Effect of micro and macro pits of journal surface on radial pressure distribution of journal bearing

Cem Sinanoğlu*, Fehmi Nair & Ercan Gökşen
Tribology Research Laboratory, Mechanical Engineering Department, Engineering Faculty, Erciyes University, 38039 Kayseri, Turkey

Received 25 January 2007; accepted 24 April 2008

The aim of this study is to find the optimum journal surface micro and macro pits (pores) to improve the load carrying capacity of journal bearing. The effects of various surface porosities on pressure distribution in radial bearings have been investigated. The journals used for experiment in this paper are separated in two groups. Various surface porosities have been achieved by subjecting the surfaces of the two aluminium journals (Al-6063) in the first group (CHE-I, CHE-II) to chemical reaction (etching process). The journals in the second group (COM-I, COM-II, COM-III) have been made of metal matrix composite (MMC) with aluminium based reinforced with SiC ceramic particles, using stir casting technique. For comparison of pressure distribution with that of non-porous journal, a smooth-surfaced journal (NPJ) is used. It has been found that an optimum size and distributive range of micro and macro-pits exists, where the load carrying capacity can be increased at approximately twice over that of an un-textured surface.

Keywords: Surface texture, journal bearing, micro and macro pits

The friction and wear behaviour of boundary lubricated sliding surfaces is influenced by the surface texture. By introducing controlled depressions and undulations in an otherwise flat surface, the tribological properties can be improved. Lubricant can then be supplied even inside the contact by the small reservoirs, resulting in a reduced friction and a prolonged lifetime of the tribological contact.1

Well-defined surface textures were produced by lithography and anisotropic etching of silicon wafers. The wafers were subsequently PVD coated with thin wear resistant TiN or DLC coatings, retaining the substrate texture. The size and shape of the depressions were varied and evaluated in reciprocating sliding under dry and boundary-lubricated conditions.2

Etsion2 investigated the performance of mechanical components by laser surface texturing (LST). Mechanical seals, piston rings and thrust bearings were used in this investigation. It was detected that LST effects an enormous improving of tribological performance. The investigation with seals showed substantial friction reduction and up to three-fold increase in seal life.

Halperin et al.3 performed an experiment on laser-textured seal rings made of steel with micro-pits (pores) of various depths to demonstrate the potential of surface texturing technology. It was found that half spherical pits with optimized depth can maximize the film stiffness and the PV factor at seizure inception over un-textured rings by at least 150% in oil lubrication. Results from analytical solutions indicate that the effect of the pit depth over diameter ratio is more significant than that of pit area ratio (area density).

Textures for non-metal seals have also been tested. Tejima4 formed micro-pits on the surface of SiC by micro blasting, and carried out the friction experiments between carbon and SiC surfaces under water lubrication. It was confirmed that the minimum friction coefficient could be reduced by pits with an optimized diameter.

The aim of this study is to find the optimum surface texture to improve the load carrying capacity of SiC bearings working in water. Micro-pits, evenly distributed in a square array, were selected as the texture pattern, and formed on one of the contact surfaces by reactive ion etching. Experiments, which simulate the working condition of thrust bearings, were carried out to evaluate the effects of micro-pits on the critical load of the transition of the lubrication mode from hydrodynamic to mix. It was found that an optimum geometric and distributive

*For correspondence (E-mail: csinan@erciyes.edu.tr)
range of micro-pits exists, where the load carrying capacity can be increased at least twice over that of an un-textured surface.

An investigation for analysing the load carrying capacity of journal bearing in a variety conditions using a proposed neural network has been done. The neural network structure is very suitable for this kind of system. The network is capable to predict the pressures of the experimental system. The network has parallel structure and fast learning capacity. It can be outlined from the results for both approaches; neural network could be modeled journal bearing systems in real time applications.

In this study, the effects of journal surface porosity on the pressure distribution and, consequently, on the load carrying capacity of journal bearing was experimentally investigated. In order to determine the effects of surface porosity on the pressure distribution, the experiments were conducted at 18°C for aluminium journal systems with different surface porosities.

**Experimental**

**Experimental set-up**

The radial bearing described in Fig. 1 consists of a clear perspex journal bearing mounted freely on an aluminium journal shaft (A). The large diameter journal shaft is directly fixed onto a motor shaft. The speed of the motor shaft (B) is accurately controlled by the standard equipment control unit. With this system a speed range of 500-3000 rev/min can be obtained.

The journal bearing (C) has twelve equispaced pressure tapings around its circumference and four additional pressure tapings along its width. The latter four tapings positioned on the topside of the bearing are sealed by the flexible rubber diaphragm (D) and the other by the clear perspex disc and sealing ring (E). A cursor (F) fixed to the journal bearing at its rear end moves against a single engraved line on a fixed frame (G). When the bearing is in its normal position, the cursor and frame mark are in line. Weights of 100 g each (H) are added to the two-rod (J) during the test to maintain the bearing in its normal position when taking pressure readings. The position of the weights is freely adjustable along the rods.

Oil film pressures are monitored in 16 manometer tubes. Clear flexible plastic tubes connect the manometer tubes to the brass pressure tapping ferrules around the bearing, and thus permit the bearing to turn freely. The upper ends of the manometer tubes are connected to a common manifold and any overflow is returned to a reservoir. This oil reservoir is adjustable in height and is connected to the bearing by a flexible plastic tube and shut-off tap. Oil from this supply reservoir enters the bearing at both ends at its lowest point and outside the actual bearing area. Journal bearing parameters are given in Table 1.

The pressure, which is constant due to on axial direction, indicated by 1, 2,...5 tubes are placed along the bearing axis. The pressure values were measured from 3, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15 and 16th tubes. The tube number also indicates the angles, i.e., 3-0°, 6-30°, 7-60°, 8-90°, 9-120°, 10-150°, 11-180°, 12-210°, 13-240°, 14270°, 15-300° and 16-330°.

<table>
<thead>
<tr>
<th>Table 1—Journal bearing parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Journal bearing parameters</td>
</tr>
<tr>
<td>Diameter of the journal</td>
</tr>
<tr>
<td>Diameter of the bearing</td>
</tr>
<tr>
<td>Effective bearing width</td>
</tr>
<tr>
<td>Overall bearing width</td>
</tr>
<tr>
<td>Dry weight of bearing width attachment</td>
</tr>
<tr>
<td>Weight of each movable load</td>
</tr>
<tr>
<td>Volume of oil carried in bearing</td>
</tr>
<tr>
<td>Dynamic viscosity (Mobil 0W-40)</td>
</tr>
</tbody>
</table>
Experimental procedure

For the experiments, in order to measure the pressure distribution, 12 manometer-tubes were placed around the circumference with 30° (θ) between the tubes and 4 more tubes were located along the bearing. Different rotational speeds (1000, 2000 and 3000 rpm) were employed. Mobil 0W-40 synthetic oil was used as lubricant. In order to investigate the effects of various surface porosities on pressure distribution in radial bearings, the journals used experiments separated in two groups.

In the first group, a total of two aluminium journals have been tested. Various surface porosities were achieved by subjecting the surfaces of the two journals (CHE-I, CHE-II) to chemical reaction. The specimens of journals to be used in the experiments have been manufactured from Al-6063 (AlMgSi0.5). The surfaces of CHE-I and CHE-II were stored in a solution of 20% caustic soda (NaOH) at 70°C for 30 and 45 min respectively to produce surface porosity. It has been observed that micro pores emerged as a result of following etching process

\[
2 \text{Al} + 2 \text{NaOH} \rightarrow 2 \text{NaAlO}_2 + 3 \text{H}_2 \text{ (Gas)} \quad \ldots \ldots (1)
\]

In the second group, the experiments were conducted at 18°C for metal matrix composite journals with different surface porosities. A total of three composite journals (COM-I, COM-II and COM-III) have been tested. The experiment specimens (COM-I, COM-II, COM-III) used in this study have been made of metal matrix composite (MMC) with aluminium based reinforced with SiC ceramic particles, using stir casting technique.

In this method, the matrix material with Al-6063 alloy is melted in a ceramic crucible and then SiC particles with 10% volume ratio are added to this solution in inert gas atmosphere. The average sizes of SiC particles are 511 µm, 167 µm and 63 µm for COM-I, COM-II and COM-III, respectively. The homogeneous particle distribution has been provided by continuously mixing with a mechanical mixer. This mixture is hardened casting in a metal mould. The metal mould is prepared in a shape similar to that of the specimens to avoid manufacturing with lathe. The specimens with 6063 Al/SiCp MMC are stored in a solution of 20% caustic soda (NaOH) at 70°C for 20 min to produce surface porosity. It has been observed that micro and macro pores emerged as a result of this etching process. While micro pores directly occurred as a result of the chemical interaction between caustic soda and aluminium, macro pores are the voids that have occurred as a result of etching the SiC particles. The sizes and ratios of these macro pores can be changed depending on SiC particle sizes and ratios. In addition, reinforcement with SiC ceramic particles not only provides the formation of pores but also protects the surface against crushing and wearing, especially during dry friction, thanks to the SiC particles left on the surface. Moreover, for comparison the pressure distribution with that of non-porous journal, a smooth-surfaced journal (NPJ) was used. The journal and nature of pit and size of particles are given in Table 2.

In the experimental study, motor rotation direction was selected to be in the clockwise and then, the motor was switched on and the speed was gradually increased to 1500 rpm. After that the speed was reduced from 1500 to 1000 rpm and the bearing was allowed to settle down in ten minutes. The required loads were added on to the shaft at the bottom, and then a angular displacement was formed in the bearing. When the manometer levels were settled down, the pressure reading on 16 manometers were taken.

Initially, oil tank was fixed 735 mm levels (oil supply head \(P_s = 735 \text{mm} \)). The positive pressure difference values (\(\Delta P = P - P_s\)) correspond to local bearing load capacity.

Results and Discussion

The bearing which has a weight of 650 g was run loadings of 100 and 200 g each on the front and back loading rods, respectively (Fig. 1). The surface textures of the CHE-I, CHE-II whose surfaces have been made porous, are given in Fig. 2 (a) and (b), respectively.

Figure 3 (a) (Case 1) and (b) (Case 2) show the pressure distributions in journals CHE-I and CHE-II, at various angular positions and velocities when they are loaded (100-100 g), respectively.

In case 1, the maximum positive pressure difference occurred at 30° angular position at the velocity of 1000 and 2000 rpm. These values were 4760 and 3760 Pa at 1000 and 2000 rpm, respectively. The maximum pressure difference also occurred at 0° angular position for 3000 rpm. Its value was 3200 Pa.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>NPS</td>
<td>no pit</td>
<td>COM-I</td>
<td>511 µm</td>
</tr>
<tr>
<td>CHE-I</td>
<td>micro pit</td>
<td>COM-II</td>
<td>167 µm</td>
</tr>
<tr>
<td>CHE-II</td>
<td>macro pit</td>
<td>COM-III</td>
<td>63 µm</td>
</tr>
</tbody>
</table>
Fig. 2a
Fig. 2—(a) Surface texture perpendicular and parallel to rotational direction (CHE-I) and details of some area and (b) surface texture perpendicular and parallel to rotational direction (CHE-II) and details of some area.
Loss of pressure differences at $30^\circ$ angular position, $\Delta P$, with increasing rpm to 3000 from 1000 was 48.36%.

When CHE-II was run under the same experimental conditions, the maximum positive pressure difference measured to be 4760 Pa at 1000 rpm at $30^\circ$ (Case 2). Oil film with positive pressure occurred between $0^\circ$ ~ $90^\circ$ and $240^\circ$ ~ $330^\circ$ for all velocities. The calculated loss of pressure was 35.29%.

As it is seen from the results, the average maximum positive pressure difference at $30^\circ$ angular position were 3480 and 3866 Pa for CHE-I and CHE-II, respectively. Since the static pressure is 5880 Pa, the maximum positive pressure developing in the oil film when CHE-I and CHE-II are run at 1000, 2000 and 3000 rpm are 9360 and 9746 Pa, respectively. CHE-I bears the total bearing load of 200 g with an oil pressure of a lower pressure level. Therefore, CHE-I puts up a more favourable performance than CHE-II. Thus, greater loads can be borne at greater pressures when CHE-I is used.

Figure 4 (a) (Case 3) and (b) (Case 4) give the pressure distribution on the oil film according to experimental approach at 1000, 2000 and 3000 rpm when CHE-I and CHE-II are loaded with 200 g each were applied to the bearing, respectively.

The maximum positive difference of pressure occurs at $30^\circ$ angular position for 1000 and 2000 rpm in case 3. These pressure difference values are 5480 and 4480 Pa. Additionally, the maximum positive difference of pressure at 3000 rpm shifted to the angular position of $0^\circ$ for this journal (3480 Pa). Loss of pressure difference with increasing rpm to 3000 from 1000 was 37.95% at $30^\circ$. The pressure values are greater than those of 100-100 g loading.

In case 4, the maximum positive pressure differences were also settled down at $30^\circ$ angular position at 1000, 2000 and 3000 rpm. In this journal, maximum differences of pressure were measured as 4720, 3680, and 2680 Pa for 1000, 2000 and 3000 rpm, respectively. Loss of pressure difference at $30^\circ$, $\Delta P$, with increasing rpm to 3000 from 1000 was 43.22%.

The average maximum positive pressure difference at $30^\circ$ angular position were 4453 and 3693 Pa for CHE-I and CHE-II, respectively. The CHE-II bears the total bearing load of 400 g with an oil pressure of a lower pressure level. Thus, greater loads can be borne at greater pressures when CHE-II is used. In addition, the capacity of load-carriage of CHE-II in these working conditions is greater than that of the other journal.

Fig. 3—(a) The pressure distributions in journal CHE-I, at various angular positions and velocities with 100 g-100 g load and (b) the pressure distributions in journal CHE-II, at various angular positions and velocities with 100 -100 g load

Fig. 4—(a) The pressure distributions in journal CHE-I, at various angular positions and velocities with 200-200 g load and (b) the pressure distributions in journal CHE-II, at various angular positions and velocities with 200 g-200 g load
As it is seen from the results, CHE-I displays a favourable performance in 100-100 g loading conditions. But, CHE-II displays more favourable performance than that of CHE-I in 200-200 g loading conditions as far as load-carrying capacity is concerned.

The surface textures of the COM-I, COM-II, COM-III, which have been made of metal matrix composite and whose surfaces have been made porous, are given in Fig. 5 (a), (b) and (c), respectively.

Figure 6 (a) (Case 5), (b) (Case 6) and (c) (Case 7) give the pressure distribution on the oil film at 1000, 2000 and 3000 rpm when COM-I, COM-II, and COM-III are loaded with 100 g each were applied to the bearing, respectively.

In case 5, the region with positive pressure difference has been achieved approximately at 1000 rpm at $0^\circ \sim 60^\circ$, at 2000 rpm at $0^\circ \sim 90^\circ$, and at 3000 rpm at $0^\circ \sim 120^\circ$. Moreover, for all velocities from $210^\circ$ through $360^\circ$, pressure values are positive. The maximum pressure difference was fixed at $30^\circ$ angular position for 1000 rpm. Its value was fixed at 4680 Pa. Maximum positive difference of pressure occurs at the angular position of $30^\circ$, at 2000 rpm and it is 3320 Pa. The positive pressure difference with a value of 2920 Pa at the angular position $\theta = 30^\circ$ and at 3000 rpm. When rpm rises to 3000 from 1000, the loss in the pressure difference is 39.31%.

In case 6, the maximum positive pressure difference with a value of 4400 Pa at the angular position $\theta = 30^\circ$ and at 1000 rpm. Loss of pressure
difference at 30°, $\Delta P$, with increasing rpm is 1800 Pa. A loss of 40.90% in pressure and hence in the load carrying capacity of the bearing was detected.

The maximum positive differences of pressure that occur at the same angular position (30°) at 1000 and 2000 rpm are 4720 and 3040 Pa (Case 7). Additionally, the maximum difference of pressure at 3000 rpm shifted to the angular position of 0° (2720 Pa). Loss of pressure difference at 30° was 45.76%.

Figure 7 (a) (Case 8), (b) (Case 9) and (c) (Case 10) give the pressure distribution on the oil film at 1000, 2000 and 3000 rpm when COM-I, COM-II, and COM-III are loaded with 200 g each were applied to the bearing, respectively.

In case 8, the maximum pressure differences were fixed at 30° angular position at 1000, 2000 and 3000 rpm. These values were 5560, 4440 and 3400 Pa for 1000, 2000 and 3000 rpm, respectively. When rpm rises to 3000 from 1000, the loss of pressure difference was calculated to be 38.84%.

In case 9, the maximum pressure differences were also settled down at 30° angular position at 1000, 2000 and 3000 rpm. In all three journals, maximum differences of pressure were measured as 6080, 3920, and 3160 Pa for COM-I, COM-II and COM-III,
respectively. Loss of pressure difference, ΔP, with increasing rpm to 3000 from 1000 was 48.02%.

In case 10, the maximum pressure difference occurred at 30° angular position at the velocity of 1000 and 2000 rpm. These values were 6080 and 4160 Pa at 1000 and 2000 rpm, respectively. The maximum pressure difference also occurred at 0° angular position for 3000 rpm. Its value was 3200 Pa. Loss of pressure differences, ΔP, with increasing rpm to 3000 from 1000 were 10.11% and 48.02% at 0° and 30°, respectively.

As can be seen from the results, COM-II bears the total bearing load of 400 g with an oil pressure of a lower pressure level. Therefore, greater loads can be borne at greater pressures when COM-II is used. In addition, the capacity of load-carrying of COM-II in these working conditions is greater than those of the other journals.

Moreover, COM-II also carries the total bearing load of 200 g with an oil pressure of a lower pressure level. In this loading condition, the average pressure difference on oil film are 3613, 3346 and 3440 Pa for COM-I, COM-II and COM-III, respectively. The load-carrying capacity of COM-II is approximately 7% and 3% greater than COM-I and COM-III, respectively. As a result, COM-II displays a favourable performance as far as load-carrying capacity is considered.

All the numerical values presented so far were related with the testing the effects of journal surface porosity on the pressure distribution on the oil film. In addition, the performance of a non-porous journal (NPJ) has been tested in the study. The surface texture of this journal is given in Fig. 8.

In the first group journals, CHE-I and CHE-II have been displayed a favourable performance running of 100-100 g and 200-200 g, respectively. In the second group journals, COM-II has been showed a favourable performance in these two loading

Fig. 7—(a) The pressure distributions in journal COM-I, at various angular positions and velocities with 200-200 g load, (b) The pressure distributions in journal COM-II, at various angular positions and velocities with 200-200 g load and (c) the pressure distributions in journal COM-III, at various angular positions and velocities with 200-200 g load

Fig. 8—The surface texture of a non-porous journal (NPJ)
conditions. Figure 9 (Case 11) shows the pressure distributions in journals CHE-I, COM-II and NPJ, at various angular positions at 1000 rpm with 100-100 g load.

COM-II carries the total bearing load of 200 g with an oil pressure of a lower pressure level (3040 Pa). In addition, CHE-II (4720 Pa) bears the total bearing load of 400 g with a lower pressure level than COM-II (6080 Pa) and NPJ (6920 Pa) at 1000 rpm. The orders of the load carrying capacity are CHE-I, COM-II, NPJ and CHE-II, COM-II, NPJ in the total bearing load of 200 g and 400 g, respectively.

Figure 10 (a) (Case 12) and (b) (Case 13) show the pressure distributions in the second group journals CHE-II, CHE-I and NPJ, at various angular positions at 3000 rpm when they are loaded with 100-100 g and 200-200 g. Especially in case 13 it is to see that the pits improve the load carrying capacity enormous. The maximum pressures is at 0° angular position and the pressure difference between NPJ and CHE-II is 1040 Pa, that means with CHE-II it is to get 28% more load carrying capacity. A same effect is to see in the work of Etsion\textsuperscript{2} he did get a reduction of from 65% to 27% in friction torque by micro-dimples. The reduction of torque or pressure means the reduction of the friction and increase of the life of the components.

Figure 11 (a) (Case 14) and (b) (Case 15) show the pressure distributions in the second group journals COM-I, COM-II and COM-III, at various angular positions at 3000 rpm with 100-100 g load and (b) the pressure distributions in journals COM-I, COM-II and COM-III, at various angular positions at 3000 rpm with 200-200 g load.

CHE-II, CHE-I and NPJ, at various angular positions at 3000 rpm when they are loaded with 100-100 g and 200-200 g. Especially in case 13 it is to see that the pits improve the load carrying capacity enormous. The maximum pressures is at 0° angular position and the pressure difference between NPJ and CHE-II is 1040 Pa, that means with CHE-II it is to get 28% more load carrying capacity. A same effect is to see in the work of Etsion\textsuperscript{2} he did get a reduction of from 65% to 27% in friction torque by micro-dimples. The reduction of torque or pressure means the reduction of the friction and increase of the life of the components.

The experiment with SiC thrust bearing\textsuperscript{5} showed similar effects; there was an optimum size of the pit diameter. In experiment specimens with pit diameter of 0, 50, 150, 250, 350, 500 and 650 µm were used. The specimens with pit diameter of 250 and 350 µm gave better results in the same conditions.
Conclusions
The following conclusions have been drawn from this study:
(i). A certain degree of porosity of the journal-surface enhances the load carrying capacity of the bearing.
(ii). The sizes and ratios of micro and macro pores in journals with Al-6063 subjected to etching process can be changed depending on etching time.
(iii). The sizes and ratios of micro and macro pores in the MMCs journals can be changed depending on SiC particle sizes and ratios.
(iv). The capability of load carrying depends on the number of revolutions in the journals.

References