier, Thomann, Moldenhauer, Debus, Aurich, Brümmer [12]. They observe the same qualitative relation between the increase of the load and the decrease of the coefficient of friction as Morstein, Dienwiebel [9] report. As this short overview reveals, solid lubricants are getting analyzed. However, these lubricants are not even remotely as understood as usual oil and grease lubricants are. Hence, different research approaches are applied for analyzing on different scales. Whereas, on lower scales the layer thickness is analyzed, on larger scales the lubrication effect on e. g. bearings is relevant. The connection of the results of different scales to develop industrial applications is hardly conducted yet. Thus, this article presents a descriptive model for graphite lubrication with the aim of transfer graphite as lubricant to industrially used machine elements.

Aim of this Research

As shown, some basic knowledge about graphite lubrication is available in literature, but the transfer of this knowledge to industrial applications is hardly done. To close this gap, C&C²-Models are used to describe the lubrication regimes and to apply the gained knowledge in the product engineering of tribological construction elements. Secondly, to show the effect of graphite as lubricant, regular axial ball bearings are lubricated by graphite dispersion and the quantitative friction effects are presented. With this knowledge in mind, the C&C²-Approach is used to facilitate the transfer of the system behavior to different technical applications and to support the synthesis of new lubricants based on graphite. Hence, not just knowledge about graphite as lubricant in bearings is gained, but furthermore an approach for evaluating different lubrication concepts in early product engineering states.

Description Approach for Lubrication Regimes

The Stribeck-curve classifies the state of a tribological system to different categories. Here on micro scale, the different states are described with a C&C²-Model (see [13, 14] for additional information about this description model). This description model aims for a better understanding of the embodiment function relation (EFR) in a system. With the use of key elements, such as working surface pairs (WSP), channel support structures (CSS) and connectors, load or information paths are identified in a system. Functions and properties are then assigned to these key elements. By this, a system is closely analyzed and the functional relations described. This approach is not just valid for mechanical designs, but further applicable to the four regular friction states solid, boundary, mixed and fluid friction. The models base on firstly presented models by Lorentz [15], Behrendt [16] and Blust [17]. Here, these models are renewed with increased knowledge about the C&C²-Approach. The models are templates for engineers to describe their individual systems (such as clutches or brakes) and to assess the influencing factors on their tribological system.

The first friction regime in the Stribeck-curve is solid friction. Figure 2 illustrates an exemplary contact on micro scale of two solid bodies. In solid friction, both bodies contact each other predominantly at the asperities. The two connectors (C) are the connection of the load path to the environment. From the lower connector, the load runs through CSS1 to WSP 1 at which both bodies contact each other. After the contact, the load runs further to the upper connector through CSS2. If solid 2 moves, friction arises in the WSPs dependent on its current embodiment. By summing up the individual friction of each WSP, the global friction can be computed. The properties of the WSP are the result of the material and the surface topography. Hence, in inhomogeneous materials (e. g. clutch pads) the properties can differ a lot.

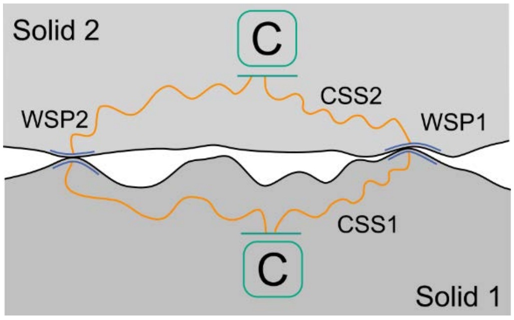

Figure 2. C&C²-Model dry friction following [15], [16] and [17].

The second mechanism is boundary friction, in which the surfaces are partially covered with lubricant molecules. WSP with lubricated surfaces and without lubricated surfaces (solid friction) occurs. Therefore, the global contact friction summarizes out of different kinds of WSP and is therefore difficult to describe.

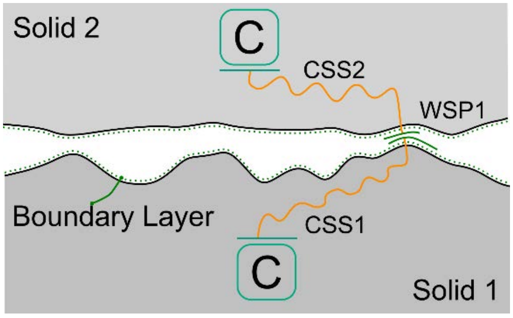

Figure 3. C&C²-Model boundary friction following [15], [16] and [17].

Mixed friction is the friction mechanism where multiple contact mechanisms occur at once. As illustrated in Figure 4 solid friction occurs in the left contact and boundary friction occurs in the right contact. Concentrating on the middle contact, two WSPs (3 and 4) form between the solid surfaces and the lubricant (yellow). Hence, the load is partially carried by a solid-solid contact, a boundary friction contact and at last by the lubricant. This interaction of the different friction states results in the complicated description of mixed lubrication in mechanical systems. The last friction mechanism is fluid friction. Both solids are separated by the lubricant. The resulting friction between both bodies is the friction, which occurs in the lubricant itself, and which effects between the surfaces and the lubricant.

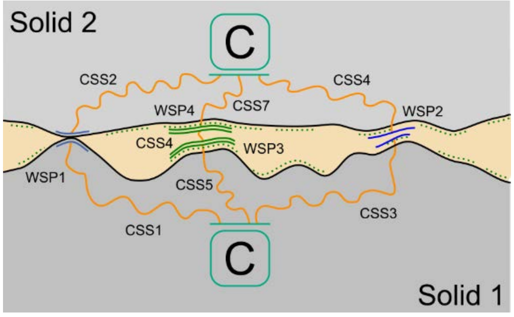

Figure 4. C&C²-Model mixed friction following [15], [16] and [17].

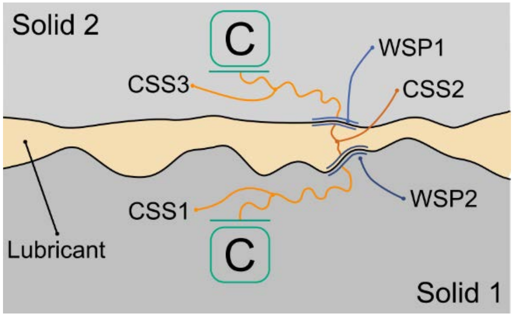

Figure 5. C&C²-Model fluid friction following [15], [16] and [17].

These four presented models are templates to describe individual tribological systems. With this help, engineers can assess the influence of the material or the surface roughness on the tribological system behavior of their individual system. The close observation and abstraction of their system support the understanding of complex tribological systems.

The application of graphite as lubricant in tribological applications requires its testing in tribological design elements. So, these tests cover one step in the categories of tribological testing. Next steps will be conducted on a system tribometer, which allows to depict real system conditions. Here, axial ball bearings of type 51208 have been selected because these bearing are easily demountable and the contact area is good reachable. The tests have been conducted on an IPEK test configuration for rotational movements and continuous axial forces (see [18–20]). The test conditions were velocities of 250 rpm, 375 rpm and 500 rpm. Axial loads have been defined to get 1 GPa (C/P=47), 1.5 GPa (C/P=14) and 2 GPa (C/P=6) contact pressure between balls and race. The evaluation of the test has been done by the coefficient of friction computed as $\mu = M_{measured}/(F_{axial}\, r_{Bearing})$.

Figure 6. Validation environment for testing graphite lubricated axial ball bearings

The aim of the investigation is the analyzation of the friction behavior of graphite lubricated axial ball bearings. The tests comprise different amounts of graphite, different pressures and varying velocities to measure the influence of pure graphite as lubricant (cf. clay in pencils). For this reason, graphite dispersion [21] has been chosen as lubricant, which has ethanol as thinner, because it quickly evaporates and graphite does not dissolve in it. An initial lubrication turned out not to last for more than a few minutes, so a relubrication unit has been designed and fixed in the test configuration. The relubrication unit (Figure 7) combines a commercial airbrush pistole with a pneumatic cylinder for actuating. measure the amount of graphite and control its relubrication time a flow sensor is used. The nozzle targets on the small gap between cage and inner ring. Hence, not the entire sprayed graphite gets into the contact but some graphite sticks at the outer surfaces of cage and ring. Some graphite is lost in the test chamber.

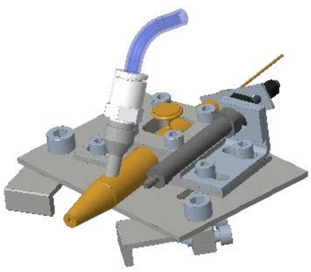

Figure 7. Relubrication unit

The first tests focus on the quantitative lubrication effect of pure graphite in axial ball bearings. Hence, the above mentioned three velocities and pressures were tested with a constant relubrication interval and a constant relubrication amount of graphite dispersion.

The relubrication interval describes the time between two airbrush sprays with graphite dispersion. The relubrication amount defines the amount, which is sprayed on the bearing.

Figure 8. Coated bearings, inner ring, cage and balls and outer ring

Before testing, the bearings are initially coated (Figure 8) to overcome possible solid friction contacts at the beginning of a run. Even during this coating process, the thinner immediately evaporates and only graphite remains. To further exclude a lubrication effect by the thinner, pure ethanol hast been tested and showed no lubrication effect at all. The results without lubrication and with ethanol have been pretty much similar. [22]

Friction Behavior of Graphite Lubricated Axial Ball Bearings

The first tests have been conducted with constant relubrication interval (1 min) and constant lubrication amount (0.4 ml per spray). Both values were experimentally chosen by pre testing. Figure 9 shows the curves of three chosen test runs with 375 rpm and 1.0, 1.5 and 2.0 GPa. The first petrol colored curve is at 1.0 GPa. The serrated course results from the graphite spraying. After each spraying, the transferred torque (and hence the depicted coefficient of friction) drops and increases again. The second, blue curve depicts the coefficient of friction at 1.5 GPa contact pressure. With ~0.006, the resulting coefficient of friction is lower than at 1.0 GPa but higher as 2.0 GPa. Furthermore, the height of the drops after each spraying are reducing with increasing pressure.

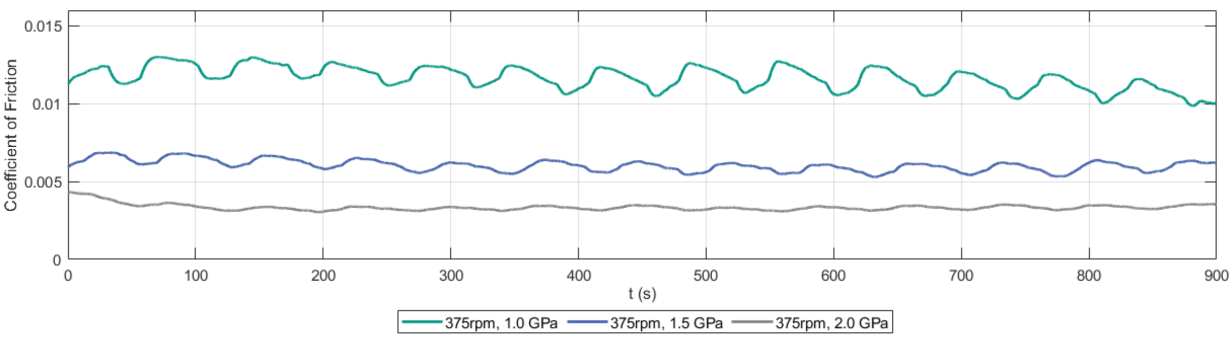

Figure 9. Coefficient of friction curves at 375 rpm and 1.0, 1.5 and 2.0 GPa contact pressure.

The entire tests have been conducted for eight bearings at the three velocities and pressures. The left box plot in Figure 10 depicts that with an increase of the rotational velocity no clear effect or trend in the coefficient of friction is remarkable. Whereas on the right side an effect of the pressure is clearly visible. The median coefficient of friction decreases from 0.006 to 0.003 with increasing pressure.

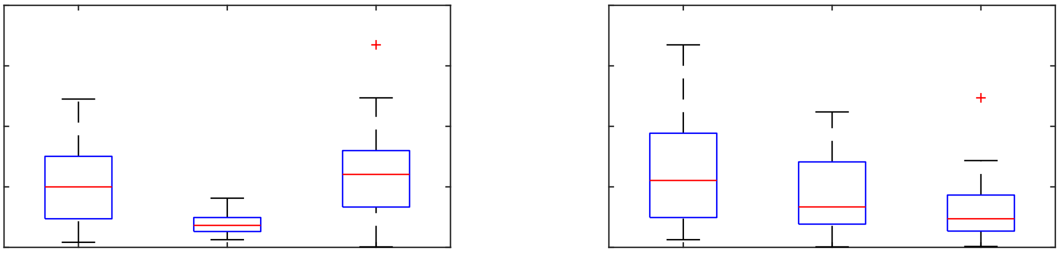

Figure 10. Box plots with varying rotational speed and contact pressure.

Further bearings of the same type were tested with constant velocity and pressure but varying relubrication interval and varying relubrication amount. The interval has been changed in steps of 60 s, 180 s and 300 s. The amount has been increased from 0.1 ml, 0.2 ml and 0.4 ml.

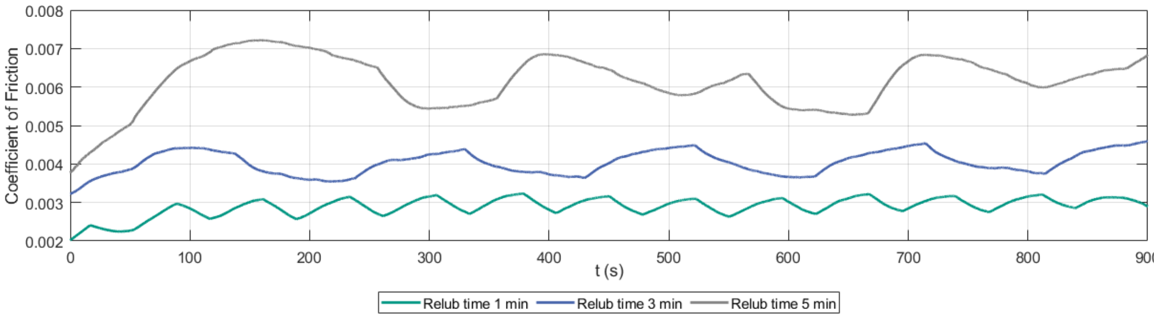

Figure 11. Coefficient of friction for different relubrication intervals.

The curves of the coefficient of friction for varying relubrication intervals are displayed in Figure 11. The velocity is set constant to 375 rpm and the contact pressure is set to 1 GPa. With a relubrication of 0.1 ml each 5 min, 0.0062 results as median coefficient of friction. With 180 s interval time, the mean coefficient of friction is 0.004 and with 1 min 0.0029. The spikes of the serrated curves give the moment of lubrication. After the lubrication took place, the bearing runs with the in-sprayed graphite for the interval time.

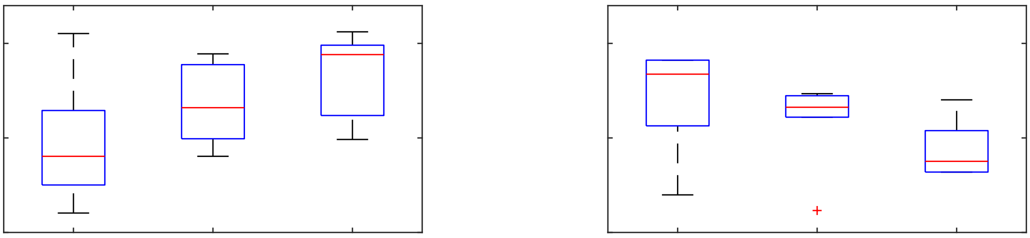

Figure 12. Box plots for the relubrication interval and amount

The tests have been evaluated with box plots in Figure 10. In the left figure, the combined results of two bearings and 25 test runs are depicted. With an increase of the relubrication time, the median coefficient of friction increases from below 0.005 to nearly 0.01. On the right side, the same relation is visible. With an increase of sprayed-in graphite, the coefficient of friction decreases from about 0.01 to light below 0.005.

Figure 13 presents a picture taken with a digital microscope of the race surface after the tests. The races of the bearings are partially covered with graphite. The gray spots are remaining graphite on the surface and the bright shiny spots show the metal surface of the race.
The friction behavior reveals that with increasing time, the graphite is carried and rubbed out of the contact and solid-solid contacts occur. Thereby, the increasing coefficient of friction is explained. After spraying graphite, the contact is lubricated and as Figure 13 illustrates, graphite-solid and solid-solid contact occurs. A lubrication effect of the ethanol is considered unlikely, since even during the initial coating, the ethanol evaporates within seconds. So, in the environment of the heated-up bearing, the ethanol evaporates even faster. Tests with in-situ optical evaluation are planned to verify this assumption (cf. [23]).

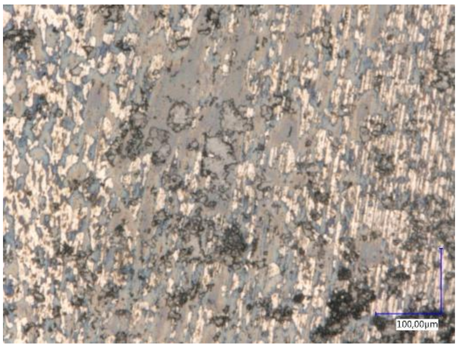

Figure 13. Bearing race after test run

For graphite lubrication the C&C²-Model in Figure 14 is suggested. The roughness valleys are partially filled with graphite and graphite-solid and solid-solid contacts occur. A complete separation of both surfaces is unlikely since the graphite is deformed and rubbed out of the contacts. In the process of running dry, some graphite sticks to the surfaces of both solids, but most graphite is rubbed out and solid-solid contact behavior occurs. The friction inside the graphite (CSS2) is then explained with the experimental results from Morstein and Dienwiebel [9].

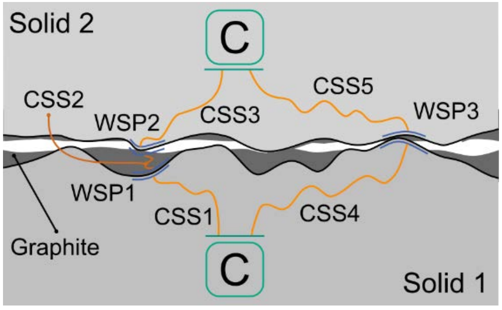

Figure 14. C&C²-Model solid friction

Discussion and Outlook

In the first part of this article, a description approach for tribological systems is presented. The description with the C&C²-Model helps engineers to analyze different types of contacts. The approach and the presented templates for different tribological contacts support engineers in getting a precise system understanding. The contact itself is closely described by the WSPs and the CSSs. From a system view, the contact is here one element in the entire system model (of e. g. a bearing or a clutch) and helps to dismantle a system to its sub-systems. So, the subsystem contact can be closely described and thoroughly understood.
As a result of the experiments, we have seen that the handling of graphite dispersion and the auxiliary equipment have been challenging. Hence, the application of airbrush lubrication in industrial environments is not recommended. However, pure graphite has shown its quantitative performance for the lubrication of bearings. For this reason, in the next step, the graphite dispersion is swapped against solid graphite for achieving life-time lubrication. However, this means additional bonding material will probably be necessary, which must be thoroughly understood before the use as clay in pencils shows.

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