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TRIBOLOGICAL BEHAVIOUR OF TIN-BASED MATERIALS – TEGOTENAX V840 IN OIL LUBRICATED CONDITIONS

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Abstract: The present paper investigates experimentally the effect of sliding distance, rotation speed of shaft and normal load on friction, surface roughness special position on bearing inner surface and wear property of radial plain bearing made of tin-based white metal alloy - TEGOTENAX v840 under lubricated contact sliding against stainless steel shaft, to do that, a test rig apparatus was designed and fabricated. Experiments were carried out in conditions of different loads and different speed. Results show that the wear rates increase with the increase of sliding speed and normal load. It is also found that friction coefficient increases with the increase of sliding speed and decreases with increase normal load. It is also found that the roughness change with sliding distance in special position bearing inner surface.
Keywords: tin-based materials, white metal alloy, Tegotenax V840, oil lubricated conditions

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]:

- Based on type of load carried
- Based on type of lubrication
- Based on lubrication mechanism

Study of mechanics of friction and the relationship between friction and wear dates back to the sixteenth century, almost instantly after the devise of Newton's law of motion. It was observed by several authors [2], [3], that the variation of friction and wear rate depends on interfacial conditions such as normal load, geometry, relative surface motion, sliding speed, surface roughness of the rubbing surfaces, type of material, system rigidity, temperature, stick slip, relative humidity, lubrication and vibration. Through these factors sliding speed and normal load are the two main factors whose play significant role for the variation of friction and wear rate.

In this paper, the journal bearings test rig [4] was modified specifically for this research to use in determining tribological properties of the bearings, as shown in Figure 1, and experimental research to study performance of the radial sliding bearing were coated with tin alloys - TEGOTENAX V840.

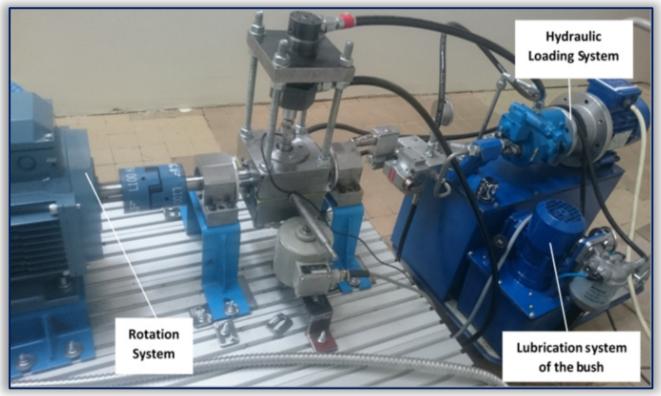


Figure 1. Illustration of journal bearing test rig

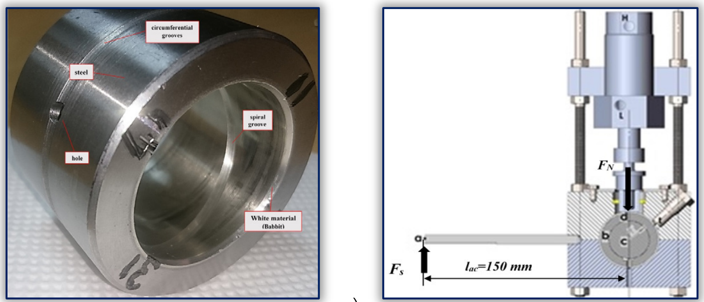
2. RADIAL JOURNAL BEARING TEST RIG

This test rig is divided into three main systems: Hydraulic Loading System, drive system, Lubrication system of the bush as shown in Fig. 1. The drive system drove the shaft. The driving system test rig consists of an asynchronous induction motor (AIM) (ABB, 400 V, 50 Hz, 3 KW, 1460 rpm). E720 - Wireless / Point laser was used to align shaft with the motor. The lubrication system supplied the lubricating oil for the bearing. Lubrication system contains of electric motor-pump assembly (ELP) (motor:

1450 rpm, 90 W, 220 V, 50 Hz; pump AMGP-03C) which was mounted on the 10 l hydraulic tank. The same type of oil, ISO VG32, was used for bush lubrication. Bearing loading system consists of electrical motor-pump assembly (REM) (CEM motor - 0.75 KW, 380 V, 1420 rpm, 50Hz; EATON pump PVQ10) which is mounted on the 30 *l* tank.

3. EXPERIMENTAL DETAILS AND CONDITIONS

In this study, radial plain bearing specimens made of tin-based alloy -TEGOTENAX V840, were used as journal bearing and AISI 440C stainless steel was used as shaft. 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. The bearing was drilled with hole (dimension r =1.5 mm) in radial direction to ease 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 include that lubrication oil arrives into the radial hole. as well, 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 Figure 2a The specimens were robbed by radial journal bearing wear test rig under lubricated condition. The tribological parameter were measured under lubricated conditions of (0.63, 0.94, 1.25) MPa average apparent prassure, (1000 and 1500) rpm sliding speed, every 1 hour for change coefficient friction, 5 houres for wear losses each specimen. The lubrication was achieved by using ISO VG32 oil. Before and after testing, the specimens were cleaned by cleaner FLUXO S190 and dried using cotton and convection oven at 60°C for a period of half an hour to remove humidity, as the weight of the samples was measured using a digital balance and recording the values before and after the test for each test. 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. Each test was repeated less than three times to ensure the accuracy of results, take into account the primary oil temperature at each test.



a) b)

Figure 2. Tested plain journal bearing

4. COEFFICIENT OF FRICTION MEASUREMENT

In the Figure 2b shown measurement procedure used in this research to determine coefficient of friction of the sliding bearing. The measurement method friction coefficient is based on the principle of torque balance resulting from the friction force and torque produced by the force of the sensor reaction. The temperature change and the values of the radial force on the bush were calculated by LabVIEW program which is installed into the computer. The data was transmitted to the recorder and this enabled monitoring and recording of the signals. The data about coefficient friction, radial load and sensor force were applied for 1/20 s and the temperature lubrication oil and load for 5 s.

5. RESULTS AND DISCUSSION

To realize this objective, many experiments were carried out under different apparent pressure and rotation speeds. In Figure 3 is model for the change of temperature versus time of steady and unsteady state process under steady apparent pressure 0.63 Mpa and sliding speed 1000 rpm for testing 5 hours was investigated to show steady and unsteady process zone.

We can distinguish from the Figure 3 two zones, unsteady and steady state zones. Under unsteady state zone the temperature within the system does crease with time, in Figure 3 is estimated unsteady state approximated (0-3500)s. second zone is steady state zone in which temperature remains approximately constant, it's here about 39°C.

The viscosity of the oil affected by the temperature, it decreases with the growth of temperature, and the coefficient of friction is directly proportional to the viscosity, so the coefficient of friction decreases in the steady state zone and remains almost stable in the steady state zone as shown in Figure 3, The average value of the coefficient of friction under apparent pressure 0.63 MPa rotation speed was 0.058, The following table shows the mean value of the friction coefficient under operating conditions. Clear from the Table 1, at rotation speed 1000 rpm and 1500 rpm for the same radial load 0.63 Mpa, the friction coefficient is increasing with the roration speed increase. The difference between frictional coefficient values for different speeds is expanding with the increase of the load as shown in table 1.

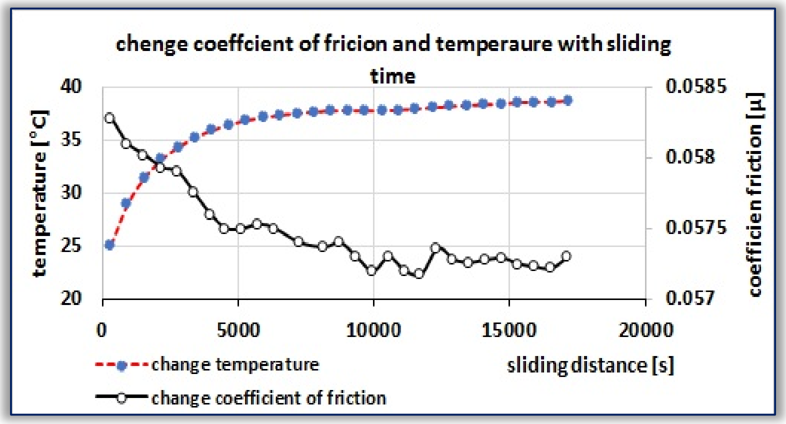


Figure 3. Change temperature and coefficient of friction with time rotating plain journal bearing under steady load 0.63 Mpa, 1000 rpm, 5 hours

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. It is also clear from the table that the duration of access to steady state increases with increasing load and speed of rotation, this is due to delayed access to the state of thermal balance between sliding parts and oil. The tin-based bearing material in lubricant sliding condition at different average apparent pressure 0.63, 0.94 and 1.25 MPa vs. sliding distance (0-453600) m, and variation rotation speed of shaft 1000 and 1500 rpm were investigated. In Figure 4 shows two unsteady-state zones and two steady-state zones of change loss material influence changes in process conditions. First unsteady-state zone has Sharply wear increases vs sliding distance for all average apparent pressures (0.63, 0.94 and 1.25) MPa, value loss material about 0.001, 0.00123 and 0.0015 mg respectively, operating distance 37800 m. Follow first unsteady state loss material a first steady state zone, long distance of first steady-state zone is 113400 m, average loss material in first steady-state wear process was 0.0011, 0.0013 and 0.0017 mg under 0.63, 0.94. and 1.25 MPa apparent pressure respectively, and rotation speed 1000 rpm. The cause of rapid loss of material at the beginning of operation i.e. first unsteady-state may be due to get to the case of compatibility between shaft and bush.

Table 1. Unsteady state of temperature and change coefficient of friction with loads and rotation speed

Rotation speed [rpm]	Apparent pressure [MP]	Average coefficient of friction μ [-]	Primary temperature T_0 [°C]	The final temperature T_{end} [°C]	Duration of unstable state [s]
1000	0.63	0.057	24.8	38.7	0-3500
	0.94	0.034	27.3	43.6	0-3630
	1.25	0.028	22.02	41.7	0-3750
1500	0.63	0.038	26.6	41.68	0-4200
	0.94	0.027	27.2	47.9	0-4300
	1.25	0.022	33.1	52.1	0-4500

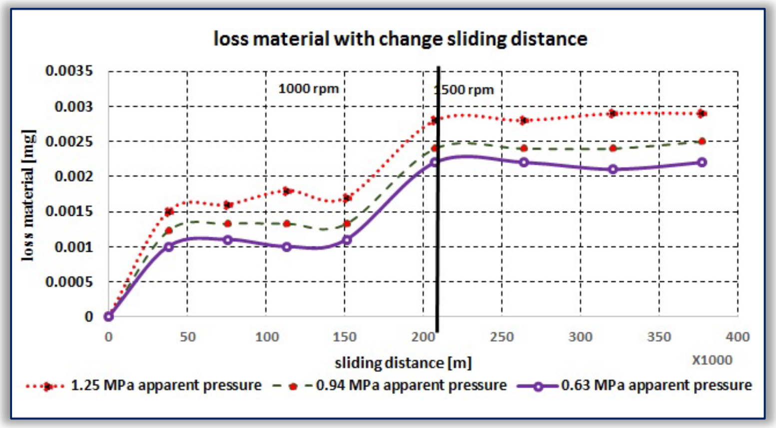


Figure 4. Relationship between weight loss and sliding distance

When rotation speed changes from 1000 to 1500 rpm, the second unsteady-state zone wear accurse in process for 56520 m sliding distance, value loss material about 0.0022, 0.0024, 0.0028 mg under 0.63, 0.94 and 1.25 MPa average apparent pressure respectively, followed by second steady-state wear process zone, Long distance of second steady-state zone is 377392 m an average loss material in second steady-state wear process 0.0022, 0.024 and 0.0029, mg under 0.63, 0.94 and 1.25 MPa average apparent pressure respectively.

6. CONCLUSIONS

Tribological properties in conditions of sliding distance contact can vary over many orders of magnitude. In this study, of tin-based materials -TEGOTENAX V840 in oil lubricated conditions was evaluated and focusing on the effect of sliding distance in change of loss material and coefficient of friction surface contact of tin-based journal bearing. The following conclusions can be drawn:

» The highest wear loss 0.0011mg in steady-state zones occurred in apparent pressure 1.25 and rotation speed 1500 rpm at sliding distance 377392 m, under similar tribological conditions., wear decreases as decease apparent pressure to lowest wear loss material 0.0001 mg happened in load 0.063 MPa. That mean wear loss tin-based bearing material directly proportional to increase the apparent pressure and sliding distance.
» The highest wear loss 0.0015 mg in unsteady-state zones occurred in load 1.25 Mpa and rotation speed 1000 rpm at sliding distance 37800 m i.e. at the beginning of the test..
» The duration of access to unsteady state increases with increasing load and speed of rotation, this is due to delayed access to the state of thermal balance between sliding parts and oil.
» Start-up of experemintal process the temperature and the friction coefficient are in a unsteady state phase, the period of unsteady state depends on the load conditions i.e. apparent pressure, rotation speed and primary temperature. Then the temperature and the coefficient of friction remain in the process of steady state almost.

Note

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