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FACULTY OF MECHANICAL ENGINEERING IN NIŠ

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Experimental Approach in Estimation of the Friction Coefficient in the Self Lubricating Plain Bearings with Graphite Filler

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Abstract— **Plain bearings are the structural parts used for constraining the relative motion between two parts only to desired motion. Opposite to the rolling bearings, plain bearings constrain motion over the sliding surfaces - the journal (part of the shaft) and the bearing surface. Such a contact is fully frictional since contact surfaces induce friction forces that decrease load capacity and efficiency of the bearing. To avoid such a contact, plain bearings are lubricated at a certain level by a certain method. A certain group of plain bearings uses graphite filler as the lubricating material. External flange of such bearing has radial holes filled with the graphite that is in contact with the journal. The goal of a research presented in this paper is to identify the impact of the graphite lamella's diameter to the coefficient of friction. The results obtained experimentally can be used in order to predict the behavior of plain bearings with graphite exploitation in real terms.**

Keywords— **plain bearing, friction coefficient, diameter graphite lamellas.**

I. INTRODUCTION

Mutual movement of parts in contact and load transfer on self-lubricating bearings is achieved by sliding, when the mobile surface of the sleeve slides over the fixed surface of the bearing. Only on the place between the sleeve and the bush a friction has been creating at self-lubricating bearing.

The process of friction as the always and everywhere present physical phenomena evokes many negative effects in production systems and the designers are trying to reduce them to a minimum, in order to reduce unproductive energy losses as well as reducing the wear of elements in tribomechanical systems. The main parameter that characterizes the process of friction between shaft and the sliding sleeves is the coefficient of friction. The measurement of this coefficient can be achieved using experimental methods. Experimental tests came up to the value of the coefficient of friction for a pair of aluminum - steel. The value of the coefficient of friction has sharply decreased from 0.8 to 0.1. Some initial tests were conducted on Tribotestor M '89. Adjusted input parameters were: sliding speed and radial load [1, 2].

By applying the experimental method it was tested for coefficient of friction bronze (CuZn25Al6) with inserted graphite blades. The results show that the coefficient of friction decreases with increasing normal load [3, 4].

The sliding surfaces are in plain bearings are mostly lubricated by solid lubricant (graphite), which can be in the dry state or combined with carrier (fat - graphite grease). A self-lubricating solid lubricants show low friction and/or wear during sliding in the absence of additional external lubricant. Self-lubricating graphite is used because its graphite crystal structure – due to small tangential force applied to the layers of graphite, they slide one to another. As such, graphite is able to form a film on the metal and as such replaces the lubricant. By using bronze bushings with graphite, graphite forms a thin film on both the contact surface and as such it is highly resistant to shocks, and retains its lubricating properties even in idle bearing [5]. Responding graphite content and hardness are the two most important factors for achieving a good-quality lubricating film [6, 7].

New technologies monitor the development and use of new materials, particularly composites, which are considered very important materials that meet high standards of modern technology. Rakowski, Donnet and Sonam [8]-[10] have studied the mechanisms of traditional lubrication and new solid lubricants, with special emphasis on application of solid lubricants in practice. Strength and deficiency are two key criteria that determine the choice of anti-friction material for sliding bearings. The choice of materials for making the sliding bearing depend on the exploitation regime, primarily on the size and character of the specific pressures and sliding speeds [11].

Wang [12] made an analysis and modeling of a friction in a sliding bearing. Based on the Grey System Theory, examined the impact speed and the load on the friction coefficient of sliding bearings. The degree of Grey Relational Analysis shows that the load has a more significant effect on the coefficient of friction compared to the number of revolutions. Based on the Grey System Theory [13], the proposed method of Grey Relational Analysis is used to analyze the influence of different parameters on the friction torque in a sliding bearing. Based on Grey Relational Analysis was performed multivariate Grey model GM (1, N, D) to determine the friction torque in a sliding bearing. The calculated results show that, compared with other factors, the coefficient of friction, load, temperature and speed of rotation have a greater impact on bearing friction moment as an example, using the default Al - alloy sliding bearing.

Friction solid body releases heat. Bearing temperature is an indicator that indicates the presence of unexpected external heat source, or bearing damage, lubrication of the bearing, the adequacy of load, speed, speed skating and others. The released heat can warm up a lubricant to the ignition temperature. To prevent this, the thermal process is simulated in plain bearings, taking into account the movement of the shaft. Authors [14]-[18] developed an algorithm that has proved effective for thermal diagnostics friction.

II. Friction and Temperature on Self-Lubrication Bearings

The results of most studies very poorly mention the influence of coverage sliding speed, a diameter of graphite flange, impact of radial force on the plain bearing, the temperature itself and even more in the most of papers these results are not even mentioned. The existing literature gave an overview about the self-lubricating bearings, but has not touch upon the answer to the question: How parameters as: sliding speed, the coverage, changes in diameter of the graphite flange and constant radial force, impact the temperature and friction coefficient of radial self-lubricating bearing?

The friction force is the basic parameter for quantification and evaluation of friction process while friction coefficient is being used for friction size determination. The coefficient of friction is a complex tribological parameter, which has a stochastic feature, due to many influencing factors. Therefore the analytical procedure for the determination of the coefficient of friction is very complex. The problem of analytical determination of the coefficient of friction at self-lubrication bearings is further more complicated by the character of the contact between self-lubricating bushings and the shafts as well as the load of self-lubricating bushings due to radial forces. Namely, self-lubricating bushing through its active surface has a variable contact with the shaft at variable speeds. Therefore, the use of experimental methods is simple procedure for coefficient of friction determination. Determination of coefficient of friction starts from Coulomb equations which associates load and friction coefficient as the function of time at contact between two real bodies.

$$F_{\mu}(t) = \mu(t) \cdot F_{n(t)} \rightarrow \mu(t) = F_{\mu}(t) / F_{n}(t) \tag{1}$$

where: μ-coefficient of friction, Fμ-friction force, Fn-normal force.

Any increase of friction leads to increased bearing temperature, which causes a change in properties of the sliding layer that is in contact with the sleeve and the bearing capacity of the layer to change the coefficient of friction, which eventually caused the destruction of the bearing. Elevated temperatures can be either a result of heat generation, or the conditions in which the contact takes place. Temperature rise as a result of development of heat caused by friction is given by equation:

$$\Delta T = C \cdot \mu \cdot p \cdot v \tag{2}$$

where is the C- overall thermal resistance from the surface of heat dissipation. Approximate values that are recommended in the literature may be in the range of $0.8 \cdot 10^{-3}$ to $1.6 \cdot 10^{-3}$ °Cms/N, μ- coefficient of friction, p, v- values of contact pressure and sliding velocity, respectively.

III. Experiment

The task of the constructor of self-lubricating bearing is to reduce friction in order to decrease non-productive losses of energy while reducing friction and wear. Experimental researches on self-lubricating bearings were performed in order to determine the coefficient of friction. Determination of coefficient of friction was based on the equation (3). The frictional force occurs when there is relative movement of the shaft in relation to the self-lubricating bearing. As the speed of translational movement of the shaft is reduced to zero, it is considered that the frictional force occurs due to the load (radial) force and torque. The minimum value of the torque that causes rotation of the shaft, which eliminates the resistance of the frictional contact, is the friction torque, which is a function of time. Moment of friction is obtained as the product of perimeter force in the bushing (using force sensors) and the distance arm.

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

The coefficient of friction is calculated over the conditions of equality torque friction in contact shells and shafts and torque that the entire system operates in support of O2. In the experiment were used self-lubricating plain bearings with graphite, 50 / 40x40 manufactured in Fasil ad, whose cribs made from basic materials CuSn12 high quality with inserts (lamellae) of graphite of a diameter of 12mm, 10mm, 8mm, which homogeneously adhere to the surface by taking 20% of the sleeve. Figure 1 shows a self-lubricating bearing contact with the shaft.

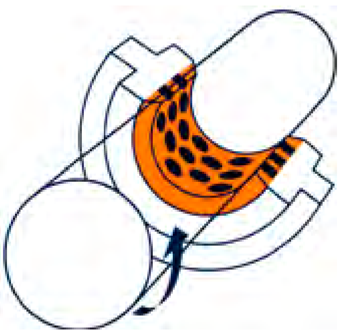

Fig. 1 Self-lubricating bearing in contact with the shaft

Fig. 2 shows a scheme of the measuring configuration for measuring the values necessary to determine the coefficient of friction at the contact of self-lubricating bearings and shafts. The main rotary motion provides a work-machine - lathe Potisje Ada PA-C30 (SG) over which the sensor torque M1 (torque sensors Hottinger Baldwin Messtechnik T1-100 Nm) and connectors (S1) of transmission input shaft (V) are mounted. On the branch of the shaft is mounted self-lubricating bearing (SL) which is radial and axial fixed upper (GN) and lower (DN) parts of the support. Through the opening of the upper support (GN) above the lower addition of the force sensor (D), acts on the radial bearing by a force which is induced by tightening the screw and the magnitude of the force measured by force sensor M2 (Hottinger Baldwin Messtechnik S7M-500N). Circumferential force, which represents the frictional force of the bearing, measured by means of a lever (P1), the length of 150 mm, which is arranged perpendicular to the axis of the bearing and stiffnes GN. Force sensors S1 (Hottinger Baldwin Messtechnik U9-5 kN) measures the intensity of this force. The temperature of the self-lubricating bearings is measured by thermo couples (T (t)) and (T1 (t)). The thermocouple (T (t)) measures the temperature at the center of the bearing bushings, and thermocouple (T1 (t)) measures the temperature at the end of the bearing bushings (SL).

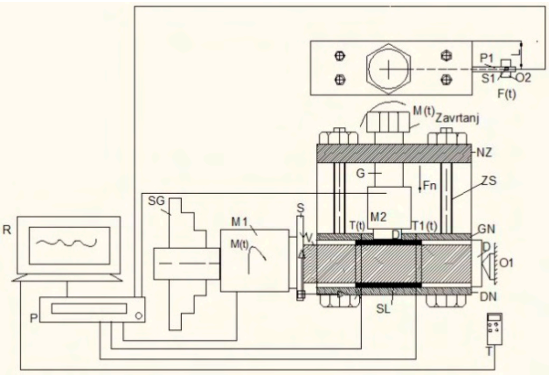

Fig. 2 Scheme of measurement configuration

Thermal camera (T) records the surface temperature of the assembly (Fig. 3). During the test, the environment temperature was 25 °C.

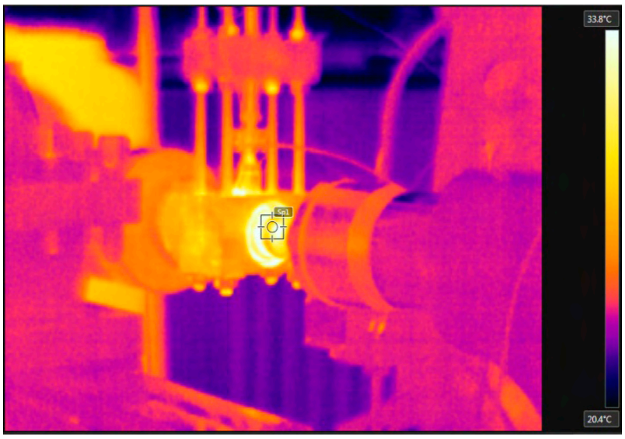

Fig. 3 The maximal temperature measured by the thermal camera T_{max}=31.3 ^{0}C.

All sensors, thermo couples and thermal camera are associated with computer unit (R) which performs acquisition of data.

Three sets of experimental research were performed, where the diameters of graphite lamellae has been varied: 12 mm, 10 mm, 8 mm respectively, while coverage of graphite was 20%, sliding speed was 0.11 m/s and radial force was 1500 N and they have had quasi constant values.

It is important to point out a relationship between diameters of the graphite lamellas and the constant coverage of constant 20%: in order to maintain the same coverage for smaller diameter of lamellas, there have to be more lamellas, e.g.: for diameter of 12 mm there has to be 9 holes with lamellas while for diameter of 8 mm it is necessary to have 20 holes.

IV. Estimated Coefficient of Friction

Estimation of the coefficient of friction for three different sets of experiments was conducted for period of work not shorter than 180 s, at the same environment temperature, sliding speed and radial loading delivered to the bearing. The value of friction coefficient has been taken as an average value for a time period of app. 3 s, for a stationary working condition – which appears app. 5 s after initial loading the bearing. The following figures 4, 5 and 6, are representing the values of friction coefficient for experiment sets 1, 2 and 3 where the diameter of lamellas was 12 mm, 10 mm and 8 mm, respectively.

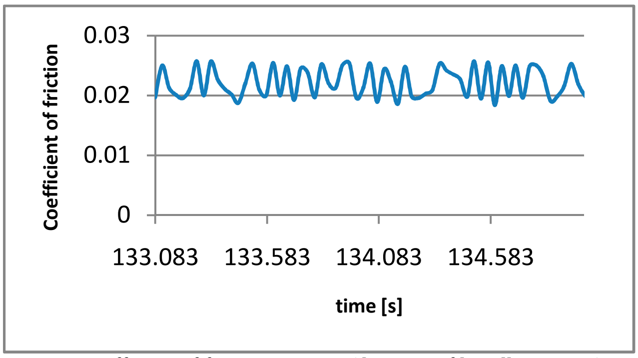

Fig. 4 Coefficient of friction μ=0.023 (diameter of lamellas 12 mm)

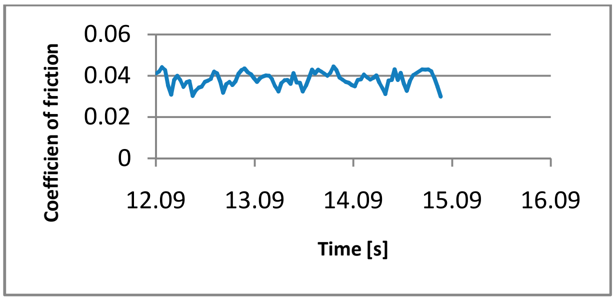

Fig. 5 Coefficient of friction μ=0.039 (diameter of lamellas 10 mm)

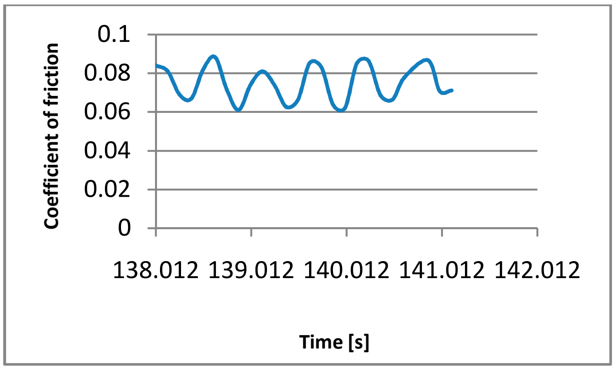

Fig. 6 Coefficient of friction μ=0.079 (diameter of lamellas 8 mm)

Time period where the friction coefficient was averaged has been selected randomly.

V. Conclusions

It is very difficult to make a full and direct connection between contact condition, coefficient of friction and geometry of a bearing which will unilaterally guide to definition of friction coefficient. As a step forward in better understanding the friction coefficient, the research in this work is done by experimental method, in order to predict the behavior of self-lubricating sliding bearings with graphite in real operating conditions. By applying the experimental research, the variation of tribological parameters, necessary to estimate friction coefficient is limited strictly to the diameter of the graphical lamellas – to make it as simple as it is possible.

The object of a research was a self-lubricating bearing 50/40×40, with diameters of graphite 8 mm in experiment set 1, 10 mm in experiment set 2 and 12 mm in the experiment set 3. The radial (loading) force 1500 N, sliding speed of 0.11 m/s and graphite coverage of 20%, as well as the rest of bearing geometry are considered as constant.

The lowest coefficient of friction was estimated during experiment set 1, at the maximum diameter of 12 mm, while the largest coefficient of friction was estimated in experiment set 3, for the diameter of 8mm. With graphite coverage of 20% for the insertion of graphite lamella 12mm it was necessary to drill 9 holes in the bearing and for diameter of 8 mm - 20 holes.

This guides to a weak conclusion that the greater diameter of the lamellas imply the smaller number of holes, and finally the smaller number of bigger contact areas between the sleeve and the shaft. Vice versa: the smaller the diameter of lamellas implies the greater number of holes what further implies the greater number of smaller contact areas.

The diameter of graphical lamellas makes a direct influence on a value of the friction coefficient during steady state and controlled working condition. The graphite coverage is important for the friction coefficient as well and since the graphite coverage and the diameter of the graphite are in direct dependency, a strong conclusion about the friction coefficient and diameter of the graphite laminate can be done after deeper analysis of the coverage, as well [19].

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