Management Systems
in
Production Engineering

2018, Volume 26, Issue 4, pp. 207-211

Date of submission of the article to the Editor: 04/2018
Date of acceptance of the article by the Editor: 08/2018

DOI 10.1515/mspe-2018-0033

THE OPTIMAL TRIBOTECHNICAL FACTORS IN THE DESIGN OF MACHINES – ENVIRONMENTAL ELEMENT IN THE PRODUCTION SYSTEMS

Ján TURIS, Pavel BEŇO
Technical University in Zvolen

Witold BIAŁY
Silesian University of Technology

Abstract:
The article deals with the experimental detection of the tribotechnical parameters for two different types of sliding bearings which are intended for operation without an additional lubrication due to the increasing requirements in ecology and environment. The dominant tribotechnical parameters of the self-lubricating bearing are the coefficient of friction and temperature. To determine these parameters, an experimental method was applied in this paper. The introductory part deals with materials of self-lubricating sliding bearings, their properties and usage. The experimental part consists of the evaluation of friction characteristics and geometric change of a surface after sliding pairs were being worn. Experimental examined sliding pairs comprise a sliding bearing and the shaft. The aim of this research was to determine the effect of radial force on tribotechnical parameters in order to predict the behavior of examined sliding bearings in real operating conditions.

Key words: *coefficient of friction, self-lubricating bearing, tribology*

INTRODUCTION

Today, the importance of environmental aspects in production systems is increasing. The importance of the research in the field of bearings with small requirements for the lubrication has increased as the leakage of even a small amount of the lubricants outside its regular working environment is a potential environmental threat. Before the choice of a sliding bearing is made all the requirements for the sliding knot have to be taken into consideration.

Sliding fit operating without lubricants are characterised as fit where friction surfaces are not separated by any lubricant layer. These sliding fit are usually used in the machines or devices where the presence of lubricants, lubricating greases and oils in sliding fit could be dangerous or unsuitable. They are used plastics and sintered carbides. Examples are in the food and pharmaceutical industries, where the risk of contamination by the lubricant forbids its application. The leakage of even a small amount of the lubricants outside its regular working environment means the environmental threat when working with forest and agricultural machinery [4]. However, specific bearings designed exclusively for boundary lubrication operation do exist. They are used especially in cases where there is not possible to create sufficiently thick layer of hydrodynamic lubricating film. Their characteristic properties include higher loss caused by the friction, low capacity and ability to operate with worn friction surfaces. Temperature, speed of a pivot, load of a bearing, subsequently grinding, and cumulative wear are the main factors affecting boundary lubrication bearing´s lifespan [1]. Boundary lubrication usually occurs under high load and low speed conditions in bearings, gears, cam and tappet interfaces, piston rings, pumps, transmissions, etc. Sintered bronzes with additives of other elements are widely used as bearing materials. Liquid or solid lubricants are often inserted into the porosity of the material. They may contain a self-lubricating tribological material, or may be sealed for life [5]. Two of self-lubricating material widely used in general engineering applications are plastic and metals containing solid lubricant fillers. Self-lubricating materials are most useful either as dry bearings or as bearings in marginally lubricated applications. Self-lubricating bearings have evolved from the steadily emerging technology of powder metallurgy [2]. Powders of copper, tin and graphite are sintered, and are then passed through a sizing die to the correct dimensions. The metal thus produced is of a porous nature, capable of holding up to one third of its volume oil [9]. A pressure on the bearing surface or a temperature rise will cause the oil to exude, so that lubrication is ensured continuously and automatically where it is needed. Porous self-lubricating bearings comprise the main types of material: sintered bronze (90% copper, 10% tin, with or without graphite), sintered iron – graphite, sintered iron – bronze (diluted bronze). Sintered bronze bearings are made from elemental copper and tin powder, prealloyed bronze powder or mixtures thereof. Compaction and sintering are arranged to allow about 25% porosity for subsequent impregnation with oil.

The significant increase in wear at increased load was recorded by Prasad [6]. Suresha et al. [10] researched the composite materials. The values of the friction coefficient increased with the subsequent increase of load. Larger wear was recorded with increasing load and sliding speeds. The coefficient of friction of the alloy CU-Al-Zn-Ci increases with the sliding speed, but decreases with the increasing pressure up to 1.5 MPa. Above this pressure value the trend reverses and the friction coefficient increases [7]. Ďuriš and Labašová researched the influence of radial load and sliding speed on the value of the friction coefficient. Experimentally at obtained the friction coefficient for the sliding pair aluminum – steel [3]. Experimental results of Sekereš were achieved for sliding bearings under radial load up to 0.8 kN. Examines lubricated and self-lubricating plain bearings with an emphasis on temperature [8]. The published results of the experiments show the interdependence of the material of the self-lubricated slidding bearings, their geometry and the friction characteristics, especially at lower loads. Experiments are often time-consuming and loading was smoothly. The aim of this the experiment was to determine the behavior of two selected sliding bearings at the jump load and at the load above 1 kN.

METHODOLOGY OF RESEARCH

Measuring was performed by measuring device Tribotestor M´12. A sliding bearing 3 (Fig. 1) along with a thermo sensors was placed into the measuring head 1 and slid by rotating onto the shaft 2 made with the margin of f7 and with the roughness of Ra 0.8 μm.

Two types of slide bearings made of different materials, but with identical dimensions – 40 mm diameter and 40 mm wide, samples were measured. The self-lubricating sliding bearing A – sinterbronze was made with the tolerance of diameter G7. The sliding bearing bronze matrix B for high speed was made with the tolerance of H7. The bearings A – they are made by powder metallurgy from metallic powders of a different grain size and form. Porous metal with porosity of 25% is saturated by oil that serves as a lubricant for sliding knot. The basic bearing material is Cu with admixture of Sn 9-11%, C 0.2% and other elements 2%. High-speed bronze alloy bearings B are made from bronze matrix. This matrix is saturated by mineral oil. The basic material is SINT A50 and the pore volume is 28%. The shaft made of steel 42CrMo4 was spun by the electric motor AP 100-L2 and the frequency transformer VQFREM to the given revs.

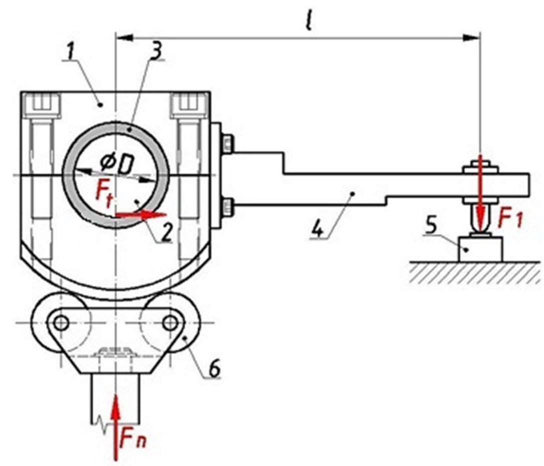

Fig. 1 The measuring principle
1 - measuring head, 2 - shaft, 3 - sliding bearing, 4 - arm, 5 - force sensor, 6 - transmission mechanism

In preparation of experiments was carried out inspection of the frequency converter via the contactless tachometer C.A. 1727 from Chauvin Arnoux, which is accurate to within 0.0006 min^{-1}.

The radial load was exerted on the measuring head by the stepper motor and the transmission mechanism. Load force generated friction force Ft. The friction force occurs when there is a relative movement of the shaft with respect to the self-lubricating bearing. It is considered that the friction force is a result of the radial load and the torque. The part of the torque that causes the rotation of the shaft, which overcomes the frictional resistance of the contact, is the friction moment, which is a function of time. The friction moment is calculated as the product of the force obtained by the force sensor 5 and its lever arm 4. The coefficient of friction is calculated by the condition of equality of the friction moment in contact of the bushing and the shaft and the torque. The sliding bearings were loaded progressively by changing loads (radial force Fn) at given constant revs n = 150 min^{-1}. The radial force value was set to 0.6 kN. The load was increased half at hour consecutively by 0.3 kN (values of 0.9 kN, 1.2 kN, 1.5 kN). Measuring was performed ten times on two shafts (h1, h2). In the different sliding bearings A and B (in three different samples from each) was tested with each shaft, a total of 60 variations of measurements (shaft h1 – bearings A1, A2, A3; shaft h2 – bearings B1, B2, B3). Measured were averaged and evaluated afterwards to graph and statistically.

During experiment we have done measuring without any additive lubricating aiming to determine the coefficient of friction values at given load and revs. The processing of the measured values and calculation of the coefficient of friction was in the software environment of the Tribotestor M´12.

The temperature of the self-lubricating bearing was measured by the resistance temperature device RTD – J IP 8509. The temperature was measured on the outer and front surface of the bearing. The diameter measuring of the bearings and shafts was performed on the measuring device CONTURA G2 with the scan sensor VAST XT Gold with accuracy of 0.0001 mm. Another studied parameter of surface quality was the roughness of sliding bearings before and after experiment. The measuring was performed by the contact method with the measuring device MahrSurf GD25.

The determination of weight loss is in general one of the ways how to assess the wear of sliding parts. Folowed weighing before and after was made on Tribotestor M´12 within the experimental sliding bearings testing. The weight of sliding bearings was ascertained with the laboratory scales Sartorius ED224S-PCE with accuracy of 0.0001g.

RESULTS OF RESEARCH AND DISCUSSION

The first monitored parameter was the friction force value. Its adapted progress with particular sliding bearing can be seen in Fig. 2. As seen in the graph, the highest values of friction force were determined in bearings A type. The friction force of bearing A is almost always the same for the different radial loads. First a peak at nearly the same level and then a decreasing friction force to the identical level. However, as seen in the graph, the bearing B had better characteristic. The increase of friction force when increasing radial force was seen at bearing types B, however, the friction force value had almost linear course after levelling off. The range of friction force value was between 0.08 and 0.18 kN.

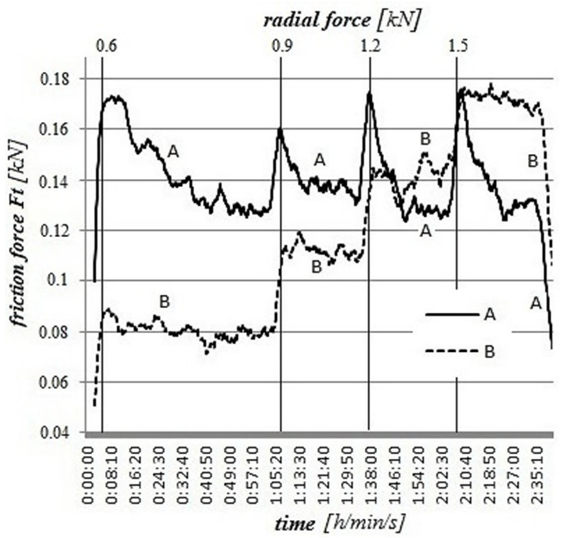

Fig. 2 The course of average frictional forces in the bearing A and B

Temperature measured on sliding bearings was another important measured parameter. As seen in Fig. 3 the average temperature was closely associated with the friction force value and it almost copied the increase of this force. Jump shifts at the increase of radiall force are visible in the graph. The average maximum of temperature bearings reached the value of 58°C (A) and 65°C (B).

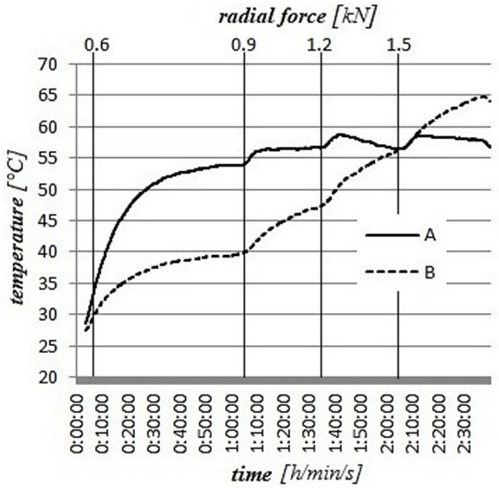

Fig. 3 The course of average temperature in the particular bearing A and B

The coefficient of friction average values are seen in Fig. 4. As seen in the graph, the coefficient of friction decrease could be noticed with the bearing A. This could be caused by the enlargement of the contact layer. The value ranged between 0.26 and 0.08. During experiment with the A type bearings the coefficient of friction was permanently significantly decreasing, even if the pressure of radial force increased. It is assumed that this was caused by a change in the contact microstructure. Almost linear character can be seen during the process of measuring with bearing B.

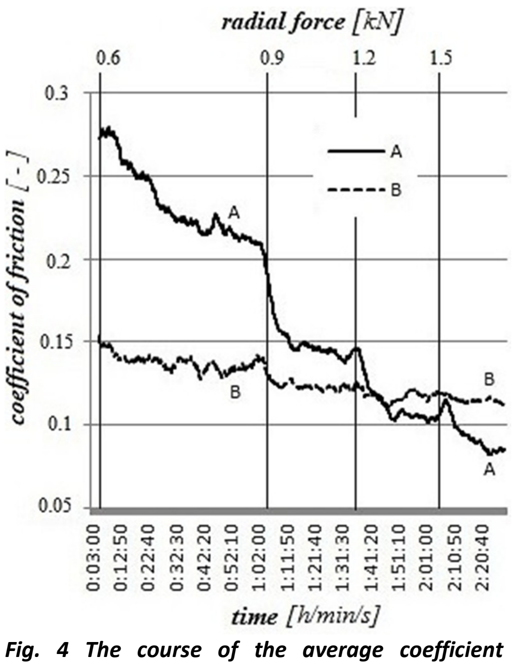

Fig. 4 The course of the average coefficient of friction in the bearings A and B

The measuring of the weight loss of sliding surface material was used to evaluate bearings wear. The weight loss in both sliding bearings (A and B) decreased practically identically. The difference in sliding bearing A was 0.2 g and the average difference in B samples was 0.3 g. No extreme was observed in this measurement. The average bearing mass of A before the experiment was 188.2 g and dropped to 187.95 g after the experiment.

Bearing B lost from the original 175.42 g exactly 0.3 g and the weight dropped to 175.12 g.

The changes of inner diameters of particular samples before and after the friction characteristics measuring are seen in Fig. 5 (bearing A) and Fig. 6 (bearing B). The diameter enlargement can be seen in every sample. Less advantageous results were reached with Type A bearings. The average difference before and after measuring was about 0.1 mm in all three bearings. As for of this dimensional measurement, type B bearings were the most advantageous for such loading conditions. The average differences in particular bearings were only in µm.

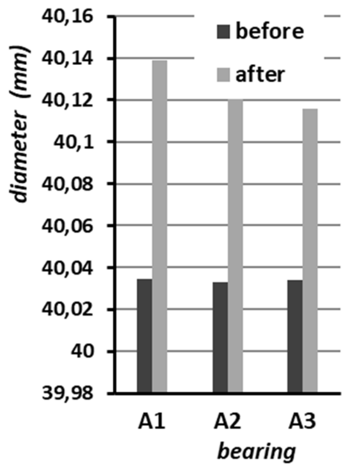

Fig. 5 The average values of the diameter of the bearings A before and after the experiment

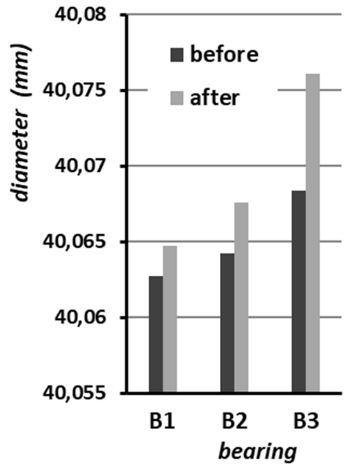

Fig. 6 The average values of the diameter of the bearings A before and after the experiment

The device CONTURA G2 ZEISS allows measurement the deviation of cylindricity. The average deviation of cylindricity of bearing A was changed from 0.0093 mm at to 0.0169 mm. The average deviation of cylindricity of bearing B was changed from 0.01 mm at to 0.0318 mm. Examining the cylindricity gives a more comprehensive view of the bearing change, as a survey of the average.

The roughness of the samples changed substantially. The measured average surface roughness Rz values before the experiment were 22 µm (bearing A) and 36 µm (bearing B). For comparison, the average roughness values of Ra were 3.1 µm (bearing A) and 4.8 µm (bearing B). After the experiment were the average values of Rz 5 µm (A) and 13 µm (B), for comparison Ra of 0.3 µm (A) and 0.65 µm (B). The roughness peaks on the bearings contact surface became abraded and as a result the overall changed surface roughness. From the graphs it is possible to deduce better properties of the self-lubricating bearing B. For such a load, it has more convenient properties.

At lower load values, the Sekereš´s experiment was confirmed. The material composition of the self-lubricating bearing A loses its strength properties at the start of the load cycle, but despite this loss, the bearing surface appears to be rapidly recovering and the coefficient of friction is lower, than expected. The friction force during the experiment with bearing A had the highest standard deviation, which also encouraged further research. It is assumed that bearing A produces substantial abrasion, particles of wear, at the start of the load. This was repeatedly monitored at the scheduled interrupted of the experiment.

The standard deviation in measured quantities in the experiment was acceptable in all measurements. The experimental values had a small scattering around the mean value in the measurements. Due to the scale, the original charts from the Tribotestor M'12 software environment were presented. It is possible to forecast the performance and life span of critical tribological elements and their friction surfaces of machine and mechanism constructions thanks to continuous temperature monitoring. Of course, after the following experiments, correlation analysis will be appropriate and necessary. The tribological laboratory will also investigate the morphology of the particle bearing wear. In the next research, it will be necessary to examine whether the friction coefficient will not culminate sharply. In this jump load, this assumption is because it indicates the temperature course. At high temperature, the bearing surface may have lost its strength properties more quickly. High-speed self-lubricating bearing SINT A50 seem to be better bearings for this kind of loading. The growth of friction force copied the increase of radial force.

CONCLUSIONS

The article presents the experimental research into the influence of radial load on the tribotechnical characteristics of the sliding bearings. Experimentally studied were frictional forces, temperature, geometric shape and size, roughness and loss of weight self-lubricated bearings werw. The friction coefficient was calculated. The effect of radial load was determined for two different bearings. The coefficient of friction value decreases with the increase in radial load probably due to plastic saturation of the contact area. The temperature of a self-lubricating sliding bearings in the increases with the increase in radial load. Further research should be directed towards understanding the tribotechnical mechanisms at higher loads and contact pressures.

REFERENCES

[1] S. Andersson, A. Söderberg and S. Björklund. “Friction models for sliding dry, boundary and mixed lubricated”. *Tribology International*, vol. 40, pp. 580-587, 2007.

[2] H. Czichos and K.H. Habig. *Tribologie-Handbuch*. Berlin: Mercedes Druck, 2010.

[3] R. Ďuriš and E. Labašová. “Experimental Determination of the Coefficient of Friction in Rotational Sliding Joint“. *Applied Mechanics and Materials*, vol. 309, pp. 50-54, 2013.

[4] M. Kučera, Z. Aleš, Z. Ivandić and Ľ. Hujo. “Possibility of hydraulic fluids with a low environmental impact application in agriculture and transport machinery”. *Journal of Central European Agriculture*, vol. 14, pp. 1592-1601, 2013.

[5] J. Marienčík. “Research of selected tribological factors of contact layers of identical shape-like smooth surfaces“. PhD. thesis, Technical University in Zvolen, Slovakia, 2014.

[6] B.K. Prasad. “Sliding wear response of a grey cast iron: Effects of some experimental parameters“. *Tribology International*, vol. 44, no. 5, pp. 660-667, 2011.

[7] T. Savaskan and O. Bican. “Dry sliding friction and wear properties of Al–25Zn–3Cu–3Si alloy“. *Tribology International*, vol. 43, no. 8, pp. 1346-1352, 2010.

[8] J. Sekereš and J. Turis. *Tribology in Design of Machine Parts*. Zvolen: TU, 2009.

[9] A.A. Shatsov, I.V. Ryaposov and D.M. Larinin. “Model of fracture, friction and wear Phenomena of porous iron”. *Advances in Tribology*, vol. 2011, 2011.

[10] B. Suresha, G. Chandramohan, J.N. Prakash, V. Balusamy and K. Sankaranarayanasamy. “The Role of Fillers on Friction and Slide Wear Characteristics in Glass-Epoxy Composite Systems“. *Journal of Minerals and Materials Characterization and Engineering*, vol. 5, no. 1, pp. 87-101, 2006.

Ing. Ján Turis, PhD.
doc. Ing. Pavel Beňo, PhD.
Technical University in Zvolen
Faculty of Environmental and Manufacturing Technology
T.G. Masaryka 24, 960 53 Zvolen, Slovak Republic
e-mail: jan.turis@tuzvo.sk
pavel.beno@tuzvo.sk

dr hab. inż. Witold BIAŁY, prof. PŚ
Silesian University of Technology,
Faculty of Organization and Management, Institute of Production Engineering,
ul. Roosevelta 42, 41-800 Zabrze, Poland
e-mail: wbialy@polsl.pl