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EFFECT OF COVERAGE OF GRAPHITE ON SELF-LUBRICATING PLAIN BEARINGS

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Abstract: *Self-lubricating plain bearings have the same role as rolling bearings, but they are operating at a different principle. For self-lubricating sliding bearings mutual mobility of parts in contact and load transfer is provided by slip, i.e. sleeve sliding on a bushing. The separation of the sleeve and the bushing, to prevent their direct contact and wear, is achieved by graphite lubrication. Separation of the sleeve and the bushing, i.e. prevention of their direct contact and wear is achieved by graphite. The basic role of the graphite is to reduce resistance to relative motion caused by friction. As coefficient of friction defines the friction resistance to motion a research has been defined to experimentally determine the friction coefficient of self-lubricating bearings with 20 and 30 % coverage of graphite. The experimental research showed that the increase of the percentage of graphite coverage reduces friction.*

Keywords: *self-lubricating bearing, lubrication, graphite, coverage, friction, experimental testing.*

1. INTRODUCTION

The basic characteristic of self-lubricating bearings is that they do not require special maintenance and lubrication during operation. The term self-lubricating graphite explained by the fact that graphite crystals require a very small tangential force to move. Graphite is capable of forming oriented coating on the surface of metals, which replaces the lubricant. That determines the application areas: in places where lubricants are ineffective, or in locations exposed to high or low temperatures, in a reactive environment, where fluids for lubrication cannot be used because of the possibility of contamination of the product or the environment.

The operating life of machines and different mechanisms essentially depends on the proper selection and application of lubricants. Wrong lubricant, or a wrong application of the lubricant, will cause abnormal and accelerated wear of surfaces that are in contact as well as the occurrence of permanent damage [1]. The main criteria for selection of lubricant are specific operating conditions of slide bearings: sliding speed, load, temperature, etc. Graphite can be defined as a material that reduces friction, but also meets some very specific and strict set requirements. It has self-lubricating properties, but also a cooling capacity, corrosion protection and thermal stability. Without these qualities graphite would not be able to meet the complex requirements of sliding bearings.

Friction as the most common physical phenomenon which causes a lot of negative effects in production systems and in principle the attempts are always made to minimize friction. One of the main parameters that define the friction process between shaft and bushings is the coefficient of friction. The determination of friction coefficient is achieved using experimental methods [2,3].

As graphite is a brittle material it cannot sustain high radial forces so it is usually combined with the bronze in order to increase load capacity and thermal conductance. In bronze bushings with graphite inserts, graphite forms a thin film on both the contact surfaces that is highly resistant to shocks and which retains position between them. The percentage of graphite cover in bushing was analyzed by Liu et al. [4] as well as the effect of composite materials using powder metallurgy. The effect of graphite on tribological (friction) properties were studied using UMT – 2MT TRIBO meters. It was determined what average friction coefficient, the maximum coefficient of friction, the range magnitude of the friction coefficient and wear resistance can be achieved with the change of graphite content up to 3 %. Responding graphite content and hardness are the two most important factors in achieving good quality lubricating film on the worn surfaces. When the bearings operate in presence of high temperature, graphite is added to maintain good sliding properties [5].

As graphite reduces stiffness of the bearings the bearings should have a simple configuration. New materials, particularly composites, are considered very important materials for combination with graphite in order to improve tribological properties [6]. In order to investigate the physical and tribological properties of new materials (nano lubricants), the coefficient of friction and temperature were determined by pin-on-disc tribometer [7]. The results show that the additives of graphite nanoparticles improve the lubricating properties. The usual bearing material (Babbitt and bronze) lead to high rates of wear and higher friction coefficient. Today, a lot of companies use PTFE (polytetrafluoroethylene) bearings in design as the material for self-lubricating bearings, which tends to reduce the coefficient of friction. The problem of high wear with PTFE can be reduced by adding graphite fillers. In comparative analysis of the three composites (PTFE, PTFE + 25 % C + PTFE and 35 % C) it has been shown that the characteristics of PTFE can be improved by adding the different contents of the graphite filler and that the best results are obtained with a carbon content of 35 % [8,9]. Dry sliding wear and tear behavior of two abrasive bodies made of graphite epoxy reinforced composite was also studied [10]. The results show that in situations of dry sliding with the increase of load and sliding speed wear the wear also increases. Excellent wear characteristics are obtained with graphite containing carbon-epoxy as filler.

2. THE FRICTION IN SLIDING PAIR

Contact of two surfaces of machine parts in relative motion represents a sliding pair. The shaft sliding sleeve on bearing creates friction. Friction always occurs when the relative motion of the body is existing. Mathematically, the friction can be represented as:

$$F_{\mathrm{tr}} = \mu \cdot N \,, \tag{1}$$

where: F_{tr} – friction force; N – normal force; μ – friction coefficient.

To determine the friction coefficient in this paper the Coulomb's equation (2) was used, which associate load and friction coefficient as a time depended functional ratio of friction force and load:

$$F_{\mu}(t) = \mu(t) \cdot F_{\mathrm{n}}(t) \rightarrow \mu(t) = \frac{F_{\mu}(t)}{F_{\mathrm{n}}(t)} \,. \tag{2}$$

As already noted, due to the relative movement of the shaft in relation to the self-lubricating bearing there is a frictional force. Moment of friction, which is a function of time, is the minimum value of torque that causes rotation of the shaft, which overcomes the frictional resistance of the contact. It can be determined by measuring the force at some distance from the rotational axis (arm's length) of the bearing as the moment of friction is product of measured force and the arm's length, i.e.:

$$M_{tr}(t) = F_n(t) \cdot L . \tag{3}$$

From the condition of equality of friction torque in contact of bushing and shafts and torque that is calculated by equation (3), one can easily determine the friction force if the distances are known:

$$F_\mu \cdot 20 = F_n \cdot 150 . \tag{4}$$

where: F_{tr} – friction force; 20 – bushing radius; F_n – measured force at arm's length; 150 – arm length.

By rearranging of equation (4) the coefficient of friction can then be calculated according to the following formula:

$$\mu = 7.5 \cdot \frac{F_\mu}{F_n} . \tag{5}$$

3. EXPERIMENT

Experimental studies were conducted to determine the effect of the graphite percentage of coverage on the value of friction coefficient. The self-lubricating sliding bearing, with dimensions 50 / 40 × 40 mm, made of CuSn12 with graphite lamella inserts with a diameter of 10 mm were used for experimental study. The number of inserts was varied in order to achieve the graphite surface taking up 20 and 30 % of the inner bushing surface (Fig. 1). The bearing was prepared for experiment by running in with special graphite grease NLGI 2 that is used for running a bearing. For the purpose of experimental investigation a special custom tribometer was developed.

Lathe Potisje Ada PA-C30 was used as a working machine. The measuring system consisted of acquisition device NI cDAQ-9178, torque transducer HBM T1 (100 Nm), force transducer HBM U2 (500 N) and force transducer HBM U9 (5 kN). During the test, the temperature was 25 °C. Figure 2 shows a schematic of measuring configuration and f the developed tribometer. The main rotary motion is provided by a workmachine – lathe (SG). The faceplate is connected with input shaft (V) over torque transducer (M1) and connectors (S1). On the sleeve of the shaft an self-lubricating bearing (SL) is mounted which is radial and axial fixed in upper (GN) and lower (DN) parts of the support. Through the opening of the upper support (GN), via the adapter for the force transducer (D), the radial load is introduced by tightening the screw. The intensity of the force is measured by force transducer (M2). Circumferential force, which represents the frictional force of the bearing, measured by means of a arm (P1 – the length of 150 mm), which is arranged perpendicular to the axis of the bearing and support GN. The transducer S1 resides on the support GN support GN. Noted transducer measures the intensity of the circumferential force. All transducers were connected with computerized unit (R) which performs acquisition of data.

Figure 1. (a) self-lubricating bearing with 20 % coverage after the testing and (b) 30 % graphite coverage after sliding before the test

The experimental research was performed for a value of the sliding speed which correspond to RPM 265 min^{-1} of the shaft and for radial force of 3000 N.

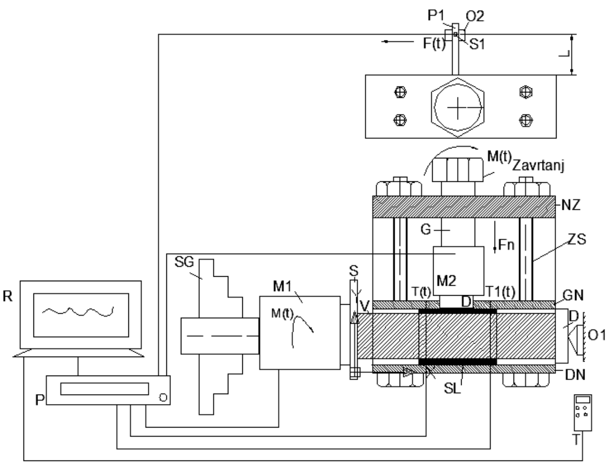

Figure 2. Scheme of the measurement configuration

4. RESULTS

The results of experimental research are presented on Figures 3 to 5. The actual value of radial force in experiment is shown on Figure 3. It can be concluded from the noted figure that a radial force had a constant value of F_r = 3069.9 N during the experimental research. Furthermore, it can be observed that the average value of friction coefficient for a 20 % coverage bushing is μ = 0.014 (Fig. 4). The oscillations of the friction coefficient shown on Figure 4 are a consequence of vibrations induced by a driving machine – lathe. Figure 5 shows the values of the friction coefficient for a 30 % coverage bushing. The average value is μ = 0.014 and the oscillations of the values are again a consequence of the vibrations induced by a driving machine.

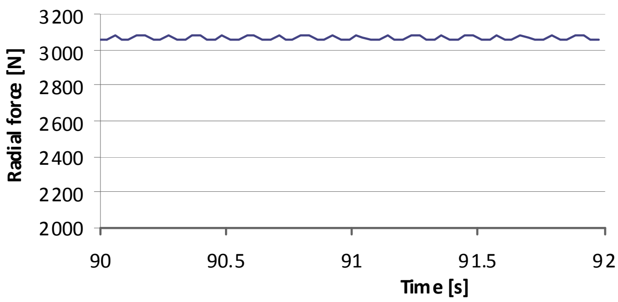

Figure 3. Actual radial force

From the presented results it can be noticed that, for same sliding speeds and radial force, the friction coefficient decreases with increasing coverage of graphite from 20 to 30 %.

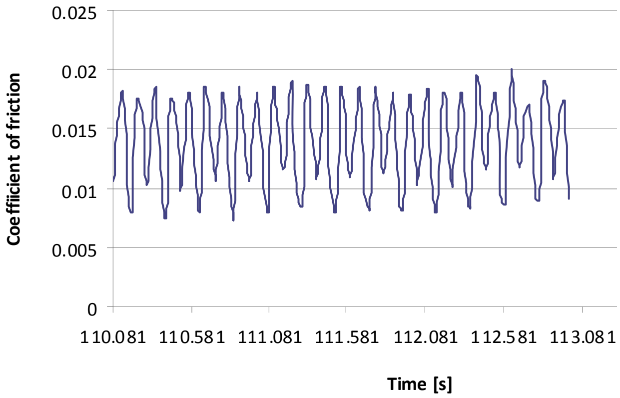

Figure 4. The coefficient of friction for a 20 % coverage bushing

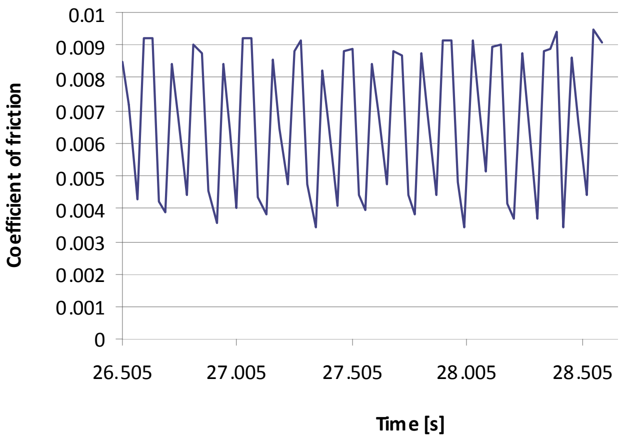

Figure 5. The coefficient of friction for a 30 % coverage bushing

5. CONCLUSION

One can conclude that the increase in graphite coverage leads to decrease of the coefficient of friction. For a self-lubricating graphite bearing 50 / 40 × 40 mm, with a diameter of graphite inserts of 10 mm, the coefficient of friction is twice smaller for coverage of 30 % than in case of 20 % coverage, for a same radial force (3000 N) and the same sliding speed. This can be explained by smaller number of contact points (graphite inserts) in case of 20 % coverage. For 20 % coverage there are twelve 10 mm diameter inserts, while for 30 % coverage the number of 10 mm diameter inserts is twenty. Due to smaller number of contact points in case of self-lubricating bearing with 20 % coverage, the coefficient of friction increases thus resulting in lower bearing lubrication. Lamellar graphite particles are easily detached from the insert. The detached particles are entering into the recesses of the surface roughness of the sliding material (CuSn12) and adhere to the material so that the bearing and shaft surfaces are not in direct contact.

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