Friction and Wear Properties of Cu and Fe-based P/M Bearing Materials

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The performances of porous bearings under different operating conditions were experimentally investigated in this study. Material groups studied are 90%Cu+10%Sn bronze and 1%C+% balance Fe iron-based self-lubricating P/M bearings at constant (85%) density. In the experiments, the variation of the coefficient of friction and wear ratio of those two different group materials for different sliding speeds, loads, and temperatures were investigated. As a result, the variation of the friction coefficient - temperature for both constant load, and constant sliding speed, friction coefficient - average bearing pressure, PV - wear loss and temperature-wear loss curves were plotted and compared with each other for two materials, separately. The test results showed that Cu-based bearings have better friction and wear properties than Fe-based bearings.

Key Words: Powder Metallurgy, Self-Lubricating Bearings, Friction, Wear, Temperature

1. Introduction

Sintered-metal self-lubricating bearings are based on powder-metallurgy technology. They are economical, suitable for high production rates and can be manufactured to precision tolerances. Sintered-metal self-lubricating bearings are widely used in home appliances, small motors, machine tools, aircraft and automotive accessories, business machines, instruments and farm and construction equipment (http://www.qbcbearings.com/B610/B610Cat.htm#Alpha).

Self-lubricating porous bearings have the advantage of reducing the need for certain lubricating equipments (oil pipes, pumps, etc.) as well as reducing other problems related to lubrication mechanism. One of the advantageous features of the porous bearings is that no external supply of lubricant is required for running-in (Naduvinamani et al., 2005).

Bronze : The most common porous bearing material. It contains 90% copper and 10% tin. These bearings are wear-resistant, ductile, conformable, and corrosion-resistant. Their lubricity, imbeddability and low cost give them a wide range of applications from home appliances to farm machinery.

Iron : Combination of low cost with good bearing qualities, widely used in automotive applications, toys, farm equipment, and machine tools. Powdered-iron is frequently blended with up to 10% copper to improve the strength. These materials have a relatively low limiting value of PV (on the V side), but have high oil-volume capacity because of the high porosity. They have good resistance to wear, but should be used with hardened and ground steel shafts (http://www.qbcbearings.Com/B610/B610Cat.htm#Alpha). Powder mixtures represent the easiest way of obtaining an alloy as the users of powder are being able to produce the desired alloy and modify it ac-

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Table 1 Experiment parameters

Inside diameter of test bearing	12.013 mm
Outside diameter of test bearing	15 mm
Diameter of test journal	12 mm
Width of test bearing	12 mm
Density (g/cc)	7
Relative bearing clearance	0.108%
Porosity (mean)	15%
Oil content	~13%
Environmental temperatures (℃)	25±2

cording to their own needs. The iron powders have a special importance in used mixtures for achievement of a product with good mechanical properties. These powders have good plasticity, providing good compressibility (resulting in high density) and a good strength for the obtained untreated pressed part, in the pressing process. (Ciupitu et al., 2003).

In the study, the tribological properties of porous Cu-based and Fe-based bearings manufactured with P/M under dynamic loading at different PV values were experimentally studied.

2. Experiment

2.1 Materials

Commercial Cu and Fe-based P/M bearings. were used in the tests. The surface of journal was prepared by cylindrical grinding and had N4 surface finish quality and its average surface roughness (Ra) was 0.305 μm. The material of the test shaft was SAE 1050 hardened steel (HRC 55). The journal bearings had nearly N4 surface finish quality with an average surface roughness Ra= 0.940 μm, and after test Ra=0.360 μm for Cu-based, Ra=0.900 μm, and after test Ra=0.540 μm for Fe-based. The hardness of the test bearing was HRB 33 for Cu-based 46 HRB. Material groups studied are 90%Cu+10%Sn bronze and 1%C+% balance Fe iron-based self-lubricating P/M bearings at constant (85%) density. These bearings are, nowadays, mostly used in the automobile components. The properties of journal bearing and experimental conditions were summarized in Table 1. Fig. 1 shows a view of the test bearings surface.

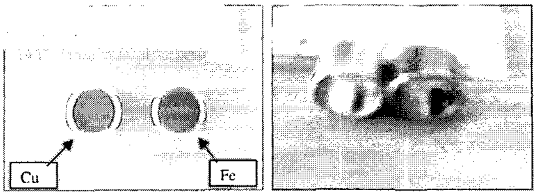


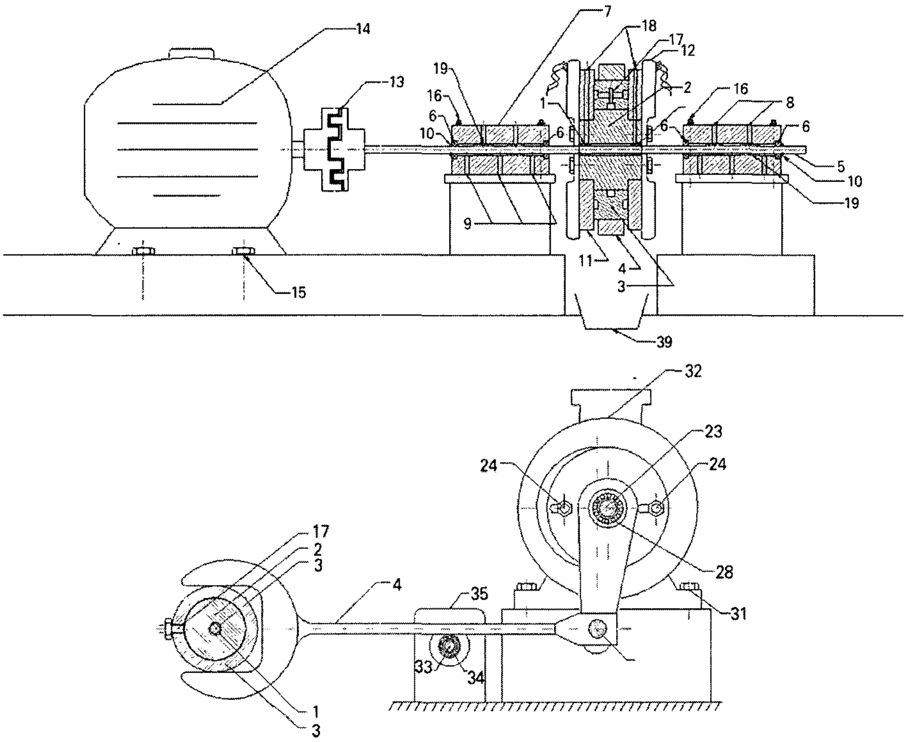
Fig. 1 Views of test bearings surface

2.2 The test rig

Test rig which was fabricated at Mechanical Engineering Laboratory of Suleyman Demirel University (Durak, 2003; Tüfekçi, 2003; Durak et al., 2002; Kurbanoğlu et al., 2002) was used to measure the friction force in the porous bearing under various loading conditions such as static, dynamic loading, especially vertical periodic loads. The test rig is schematically shown in the Fig. 2 and views of the test rig are shown in Fig. 3. It consists of a 1.5 kW, 1500 rpm speed D.C. motor (14) driving the test journal (5) through an elastic-coupling (13). The test speed can be adjusted by frequency changer. The test porous bearing (1) was fitted in the housing (2) with a very light press fit using a specially designed fixture.

An eccentric disc and lever (4) apply vertical periodic loads to the test porous bearing (1). The eccentric disc is driven by an A.C. motor (32) (1.1 kW, 1500 rpm). The frictional torque between the journal (5) and the test bearing (1) had to be measured independently of other torques (Cusano and Phelan, 1973). To make this possible, the load was transmitted vertically down through squeeze oil film that was formed between journal bearing housing (2) and the loading circle (3).

When the test journal (5) is rotated, the friction torque on the bearing will try to rotate the bearing housing (2). Figure 4 shows the detail of the test rig measuring frictional torque. The friction transducer measures the reaction force due to forming this torque. The friction transducer consists strain - gauges (42) in a Wheatstone bridge configuration. The force and the coefficient of



1-Test P/M bearing 2-Bearing Housing 3-Loading ring 4-Lever of the loading
5-Journal (shaft) 6-O-ring 7-Housing of support bearings
8-Entry exit of oil supplier of support bearing 9-Exit of oil supplier of support bearing
10-Segment 11-Discs of the friction force measurement 12-Heaters
13-Flexible coupling 14-Electricity motor 15-Bolts (M14) 16-Bolts (M6)
17-Oil channel of squeeze oil film 18-Chanells of the temperature measurement
19-Bearings of the supporting 23-Pin of the loading 24-Eccentric adjustment bolts 26-Pin
28-Rolling bearing 31-Bolts (M16) 32-Electrcity motor (for loading)
33-Pin 34- Rolling bearing 35-Supports loading lever
36-Resistans of the heat 39-Oil cup

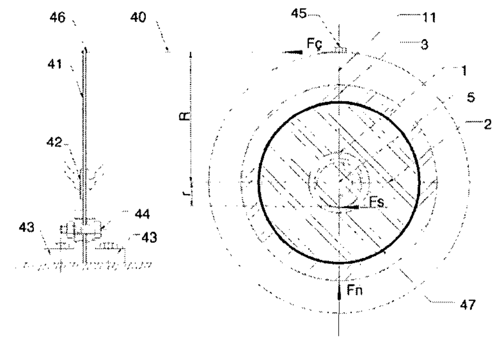
Fig. 2 Cross sections of the test rig

Fig. 3 Views of the test rig

friction were recorded simultaneously with respect to time using a computer-based data acquisition system.

2.3 Experimental procedure

Figure 5 shows the selected load patterns for this experimental study. The test bearing was



1-Test P/M bearing 2-Bearing Housing 3-Loading ring
4-Lever of the loading 5-Journal (shaft) 11-Lever of pull
40-Lever of pull 41-Measurement plate 42-Strain-gauges
43-Supports 44-Bolts 45-Bolts
46-Screw 47-Squezz oil film

Fig. 4 Drawing detail of measuring of frictional torque in the test bearing

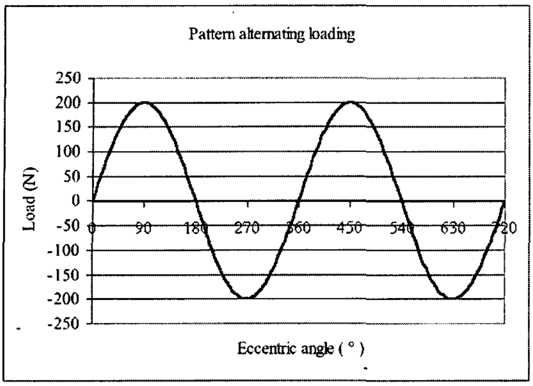


Fig. 5 Test load patterns

subjected to a running-in process initially. A few drops of the same oil as impregnated in the test bearing were placed on the journal surface enabling the flushing out of wear debris in the tests. Before beginning the experiments, the test bearings were run for 5 minutes in order to make smoother surfaces and to flush out of wear debris during the running-in process under low speed and low load conditions.

Tests were carried out under 20℃, 40℃, 60℃, 80℃, 100℃ temperatures, at 100, 300, 600, 1000, 1400 rpm speeds and at ±50 N, ±100 N, ±200 N, ±400 N, ±600 N loads. Thus, each test bearing was carried out experiment conditions such as five different speeds, five different loads, and five different temperatures. In this study, 125 total tests were carried out. The tests were carried out until the sliding distance would be 3 km. In every 5 seconds, friction force was measured and recorded by the computer. Friction coefficients for each set of experimental conditions are mean values of three tests with deviations lower than 10%. The average data were taken into account to plot the results. All the bearing specimens were cleaned with hexane and then dried before impregnated process and after tests for determining of wear amount.

3. Results and Discussions

Durak (2003) concluded that the experimental results obtained in his study indicated that the correct selection of lubricant and suitable running conditions is very important on tribological properties of porous bearings. Lubrication performance characteristics (load capacity, the friction coefficient, wear, bearing temperature, etc.) of porous oil bearings were reported by several researchers. However, a direct comparison of data among available reports was not always of much significance due to lack of normalization for operation conditions, bearing parameters. Thus, no systematic understanding for performance characteristics of porous oil bearing was gained through analysis of available experimental data. Actually, mode of lubrication realized in porous oil bearings scatters widely ranging from hydrodynamic lubrication to mixed lubrication and boundary lubrication. Further, there are more factors influencing the lubrication performance of porous oil bearing (oil content in bearing pores, permeability of the porous bearing body, porosity of the bearing body, chemical and physical properties of impregnated oil, etc.) than those of common solid (sliding) bearing (Kaneko, 1993).

Surface pressure of 1 MPa and sliding velocity of 100 m/minute are the standard service conditions of oil-containing sintered metallic bearing. In the range of surface pressure higher than 1 MPa and sliding velocity slower than 100 m/minute, lubrication layer over the porous sliding surface would become unstable and, in the range of sliding velocity appreciably faster than 100 m/minute, undesired consumption of lubricant due to friction heat and instability for friction coefficient might be caused because of inherently the imperfect nature of the fluid lubrication in this type of bearing. In any event, manufacturers of self-lubricating bearings invested their efforts to develop advanced materials to cope with service conditions severer than the standard ones (Kasahara, 1997). Cusano and Phelan (Cusano and Phelan, 1973; Raman, 1998) conducted experiments simulating the practical conditions by not supplying additional oil and found that oil stored inside the bearing alone was not sufficient to provide hydrodynamic lubrication conditions when PV values were as high as 105 MPa.m/minute; bearings could work only in boundary lubrications. However, at lower PV values in the order of 69

MPa.m/minute, they found that bearings could work in hydrodynamic condition. Also, the limiting values of PV for porous bearing in the boundary lubrication conditions were given as 110 MPa.m/minute (for speed 45—60 m/minute) (Cusano, 1997). Also, it was clearly seen that PV value used in this study was the PV value of permissible. The calculated PV values as according to test loads and speeds were given in Table 2.

The coefficient of friction (μ) versus temperature in the different PV values was shown at Fig. 6 for Cu-based bearing, and at Fig. 7 for Fe-based bearing.

The test speed and loads carried out in Fig. 6 were given as 100 rpm and 50, 100, 200, 400, and 600 N. PV was variable because of the different loads, but speed was constant. Thus, the effect of average bearing pressure (load-supporting capacity) to the friction coefficient (μ) was investigated. Minimum μ was measured for all the PV values at 40℃ temperature in the Cu-based bearing. By increasing temperature, at first μ decreased (especially at lower PV values) suddenly, and then it increased again. Minimum μ for this test bearing and test conditions was found the most suitable condition at 40℃ temperature. Also, with the increase of average bearing pressure, variations of the μ versus temperature reduced.

Figure 7 shows the results of μ for the test conditions in Fig. 6 for Fe-based bearing. As shown in Fig. 7, while Fe-based bearing shown similar behaviors in the lower PV values as Cu-based bearing, in the higher PV values, the effect of the temperatures were not similar behaviors in the μ. Minimum μ in the Fe-based test bearing was obtained at 40～60℃ temperatures. The effect

Table 2 PV [MPa m/minute] values for Cu and Fe-based P/M bearing

Load → / Speed of the journal ↓	F=50 [N]	F=100 [N]	F=200[N]	F=400 [N]	F=600 [N]
n=100 [rpm]	0.65	1.3	2.6	5.2	7.8
n=300 [rpm]	1.95	3.9	7.9	15.9	23.7
n=600 [rpm]	3.9	7.8	15.6	31.2	46.8
n=1000 [rpm]	6.5	13	26	52	78
n=1400 [rpm]	9.1	18.2	36.4	72.8	109.2

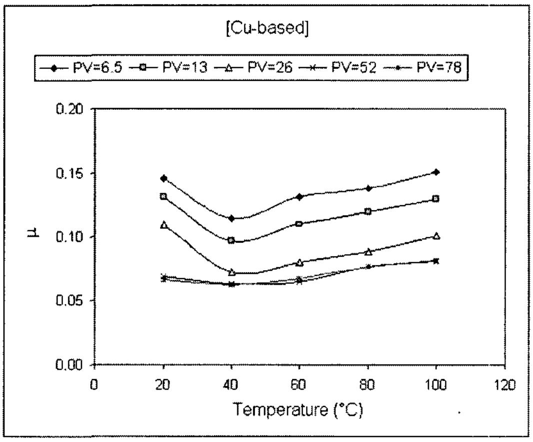


Fig. 6 Coefficient of friction as a function temperature at different PV for Cu-based bearing (1000 rpm constant speed, 50, 100, 200, 400, and 600 N loads) values

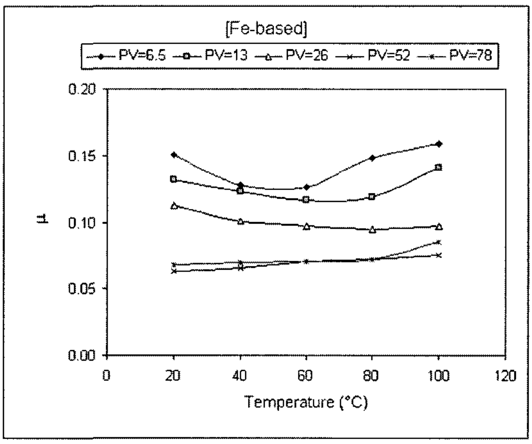


Fig. 7 Coefficient of friction as a function temperature at different PV for Fe-based bearing (1000 rpm constant speed, 50, 100, 200, 400, and 600 N loads) values

of the temperature was not in the higher PV values for Fe-based test bearing.

Figures 8 and 9 were given test results showing the effect of the speed to variable PV. In Fig. 8 was shown variations of μ versus temperature at 400 N load, speed of the journal such as 100, 300, 600, 1000, and 1400 rpm. It is noted that the test results were obtained at first μ decreases with increasing temperature and then it increases again and nearly go constant to last temperature. Variations of μ versus temperature were very small as Fe-based bearing when compared with Cu-based bearing. But minimum μ was obtained at 40℃ temperature again.

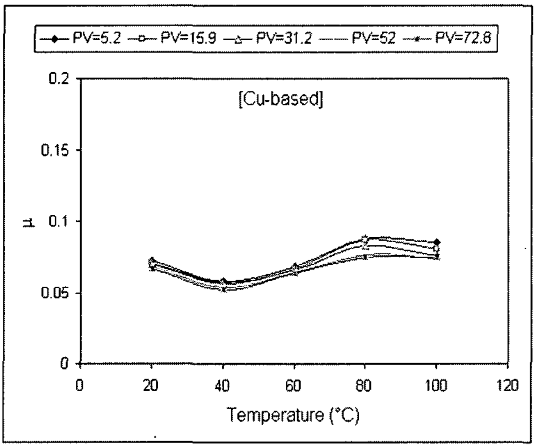


Fig. 8 Coefficient of friction as a function temperature at different PV values for Cu-based bearing (at 400 N constant load, 100, 300, 600, 1000 and 1400 rpm speeds) values

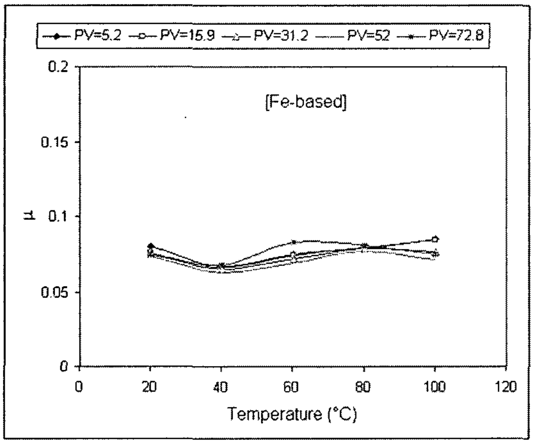


Fig. 9 Coefficient of friction as a function temperature at different PV values for Fe-based bearing (at 400 N constant load, 100, 300, 600, 1000 and 1400 rpm speeds) values

In the lower loads, with the increase of the temperature to a certain point, the amount of lubricant in the test bearing pores come out of the pores easily with the decrease of viscosity of the lubricant. Thus, a partially lubricating film begins forming between the journal and the test bearing (Durak, 2003). In these conditions, a decrease in the μ was observed. At higher loads, with the increase of temperature, lubricant viscosity reduces, and it reduces the load supported capacity, and the μ increases.

Friction forces increase with the increase of the bearing pressure (Figs. 6 and 7). The increasing ratio in the friction force was lower than the increasing ratio in the bearing pressure. Thus, μ was reduced with the increase average bearing pressure. This reducing continued to a certain point and then friction coefficient increased again (Figs. 10 and 11). The lubricant film can be formed with the amount of lubricant in the test bearing pores come out of the pores easily with the increase average bearing pressure. Therefore, the friction coefficient may reduce with the increase average bearing pressure.

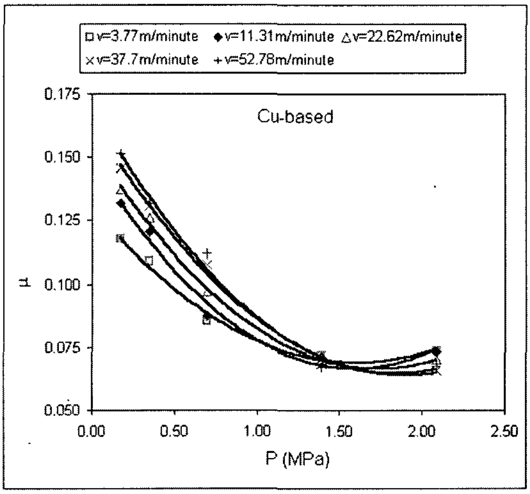


Fig. 10 Coefficient of friction versus average bearing pressure (P) for Cu-based bearing

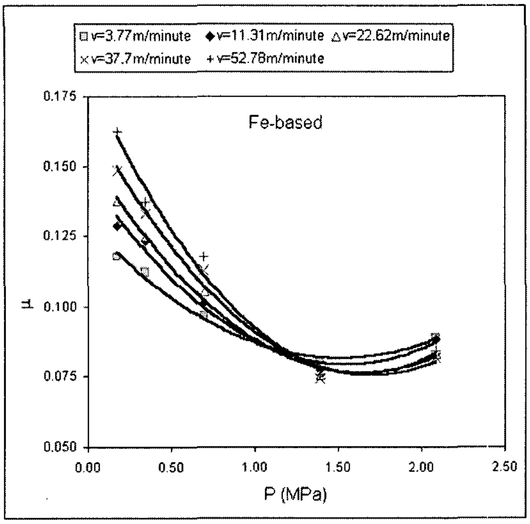


Fig. 11 Coefficient of friction versus average bearing pressure (P) for Fe-based bearing

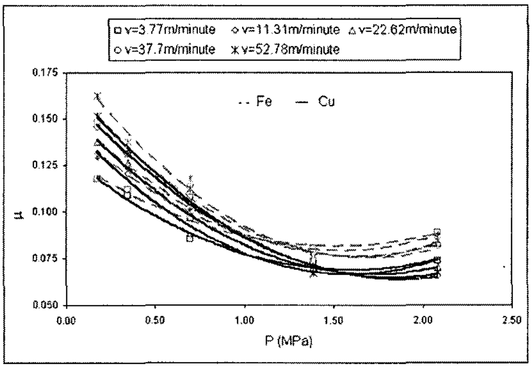


Fig. 12 Coefficient of friction versus average bearing pressure (P) for Fe and Cu-based bearing

All the μ versus average bearing pressure graphics have same variation (Figs. 10 and 11). These test results were described as an optimum bearing pressure for sintered bearing. Although this value is 1.25~1.75 MPa for Cu-based test bearing, this value can be changed by circumferential speed. Cusano and Phelan (1973) concluded that the coefficient of friction is relatively higher at lower loads ; it decreases to a minimum, and then stars an upward trend as the load is further increased. Also, they concluded that the coefficient of friction was found to vary with load and to be almost independent of speed for the bearings tested under boundary lubrication condition. Therefore, our test results were confirmed experimental results obtained by Cusano and Phelan as well.

In the graphics of μ versus bearing pressure, μ was almost the same both at Fe-based bearings and at Cu-based bearings at the lower load. But at higher loads, μ was different between Fe-based and Cu-based bearings (Fig. 12).

The peaks of surface roughness in the loaded part of the porous bearing will be worn away by the rotating journal. When two surfaces come into the sliding contact, the friction heating produced at the interface can cause rapid temperature rises. The temperature rises may cause surface oxidation, peak deformation and thermomechanical wear during process (Horng et al., 2002). Fu et al. showed in their paper (Fu et al., 1998) that there was a sudden increase in coefficient of friction at the initial stage and this was attributed to the breakage of the mechanical bonds or local welding between the roughness peaks of two counterfaces. In addition, Andersson et al.(1996) assumed that the sliding surfaces tested in a lubricated environmental become polished during running-in, they also observed that the lubrication mechanism was transformed from boundary or mixed lubrication to full film lubrication. Besides, Kaneko (1993), explained that two effect had considered to be responsible for oil film formation in the clearance of porous oil bearing; one was oil feed effect known as pumping effect arising from oil film pressure in the clearance and another is oil seepage from pores to bearing surface by reduced oil viscosity and by increased oil volume due to thermal expansion induced by frictioning heat generated at the bearing sliding surface.

Wear tests were performed within the sliding distance of 3 km the duration of the test according to journal speeds was different. Wear ratios were determined loss weight wear. At Fig. 13. Cu-based and Fe-based bearings at room temperature in the different PV values were shown. While Fe-based bearing at the low PV values performed good wear resistance, with the increase of PV this resistance reduced, on the other hand, Cu-based bearing remained constant nearly at different PV values. The similar state was observed in the test

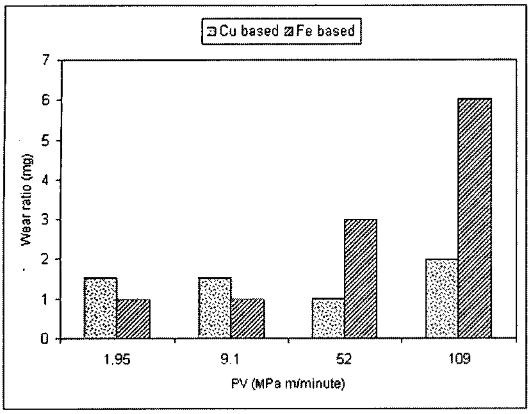


Fig. 13 Experimental wear loss for Cu-based and Fe-based bearings in the different PV values at room temperature for 3 km sliding distance

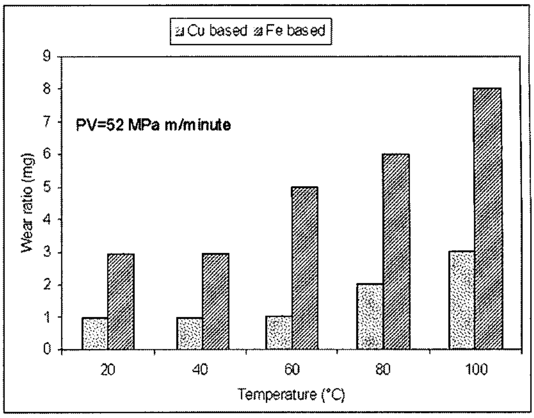


Fig. 14 Experimental wear loss for Cu-based and Fe-based bearings at PV=52 MPa m/minute values at different temperatures for 3 km sliding distance

as shown Fig. 14 with constant PV and different temperature. Fu et al. expressed that (Fu et al., 1998), this is due to the formation of a coherent oil film between the surfaces of the journal and bearing, which prevents the actual contact between the two surfaces. As a result, the adhesion, plastic and shear deformation of the asperities are significantly reduced, leading to decrease in the material removed. The tests show that, for low PV values, an oil film can be formed in porous bearings which are lubricated only by the lubricant within their structure. Thus, wear ratios in the low PV values are smaller than higher PV values both in the Cu-based and Fe-based bearings. Also, Varol and Büyükdavraz showed in their paper (Varol and Büyükdavraz, 2002) that wear ratio of the Fe-based porous bearing was higher than Cu-based porous bearing.

The temperature of the test bearing increased with the increase of the load and speed. A higher temperature increases the probability of getting some oil to the surface of the bearing because of the reduction in the capillary forces tending to hold the oil within the bearing, but it also decreases the viscosity of the lubricant and, therefore, increases the probability that the bearing will run under boundary lubrication conditions. Of course, too high a temperature will result in the carbonization of the lubricant (Cusano and Phelan, 1973). Therefore, wear ratios may be increased in the boundary lubrication regime because of the increasing PV values.

As both bearing material and journal material are the same, wear of the surface may increase. The higher wear at the Fe-based bearing may be explained as insufficient lubrication in the surfaces and adhesion wear may occur between the journal and bearing.

4. Conclusions

In the paper, the tribological properties of porous bearings manufactured with P/M under dynamic loading were studied. Cu-based and Fe-based bearings were used in the tests at different PV values.

(1) The results of tests indicated that Cu-based bearings have better friction and wear properties than Fe-based bearings.

(2) The results of tests showed that the change of friction coefficient with temperature have smaller values of Fe-based bearing when compared with Cu-based bearing.

(3) The results of tests showed that minimum the coefficient of friction for this two group test bearings and test conditions was found the most suitable condition at 40℃ temperature of the bearing.

(4) The results of tests indicated that friction coefficients were reduced with increasing of the loads under constant speed for two test bearing materials. However, friction coefficients were almost same values by the increasing of the speeds under constant load for two test bearing materials.

(4) The antiwear properties of Cu-based porous bearing are more effective in the boundary lubrication regime than Fe-based porous bearing.

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