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THE EFFECT OF SLIDING TIME ON STATE OF THE LUBRICATION TEMPERATURE AND COEFFICIENT OF FRICTION OF RADIAL PLAIN BEARING MADE FROM AN ALLOY OF TIN - TEGOTENAX V840

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Abstract

An experimental research of the effect that the oil supply temperature do on the performance of a 60x40x40 mm plain journal bearing is completed on the tin-based white metal material. Numerous experiments have been performed under forced lubrication; different static loads from 1045 to 3601 N and two different values of the shaft rotational speed were tested (1000 and 2000 rpm) by using the custom-built journal bearing test rig. The lubrication temperature of the bearing and the coefficient of friction vs. sliding time were measured. The realization of the frictional coefficient change is performed from the start of operation until the steady-state oil supply temperature. The results from this research demonstrate significant decreases of the frictional coefficient from startup till steady-state.

Key words*: plain journal bearing, white metal (Babbitt), coefficient of friction.*

1 INTRODUCTION

Bearings are machine elements which are used to support a rotating part, namely a shaft. They transmit the load from the rotating part to a stationary part known as frame or housing. They permit relative motion of the two parts in one or two directions with minimum friction, and also prevent the motion in the direction of the applied load. The bearings are classified broadly into two categories based on the type of contact they have between the rotating and the stationary part:

- Sliding contact
- Rolling contact

Sliding contact bearings are classified in three ways [1]:

a) Based on type of load carried
b) Based on type of lubrication
c) Based on lubrication mechanism

A number of research projects contributed to the experimental dataset for hydrodynamic journal bearings in the past 15 years. One of these groups led by Ron Flack in the 90's at the University of Virginia determined characteristics of many different bearing geometries using a test rig in which the bearing housing was shaken against a rigidly mounted shaft [2, 3]. Of note in this series of works is that the error between the theoretical and the experimental values is generally more reasonable than that of the majority of research work in this field. These works provide valuable bearing data regarding a number of bearing configurations.

High viscosity index lubricants were investigated in a low speed journal bearing by Kasai et al. [4] finding that high viscosity index VI lubricants lead to reduced bearing friction in some cases and higher maximum oil film pressure when compared to a polyalphaolefin base oil.

The test equipment was then modified by D W Childs et al. [5] to include dynamic testing capabilities using an arrangement of springs and hydraulic shakers, with a detailed analysis of the experimental uncertainties and parameter determination methods. In N M Franchek et al. [6] the rig was used to compare the performance of several different hybrid bearing geometries. This study found that while each of the four geometries tested had specific advantages, the use of an inlet angled against the flow of rotation provided the most beneficial characteristics.

In Jerzy T Sawicki et al. [7] several methods are compared for extracting journal bearing dynamic coefficients from experimental data and several useful observations are made regarding test rig design and construction. This work was continued with Jerzy T et al. [8] using a test rig with a controllable counter-rotating shaft initially reported in Jerzy T Sawicki [9], with the conduction of experimental work that produced bearing dynamic coefficients within 30% of theoretically predicted values.

A new developed bearing test rig [10] for measurement of hydrodynamic pressure and friction moment of plain journal bearing was presented by Mazdrakova et al. The stand is designed and constructed with exchangeable bearing bushes which are coated on the inner side with different finishes with a uniform thickness and various elastic characteristics. Static measurement capabilities include operating eccentricity and continuous circumferential pressure at multiple planes in axial direction.

The thermal effects within the journal were investigated by A K Tieu et al. [11] under misalignment and with varying loads, finding that temperature in the bearing was in the expected range with the highest temperatures occurring in or immediately after the region of the lowest film thickness. Further work included harmonic excitation of the shaft via a journal bearing [12], [13] with a detailed analysis of the calculation method and uncertainty in the measured values of the bearings' dynamic coefficients.

An experimental study of the influence of oil supply temperature and supply pressure on the performance of a 100 mm plain journal bearing with two axial grooves located at ±90 deg to the load line was carried out. The hydrodynamic pressure at the mid-plane of the bearing, temperature profiles at the oil-bush and oil-shaft interfaces, bush torque, oil flow rate, and the position of the shaft were measured for variable operating conditions by F. P. Brito et al. [14]. It was found that the existence of the downstream groove significantly affects the temperature profile at the oil-bush interface except for low load, low feeding pressure cases, where the cooling effect of the upstream groove is significant.

Javorova et al. [15] studied the performance of a finite length journal bearing, taking into account effects of non-Newtonian Rabinowitsch flow rheology and elastic deformations of the bearing liner. According to the Rabinowitsch fluid model, the cubic-stress constitutive equation is used to account for the non-Newtonian effects of pseudoplastic and dilatant lubricants. Integrating the continuity equation across the film, the nonlinear non-Newtonian Reynolds-type equation is derived. The elasticity part of the problem is solved on the base of Vlassov model of an elastic foundation [16].

The tribological behaviors of Babbitt alloy16-16-2 sliding against aluminum bronze ZnCuAl9Mn2 lubricated by sea water were systematically investigated by Hairong Wu *et al*[17]; the results indicated that the friction coefficient decreased as the load increased to 30N and then remained at a steady level at high loads, but decreased with increase in sliding speed. Zeren, A. et al. [18] studied the tribological behavior of two different tin-based bearing materials in dry sliding conditions, one of these alloys with low Sb content (7%) is known as SAE 12 and is widely used in the automotive industry and the other with high Sb content (20%) is a Sn–Sb–Cu alloy. Search results have proved that WM-2 and WM-5 alloys can be used in dry sliding conditions, It is shown that performance of WM-5 under heavy service conditions is better than WM-2 due to its alloying elements.

Experimental research to study performance of the plain journal bearings, made from white material (babbitt) in the case of forced lubrication, is conducted in order to determine change of the friction coefficient with respect to the temperature change during operation. In this paper, two studies were conducted to investigate the effects of the friction coefficient changes with respect to the temperature changes of the hydrodynamic plain journal bearings. The first defined the changes if the friction coefficient changes along with the temperature change as function of time at the start of the operation, i.e., before the stability of oil temperature. The second did the same but when the oil temperature became stabilized.

2 THEORETICAL FOUNDATION OF THE PLAIN JOURNAL BEARING FRICTION

In 1883, Petroff published his work on bearing friction based on simplified assumptions:

a. No eccentricity between bearings and journal and hence no "Wedging action".

b. Oil film is unable to support load.

c. No lubricant flow in the axial direction.

Regarding Fig. 1, an expression for viscous friction drag torque is derived by considering the entire cylindrical oil film as the "liquid block" acted upon by force F. From Newton's law of viscosity:

$$\tau = \eta \frac{du}{dy} \tag{1}$$

If the supposition is made that the rate of change of velocity or the rate of shear stress is a constant, then *du*/*dy*= *u/c* and Eq. (1) becomes:

$$\tau = \eta \cdot \frac{u}{c} \tag{2}$$

$$u = \omega \cdot r \tag{3}$$

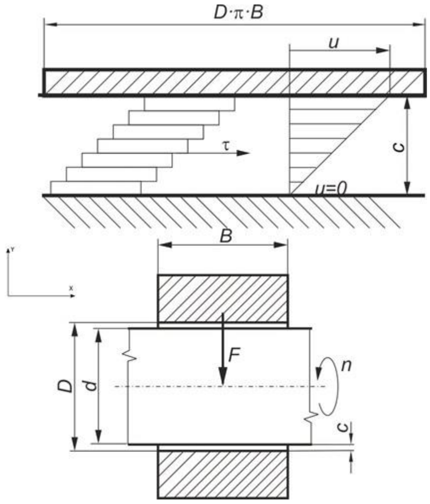


Fig. 1 *Loaded plain journal bearing*

As well:

$$\tau = \frac{F_t}{A} \tag{4}$$

From Eqs. (2), (3) and (4) the frictional force can be derived as:

$$F_t = \tau \cdot A = \frac{\eta \cdot u}{c} \cdot 2 \cdot \pi \cdot r \cdot B = \frac{\eta \cdot \omega \cdot r}{c} \cdot 2 \cdot \pi \cdot r \cdot B \tag{5}$$

The frictional torque is then:

$$M = F_t \cdot r = 2\pi \cdot \frac{r^3 \cdot B}{c} \cdot \eta \cdot \omega \tag{6}$$

And bearing load (normal force) can be expressed in another way:

$$M = \mu \cdot F \cdot r \tag{7}$$

Then Eqs. (6) and (7) give the coefficient of friction and the bearing pressure, respectively, as:

$$\mu = 2 \cdot \pi \cdot \frac{r^2 \cdot B}{c} \cdot \frac{\eta \cdot \omega}{F} \tag{8}$$

$$p = \frac{F}{2 \cdot r \cdot B} \tag{9}$$

The coefficient of friction can also be presented as the Petroff's law:

$$\mu = 2 \cdot \pi \cdot \frac{r^2 \cdot B}{c} \cdot \frac{\eta \cdot \omega}{2 \cdot p \cdot r \cdot B} = \pi \cdot \frac{r}{c} \cdot \frac{\eta \cdot \omega}{p} \tag{10}$$

The first quantity in the br stands for bearing modulus and the second one stands for clearance ratio. Both are dimensionless parameters of the bearing.

3 EXPERIMENTAL DETAILS

3.1. Radial Journal Bearing Wear Test Rig

Bearings materials in journal bearings are generally selected from materials, which have lower wear strength than the shaft material, that way dropping the wearing of the shaft safely. Therefore, journal bearing wear test apparatus are designed to examine the wearing of bearing materials. In Fig. 2 shows the test rig which was modified specifically for this research to use in determining tribological properties of the bearings [19]. Therefore, it is possible to investigate different bearing and shaft materials and the effects of heat treatments on these materials. This test rig is divided into three main systems: Hydraulic Loading System, Rotation System, Lubrication system of the bush as shown in fig (2). Hydraulic Loading System includes rotating electrical machines (REM) type CEM, IEC (0.75KW, 380V, 1420-1745 rpm, 50Hz), which is mounted on the hydraulic tank 20*l* and connected into high-pressure hydraulic flexible tubes which feed hydraulic into the hydraulic cylinder. The hydraulic cylinder is in contact with radial force sensor (U9c/10kN) which is located below it. The testing was done with the oil type ISO VG32. The Rotation System is consists of an asynchronous induction motor (AIM) (ABB, 400V, 3Ph, 50Hz, 3KW, 2860 rpm), which is attached by a flexible coupling to a shaft. The shaft is supported by two roller bearings and the test bearing is mounted between these two bearings as shown as Fig. 2.

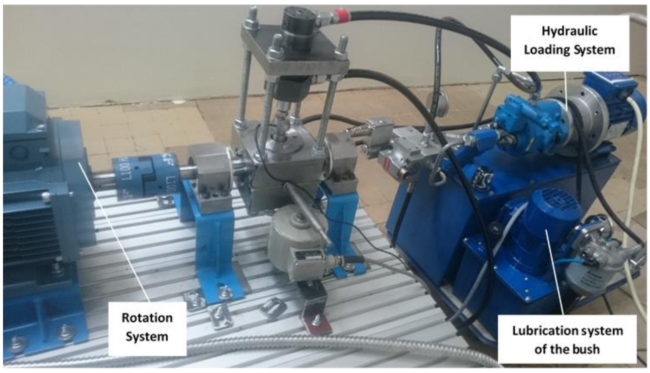
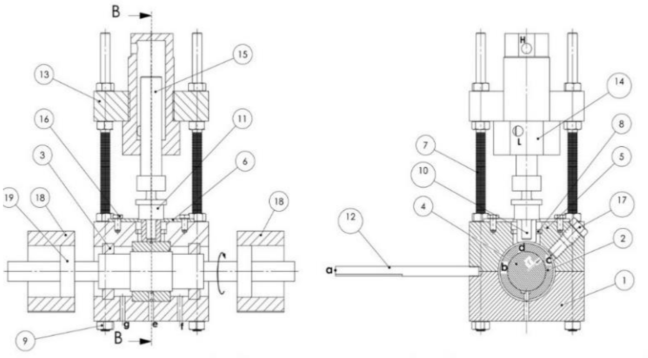


Fig. 2 *View of hydrodynamic journal bearing test rig*

Lubrication system of the bush contains electric lubrication pump (ELP) AMGP-03C, 05C Series (1450rpm, 500W, 220V, 50Hz) which was mounted on the hydraulic tank (10*l*). The lubricant is supplied by the ELP to the housing bearing through two flexible tubes which are threaded up the housing through an assembly steel tube. E540 - Wireless / Point laser was used for shaft alignment. The same type of oil, ISO VG32, was used in the testing.

Fig. 3 shows that the sleeve of the shaft (4) is mounted on the plain bearing (2) which is radially and axially fixed by upper (5) and lower (1) parts of the support. Through the opening of the upper support (5) a force is induced by the hydraulic cylinder and transmitted to the bearing trough radial load



ab	Arm of force transducer	7	Threaded rod M10X700
c	Contact area between the thermocouple and bushing	8	Seal
d	The contact surface between the bush and the radial load supply	9	Hexagon nut ISO 4032-M10-W-N
	Inlet bushing lubricant	10	Radial load supply
f,g	Outlet bushing lubicant	11	Force transducer
H	Inlet hydraulic pressure for radial load	12	Lever (arm)
L	Outlet hydraulic pressure for radial load	13	Cylinder support
1	Lower support	14	cylinder
2	Bush	15	piston
3	Seal shaft	16	Screw ISO 4017-M6-X12-N
4	Shaft	17	Thermocouple
5	Upper support	18	Ball bearing housing
6	Cover seal	19	Ball bearing SKF-6304-8-SI-NC,8_68

Fig. 3 *Schematic of the tool developed for testing of plain journal bearings of configurations*

supply (10). The magnitude of the force measured by the force sensor (11). The circumferential force, which represents the frictional force of the bearing, is measured by means of a lever (12), which is arranged perpendicularly to the axis of the bearing, and a force transducer (Fig. 2). The temperature of the bearing is measured by thermocouple (17). The sleeved thermocouples are placed inside fully drilled holes. The active part of the thermocouple is in contact (region c) with the outer surface of the bush. Upper and lower supports are made of steel; there are also seals (3,8) to prevent oil leakage during the system operation. Upper and lower support were assembled and mounted by standard nuts and a screw (7, 9, 16).

3.2. Preparation of experimental materials and conditions

In the experimental set up used, the shaft was made of AISI 440C stainless steel and bearings were made of tin-based white metal, whose chemical compositions of specimens (WM) is shown in Table 1. Dimensions of bearing specimens were as follows: inner diameter is 40+0.05 mm, width 40 mm, outer diameter 60 mm, the relative bearing clearance 0.025 mm, the thickness of the white metal material 3 mm.

Table 1 Chemical compositions of specimens WM

TEGOTENAX (V840) (big sticks), in TKL NOVA TVORNICA KLIZNIH LEZAJEVA (SLIDING BEARING MANUFACTURING), CROATIA											
WM	Sn	Sb	Cu	As	Bi	Ni	Pb	Cd	Fe	Al	Zn
Actual Value %	88.7	7.6	3.7	0.009	0.002	0.003	0.008	0.010	0.009	0.000	0.002
Lower Limit Value %		7	3.0	-	-	-	-	-	-	-	-
Upper limit Value %		8	4.0	0.060	0.050	0.060	0.060	0.010	0.030	0.005	0.005
The alloy is free of lead and free of cadmium in compliance with RoHS Regulation (EU-Directive 2002/95/EC)											

The mechanical and physical proprieties of bearing are given in Table 2. Tested bearings contain lead, tin, aluminum and copper, these elements are coated to steel foundation due to their superior antifriction properties. These alloys are produced by casting and spray deposition method [20]. The inner part of the bearing, depicted in Fig. 4, is built of white material. The bearing was drilled with hole (dimension r =1.5 mm) in radial direction to facilitate lubrication oil flow into the contact zone. Circumferential groove was also made onto the outer surface of bush (width 2 mm, depth 0.5 mm) to ensurethat lubrication oil arrives into the radial hole. Likewise, the spiral groove is made onto inner surface of sample (width 2 mm, average depth of 30 µm) to improve the lubrication process between the shaft and the bush, as shown in Fig. 4. The surface profiles of the plain journal bearing were determined prior the tests by using the Perthometer SJ-301. After the profiles of plain journal bearing were investigated, it was found that the average roughness height (R_a) of shaft value is 0.32 µm and the average roughness depth is 1.78 µm, while for bearings average height is 0.28 µm and the average roughness depth is 1.87 µm. It means that even the presumably smooth plain journal bearing has some indents and protrusions.

Table 2 Mechanical and physical proprieties of bearing

White metal alloy TEGOTENAX V840			
Hardness	HB 10/250/180 (DIN ISO 4384-2)	20 [°C]	23
		50 [°C]	17
		100 [°C]	10
		150 [°C]	8
Young´s modulus	E	56500 10^6 [Pa]	
Density	ρ	7400 [kg/m³]	
Lower melting point	T_{lm}	233 [°C]	
Poisson ratio	v	0.33	
Upper melting point	T_{hm}	360 [°C]	
Casting temperature	T_c	440 [°C]	

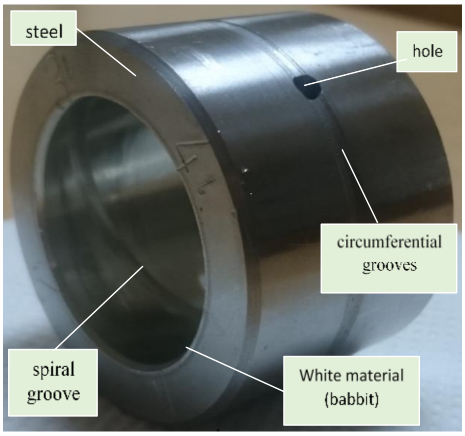


Fig. 4 *Tested plain journal bearing*

A complex Lab VIEW program for data collection and control of the test rig was developed which also enabled the remote control of the test rig. The most common solution is to operate the bearing with no load at start of test rig operation till reach of desired test speed [21]. This type of operation causes the shaft to converge as near to zero eccentricity as possible ie. that shaft and the bearing centers coincide as close as possible.

3.3. Test Specimen and Procedure

The tests were carried out for 5 hours' duration; the operating sequence was ELP, AIM than REP, respectively. Pressure of the lubrication oil sent to the test bearing was 4 bar. Test was conducted to determine time to fill housing of bearing with oil by removing the upper support seal and switch on the lubricating pump. The time it takes to flooding housing is 15 seconds. The objective of this procedure, to ensure the flooding Bushing during the test. The values were always kept under control throughout the tests.
Each test was repeated less than three times to ensure the accuracy of results, take into account the primary oil temperature at each test. of 1/20 s, and the temperature lubrication oil and load for 5 s to be accurate.

3.4. Meassurment of coefficient of friction

Coulomb equations associate load and friction coefficient as a function of time in the touch of two real bodies:

$$F_f(t) = \mu(t) \cdot F_N(t) \tag{11}$$

Moment of friction as a function of time was calculated as the product of the normal force of the sleeve (using reaction force transducer) and the distance on lever between the center of the bush and contact point of the force transducer.

$$M_f(t) = F_s(t) \cdot l_{ab} = F_f(t) \cdot r \tag{12}$$

From (11), (12)

$$\mu(t) = \frac{M_f(t)}{r \cdot F_N(t)} = \frac{F_s(t) \cdot l_{ab}}{r \cdot F_N(t)} \tag{13}$$

as shown in Fig. 5.

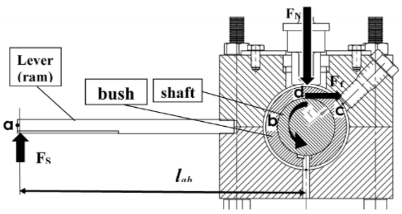


Fig. 5 *Scheme of the frictional force measurement*

4 RESULTS AND DISCUSSION

In order to achieve previously mentioned research objective, numerous experiments were performed under different static loads and rotation speeds. The corresponding curves were then generated by relating the coefficient of friction to the operation time for different loads and rotation speeds. Temperature of oil versus operated time were also plotted for the same deferent loads and rotation speeds for the sake of comparability.

Fig. 7 shows the curves for 1045 *N* average load and rotation speed of 1000 rpm as representing the effect of the lubrication temperature change on the coefficient of friction on the surfaces of the plain journal bearing. It can clearly be seen from Fig. 7 that the coefficient of friction decreased with the operated time, while the generated heat in the bearing causes increase of oil temperature. As mentioned earlier, the coefficient of friction is directly proportional to dynamic viscosity (η) (see eq.10), and the oil temperature rise leads to decrease of the dynamic viscosity which certainly leads up to decrease of the coefficient of friction. Also, the increasing workload causes decrease of the friction coefficient as it was determined that for loads of 2021 N, 3349 N and 3601 N, the average coefficient of friction was 0.024, 0.023, and 0.0136 respectively.

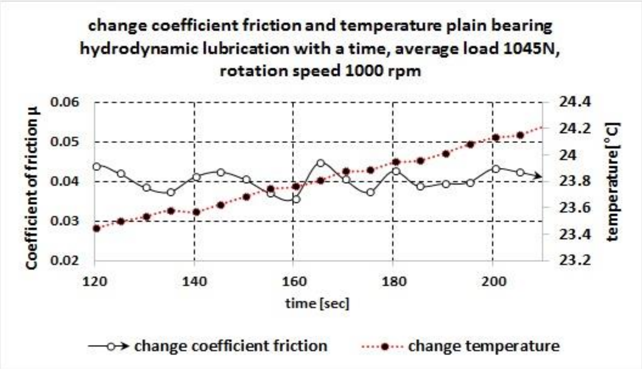


Fig. 7 *Friction behavior and temperature of the plain journal bearing under average load of 1045 N*

As previously mentioned, the coefficient of friction is directly proportional to the rotation speed. Fig. 8 explains the relationship between the frictional coefficient and the radial load for two rotation speeds, 1000 and 2000 rpm, at the start of the operation.

At rotation speed 1000 rpm and 2000 rpm for approximately the same radial load (1045 N vs. 1013 N) the friction coefficient is increasing with the speed increase i.e. for 1000 rpm its value is 0.041 and for 2000 rpm its value is 0.063. The difference between frictional coefficient values for different speeds is expanding with the increase of the load as shown in Fig. 8.

The increase of the frictional coefficient of babbit with the increase of the sliding speed may be due to the change in the shear rate which can influence the viscosity properties of the lubricant materials, i.e., increase in viscosity produces more bearing friction, thereby increasing the forces needed to shear the oil film. In the stable operating condition, the equilibrium conditions are reached in 500-900 sec for all specimens (bushes; the average or equilibrium temperature of the oil would not exceed (38-52 °C), at speed of 1000 rpm, and lubrication pressure 4 bar.

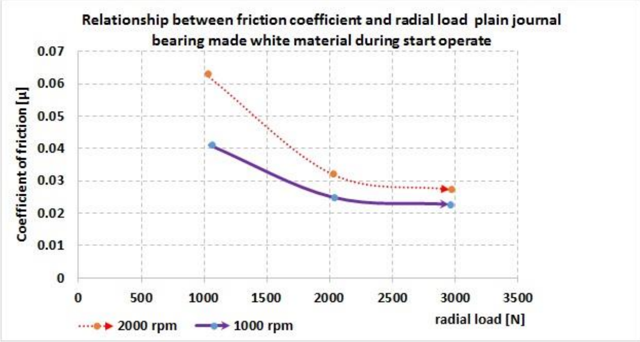


Fig. 8 *Relationship between friction coefficient and bearing load at different rotational speeds during unsteady state*

Figs. 9-12 show the friction coefficient and temperature vs. rotation time curves for plain journal bearing. The frictional coefficients show similar behavior for all specimens (bushes) while maintaining approximately constant average values over rotation speeds, 0.029, 0.023, 0.0149 and 0.012, respectively.

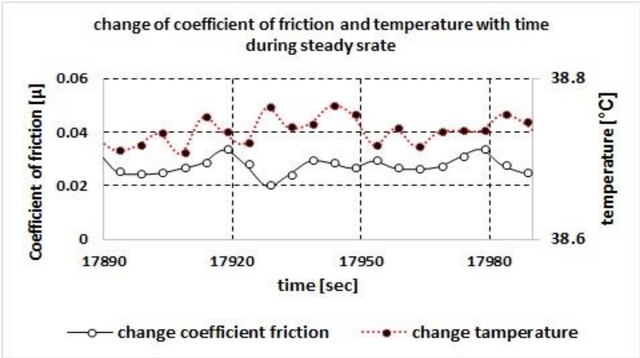


Fig. 9 *Change of the friction coefficient with temperature vs. rotation time in steady-state, load 1045 N, 1000 rpm*

The time required to reach the state of thermal equilibrium depends on radial load, rotation speed, oil viscosity and ambient temperature, whereas that the temperature of the plain journal bearing approximately remains constant at 38.74, 43.65, 41.32 and 40.83 ^{o}C, respectively.

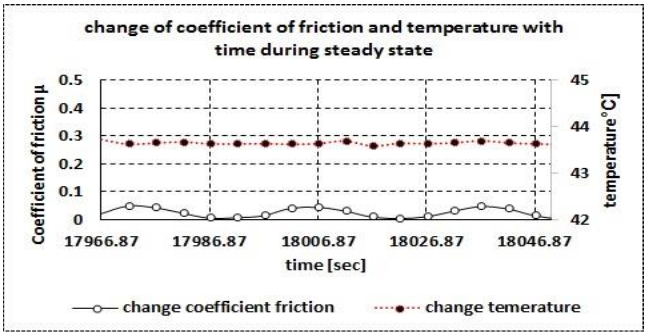


Fig. 10 *Change of the friction coefficient with temperature vs. rotation time in steady-state, load 2021 N, 1000 rpm*

The difference in the coefficients of friction obtained for the starting operation and for the lubrication oil stabilized temperature s against the rotational time is given in noted figures also. It is clear from the graphs that under the same conditions (load, rotation speed, and lubrication oil pump pressure), during start the plain journal bearing exhibits higher coefficients of friction than at the stabilized oil temperature.

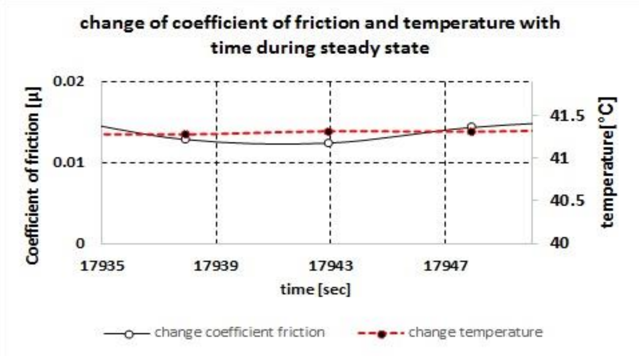


Fig. 11 *Change of the friction coefficient with temperature vs. rotation time in steady-state, load 3349 N, 1000 rpm*

When the bearing loads is 1045, 2021, 3349 and 3601 N, the coefficients of friction bearings are 0.41, 0.024, 0.0237 and 0.0136, whereas they decrease up to 0.29, 0.023, 0.0147 and 0.0125, respectively, once the oil temperature is stabilized.

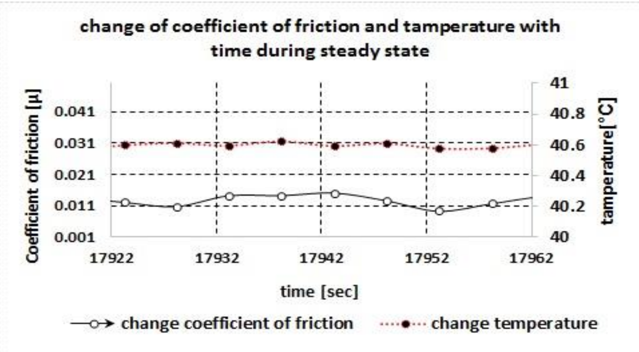


Fig. 12 *Change of the friction coefficient with temperature vs. rotation time in steady-state, load 3601 N, 1000 rpm*

The highest values of friction coefficients occurred during thermally unstable process operation (12000-14000 s) from the beginning of the operation, whereas the lowest friction coefficients occurred for the equilibrium temperature of the plain journal bearing and the lubrication oil, i.e., steady-state operation as shown in Fig. 13. It is obvious from the above given figures that the equilibrium temperature does not depend significantly on the amount of radial load.

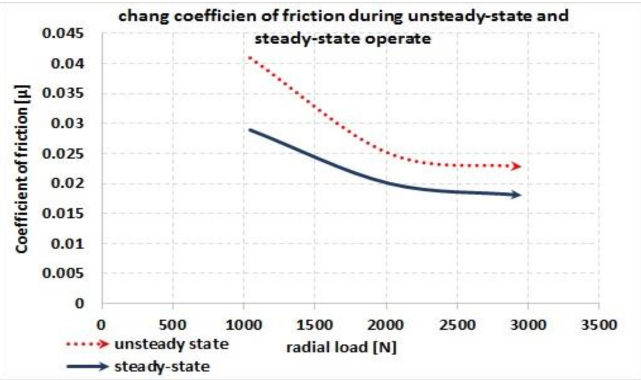


Fig. 13 *Coefficient of friction upon start-up and steady-state operation*

5 CONCLUSIONS

In this research, the frictional behavior of plain journal bearings made from white metal from startup to thermal equilibrium was investigated and compared using a custom-built journal bearing test rig. The work presented in this paper leads to the following conclusions:

- The lubrication temperature of the bearing grows until reach a steady state operation vs sliding time.
- The highest value for the coefficient of friction was obtained in the period of thermally unstable operation (12000 – 14000 s) upon startup. The coefficient of friction value decreased during approach to thermal equilibrium and achieved minimal values in equilibrium operation.
- The friction coefficient of the plain journal bearings decreases with the increase of bearing temperature, as in the theory of the plain journal bearing, due to decrease in viscosity with temperature rise.
- The friction coefficient with respect to rotation speed is getting higher with the increase of rotational speed. This result is consistent with journal bearing theory, as the pressure in oil film is increasing with the increase of rotational speed.

NOMENCLATURE

η	[Pas]	dynamic viscosity of lubricant
u	[m/s]	linear speed lubricant
c	[µm]	bearing clearance
ω	[s^{-1}]	angular speed shaft
D	[m]	diameter bearing
d	[m]	diameter shaft
r	[m]	reduce shaft
B	[m]	width bearing
F_f	[N]	frictional drag force
A	[m^2]	contact are between bush and shaft
F_N	[N]	normal load
μ	-	coefficient of friction
M_f	[Nm]	frictional moment
F_s	[N]	reactional force sensor
R_a	[µm]	average rightness
l_{ab}	[m]	leg distance of lever
t_{av}	[°C]	average temperature
τ	[Pa]	shear stress

REFERENCES

1. Gopinath, K., Mayuram, M., M., *Sliding contact bearings, Indian Institute of Technology Madras*, Machine Design II Indian Institute of Technology Madras.
2. Flack, R.D., Kostrzewsky, G.J., Taylor, D.V., 1993, *A hydrodynamic journal bearing test rig with dynamic measurement capabilities*, Journal of Tribology Transactions, Volume 36 (4), pp. 497–512.
3. Kostrzewsky, G.J., Flack, R.D., 1990, *Accuracy evaluation of experimentally derived dynamic coefficients of fluid film bearings part 1: development of method*, Journal of Tribology Transactions, Volume 33 (1), pp. 105–114.

4. Kasai, M., Fillon, M., Bouyer, J., Jarny, S., 2012, *Influence of lubricants on plain bearing performance: Evaluation of bearing performance with polymer-containing oils*. Journal of Tribology International, Volume 46, pp. 190–199.
5. Childs, D.W., Hale, K., 1994, *A test apparatus and facility to identify the rotor dynamic coefficients of high-speed hydrostatic bearings. Journal of Tribology*, Vol. 116, pp.337–343.
6. Franchek, N.M., Childs, D.W., 1994, *Experimental test results for four highspeed, high-pressure, orifice-compensated hybrid bearings*. Journal of Tribology, Vol. 116, pp.147-153.
7. Sawicki, J., T., Adams, M., L., Capaldi, R., J., 1996, *System identification methods for dynamic testing of fluid-film bearings.* International Journal of Rotating Machinery, 2(4), pp. 237–245,
8. Sawicki, J., T., Capaldi, R., J., Adams, M., L., 1997, *Experimental and theoretical rotordynamic characteristics of a hybrid journal bearing.* Journal of Tribology, 119 (1), pp.132–141.
9. Sawicki, J.T., 1992, *Experimental and theoretical determination of hydrostatic/ hybrid journal bearing rotordynamic coefficients,* PhD thesis, Case Western Reserve University.
10. Mazdrakova, A., Javorova, J., Radulescu, A., Mirev, A., Rakanov, Y., 2016, *Hydrodynamic journal bearing test rig with pressure measurement at elastic deformations of contact surfaces capabilities*, XV International Scientific Conference “RE&IT 2016”, Bulgaria, pp.33-36,
11. Tieu, A.K., Qiu, Z.L., 1994, *Identification of sixteen dynamic coefficients of two journal bearings from experimental unbalance responses.* Journal of Wear, 177, pp. 63–69.
12. Tieu, A.K., Qiu, Z.L., 1996, *Experimental study of freely alignable journal bearings – part 1: static characteristics.* Journal of Tribology, 118, pp. 498–502.
13. Qiu, Z.L., Tieu, A.K., 1996, *Experimental study of freely alignable journal bearings – part 2: dynamic characteristics.* Journal of Tribology, 118, pp. 503–508.
14. Brito, F.P., Miranda, A.S., Bouyer, J., Fillon, M., 2006, *Experimental Investigation of the Influence of Supply Temperature and Supply Pressure on the Performance of a Two-Axial Groove Hydrodynamic Journal Bearing*, Journal of Tribology 129 (1), pp. 98-105.
15. Javorova, J., Mazdrakova, A., Andonov, I., Radulescu, A., 2016, *Analysis of HD journal bearings considering elastic deformation and non-Newtonian Rabinowitsch fluid model*, Tribology in Industry, 38 (2), pp. 186-196.
16. Javorova, J., 2014, *Numerical investigation on the performance of HD journal bearings under elasticity contact conditions*, 8th International Conference on Tribology BALKANTRIB’14, Romania, pp. 184-189.
17. Hairong, W., Qinling, B., Shengyu, Z., Jun, Y., Weimin, L., 2011, *Friction and wear properties of Babbitt alloy 16-16-2 under sea water environment*, Journal of Tribology International, 44 (10), pp.1161–1167.
18. Zeren, A., Feyzullahoglu, E., Zeren, M., 2007. *A study on tribological behaviour of tin-based bearing material in dry sliding,* Journal of Materials & Design, Vol. 28, Issue 1, pp. 318–323.
19. Bojić, N., Milčić, D., Banić, M., 2015, *Effect of coverage of graphite on self- lubricating plain bearings*, Serbiantrib ’15, May 13-15, Belgrade, pp. 309-313.
20. D. Dowson, 1998, *History of Tribology*, Professional Engineering Publishing, pp.768.
21. Brockwell, K., Decamillo, S., Dmochowski, W., 2001, *Measured temperature characteristics of 152 mm diameter pivoted shoe journal bearings with flooded lubrication*, Tribology Transactions, 44 (4), pp.543–550.

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