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RESEARCH

Experimental Investigations on Temperature and Frictional Behaviors of Textured Journal Bearings

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A B S T R A C T

Experimental work provides a greater insight into the performances of journal bearings under actual operating conditions. Moreover, experimental findings also provide a platform for validation of numerical model based results and helps to point out suggestions needed to pursue further research. This article has been written in the direction of providing few factual experimental observations using the fabricated bush bearings having conventional, full textured (0^0 to 360^0) and half textured (0^0 to 180^0 and 180^0 to 360^0) bores. The textured bores involving cylindrical dimples were fabricated using electric discharge machining (EDM). Experiments were conducted at two loads (load per unit projected area = 0.3 and 1.5 MPa) and two journal speeds (2 and 5 m/s) employing SAE-30 lubricating oil. The experimental results revealed that the bearing with textures lying in the converging zone (0^0 to 180^0) yielded maximum reduction in coefficient of friction as compared to conventional (smooth) and other textured cases considered herein. Temperature rise with this textured bearing (0^0 to 180^0) also found less among all the textured bore cases.

1. INTRODUCTION

Due to the growing demand of energy efficient, compact, and hardworking machinery in industries, great efforts are being made by the researchers across the globe to develop compact and energy efficient fluid film bearings for machines to sustain light to heavy loads at medium to high speeds with improved stability. Hydrodynamic journal bearings are widely used in turbines, compressors and pumps due to their excellent dynamic behavior. Since recent past, number of design modifications have been proposed by the researchers in order to improve the performance of fluid film journal bearings.

Recently surface textures (systematic patterning of micro-dimples and micro-grooves on stationary and moving surfaces) have emerged as a viable technology for manipulating the surface topography of mating solids for achieving the operational benefits of textured contacts. It is worth mentioning here that the textured surfaces are characterized by the size, shape, and pattern

orientation of nano/micro features (dimples and grooves). There are various shapes of micro-geometries such as rectangle, circular, triangle, ellipsoidal etc., which are commonly used in creating the surface textures. The presence of surface textures influences the tribological behaviors of contacts in the following ways:

- Combinations of innumerable tiny flat land and micro dimple/groove at a textured surface act as innumerable micro-hydrodynamic bearings when the surface comes in the contact of a moving counter surface.
- Micro-dimples/grooves retain and supply the lubricant for smearing at the interface if the interface operates under mixed and boundary lubrication regimes.
- Debris and contaminants generated at the interface get trapped in the dimples/groove.

A lot of computational work has been reported by researchers witnessing the positive aspects of surface textures. Many researchers have emphasized on adoption of partial surface textures instead of full textures by optimally utilizing other crucial parameters such as location, size, geometry etc. of the textures [1-10]. Researchers have also carried out experimental studies on the hydrodynamic bearings to understand the bearing performance parameters under the real operating conditions. Lu and Khonsari [11] performed experiments using dimpled journal bearing. The authors used machining and chemical etching techniques to create dimples on the bushes and concluded that proper dimple size, shape and depth are essential to improve the friction performance. Yu et al. [12] investigated the effects of different dimple shapes on the tribological performance of textured parallel surfaces. The authors have reported significant friction reduction with textured surface in comparison to conventional smooth surface. It has been reported that elliptical dimples always yielded positive results under different test conditions as compared to square and circular dimples.

Experimental comparisons of tribological performance behaviors of smooth and laser textured discs having three different dimple depths have been presented by Vilhena et al. [13]. It has been noticed that the tribological behaviors of textured surfaces depend on the depth of micro-dimples. It has also been observed that the beneficial effect of micro dimples become significant with an increase in dimple depth at higher speeds. Galda et al. [14] studied the effect of shape and distribution of oil pockets on the characteristics of Stribeck curve. It has been shown that with proper shape and dimensions as well as having optimized area density of oil pockets, the friction characteristics of the sliding pairs could be improved in comparison to smooth surface cases. Wang et al. [15] investigated experimentally the effect of dimple depth and dimple shape on the drag force. It has been reported based on their study that the dimple shape and distribution had little effect on the drag reduction.

Adatepe et al. [16] have conducted an experimental investigation on frictional behavior of statically loaded micro grooved journal bearing. The authors concluded that the Stribeck curves obtained experimentally for the circumferential and transversal micro-grooved journal bearings are similar to that of the plain journal bearing. The highest value of coefficient of friction has been recorded with the transverse micro-grooved journal bearing followed by the circumferential micro-grooved and plain journal bearings. The coefficient of the friction with micro-grooved journal bearing decreased with the increase in the bearing load. The maximum frictional torque was observed with the transverse micro-grooved journal bearing followed by the circumferential micro-grooved and the plain journal bearings.

Marian et al. [17,18] investigated the effects of partial surface textures under circumferential and radial direction of motions and concluded that the textured bearing yielded lower friction coefficient in comparison to plain bearings. Meng et al. [19] explored the influence of rectangular dimple with flat bottom on the friction between the parallel surfaces. Decrease in the friction force has been reported due to the dimple effect. At the low values of load and speed, insignificant effect of the dimples on the friction force is found. The authors have reported that dimples on parallel surfaces can decrease the coefficient of friction for the cases having smaller ratio of film thickness to roughness. Scaraggi et al. [20] measured the coefficient of friction at the lubricated textured surfaces having two patterns of textures. One is made of a square lattice of micro holes and the second comprises with series of microgrooves. The regular array of micro holes allowed friction reduction for the entire range of lubrication regimes. However, on the

contrary, the parallel microgrooves lead to increase in friction in comparison to the plain surface.

Kumada et al. [21] found that increased oil flow with micro grooved surfaces reduced the temperature rise in bearing. Presence of micro-grooves even prevented the seizure chances at the interface under the starved lubricating conditions. However, very high value of coefficient of friction has been recorded in the presence of cross lubricating channels on mating surfaces [22]. Shen and Khonsari [23,24] reported that the value of cavitation pressure plays very important role on the bearing performance. The authors have demonstrated that low cavitation pressure is very much beneficial. Zhang and Meng [25] also reported advantage of low value cavitation pressure over the bearing performance. Wang et al. [26] experimentally investigated the lubricant extraction from a micro-pocket. The authors concluded that micro-pockets serve as oil reservoir to provide lubricant under boundary and starved lubrication regimes. Using pin-on-disc configuration, Segu et al. [27] have experimentally investigated the effect of multi scale textures on lubrication regimes. It was found that the dimples with an area density of about 12% was more effective in friction reduction. The beneficial effects of multi scale dimples becomes significant with increase in dimple depth and sliding speed.

Nakano et al. [28] investigated the effect of grooves and dimples at the lubricated interface using a pin-on-block tribometer. The authors observed that the surfaces with groove patterns yielded high friction coefficient. However, it has also been noticed that the hydrodynamic pressure in lubricating film enhances in presence of dimples. The tribological behavior of Babbitt alloy having surface textures has been investigated by Zhang et al. [29] under the mixed and starved lubricated conditions. Lower and stable friction coefficient have been recorded by the authors. The experiments have been carried out to investigate tribological performance of triangular textures with water as lubricant [30]. The authors have concluded that the textures improve the load carrying capacity. Suh et al. [31] have studied the effect of micro-cross hatching patterns on lubricated sliding friction using pin on disc configuration. The results showed that friction can be controlled by fabricating the micro grooved cross hatchings on the contacting surfaces. It has been reported that geometrical parameters of textures influence the friction significantly. Ramesh et al. [32] investigated the friction characteristic of microtextured surfaces under mixed and hydrodynamic lubricating conditions. During hydrodynamic lubricated condition, the textured surfaces exhibited about 60-80% reductions in friction force as compared with plain surfaces. Lu et al. [33] have also investigated the friction performance of textured journal bearing using phyllotactic pattern on mating surface. It has been reported that the friction performance of journal bearings evidently improved in presence of texture.

It has been observed from above review that very few experimental studies on textured journal bearing performances have been reported particularly considering the presence of segmental textures at bore surface. Therefore, the aim of present study is to experimentally investigate the friction and temperature rise performances of textured journal bearings for providing a platform for researchers to further pursue their research work in this area by utilizing these observations.

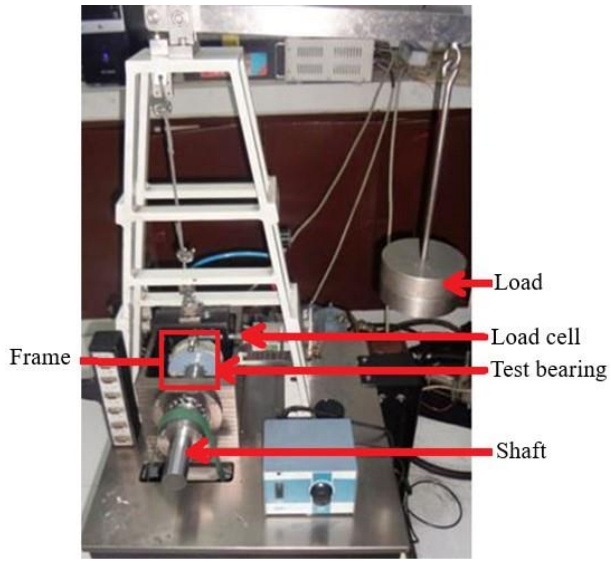
2. MANUFACTURING DETAILS AND EXPERIMENTAL PROCEDURE

2.1 Journal bearing test set-up

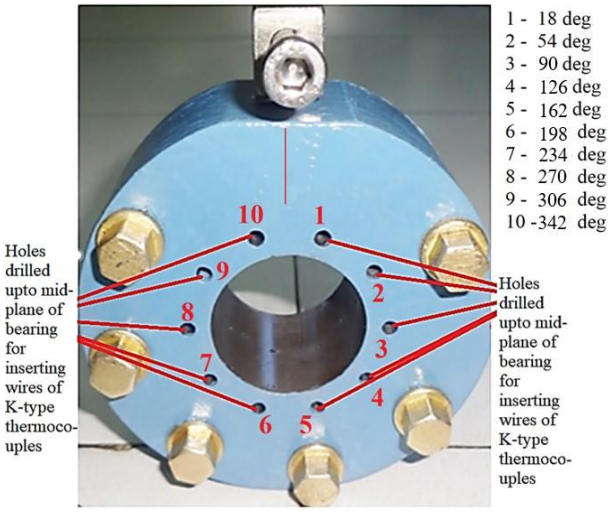
Figures 1(a) and 1(b) present the photographic views of test rig and a test bearing, respectively, employed in the present investigation. The main components of the test rig are: AC induction motor, driven pulley (1:3), bearing, shaft, load cell sensor (for measuring frictional torque), and lubricant reservoir. The specifications of the test rig are also provided in Table 1. The shaft is mounted horizontally on two self-aligned ball bearings (single piece mounted at each end). AC motor was used to rotate the shaft via belt drive system. The test bearings freely slid axially over the journal.

Table 1. Specifications of test rig.

Parameter	Value
Shaft diameter (mm)	39.98
Length/diameter ratio	1
Maximum load (kg)	300
Shaft speed (rpm)	400- 2500
Resolution of load cell (N)	0.1



(a)



(b)

Fig. 1. (a) Photographic view of experimental setup, (b) View of a test bearing with temperature measuring locations.

The vertical load is applied to the test bearing from bottom side via a metallic frame pulling it up through a wire rope connected to one end of lever while dead weights were applied at the other end of the lever. The height of frame was kept considerably high to diminish the load due to the inclination of loading lever. The frictional moment due to additional force was accounted.

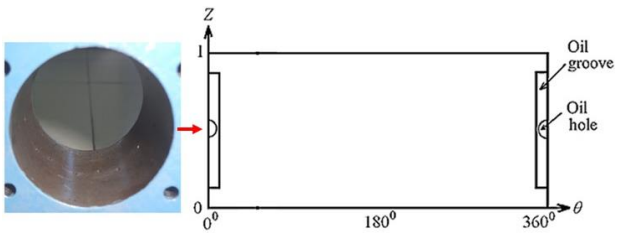
2.2 Manufacturing of bores and incorporation of surface textures

In the present study, three Gray cast iron (Hardness=20-22 HRC) bearings were got fabricated through a vendor. The details of these bearings are as follows: Bearing no. 1: Conventional bore journal bearing; Bearing no. 2: Full Textured (0° to 360°) bore bearing; Bearing no. 3: Textured (0° to 180° and 180°-360°) bore journal bearing (This bearing was used to perform the experiments by positioning in such a way that texture get placed in the zone of 180°-360° i.e. rotating the face).

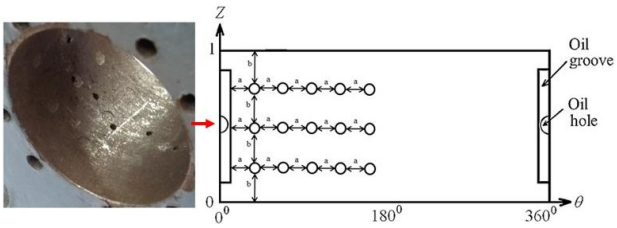
Table 2. Bore details of Gray cast iron journal bearings.

Bearing number	Surface roughness R_a (µm)	Actual bore diameter (mm)	Cylindricity (Tolerance zone, mm)
1 Conventional bore	2.1241	40.092	0.0214
2 Full textured bore	2.1456	40.076	0.0215
3 Textured bore I & II	2.1347	40.098	0.0211

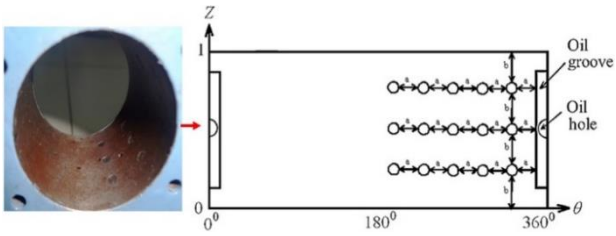
The bearings were examined for surface roughness (R_a) and cylindricity using stylus profilometer and coordinate measuring machine (CMM), respectively. Roughness and cylindricity data have been provided in Table 2. Shaft is made of carbon steel (C - 45) having hard chrome plating (Hardness = 55 - 60 HRC). The surface roughness and cylindricity of shaft are Ra= 0.4 - 0.6 µm and 0.008 mm, respectively. The measured shaft/journal diameter is 39.98 mm.



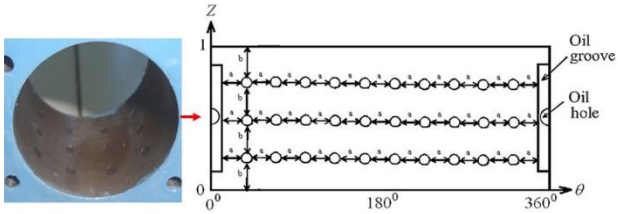
(a) Conventional bore (Left hand side-photographic image and Right hand side- schematic illustration of unwrapped view).



(b) Texture-I (Left hand side-photographic image and Right hand side- schematic illustration of unwrapped view).



(c) Texture-II (Left hand side-photographic image and Right hand side- schematic illustration of unwrapped view).



(d) Full Textured bore (Left hand side-photographic image and Right hand side- schematic illustration of unwrapped view).

Fig. 2. (a) - (d): Photographic images of textured and plain (conventional) bores and their schematic unwrapped views.

The textures on the inner surface of bearings were created using an Electrical Discharge Machining (EDM). Figures 2(b) to 2(d) show the textures having cylindrical dimples of average diameter 3.2 mm and average depth 70 μm. The schematic views of unwrapped bores of all four test bearings have been given in Figs. 2 (a) to 2(d). The texture zones have been clearly described for each bearing. The detailed dimensions of textures and distance between two textures have been depicted by an enlarged view of photograph of a test bearing shown in Fig. 3.

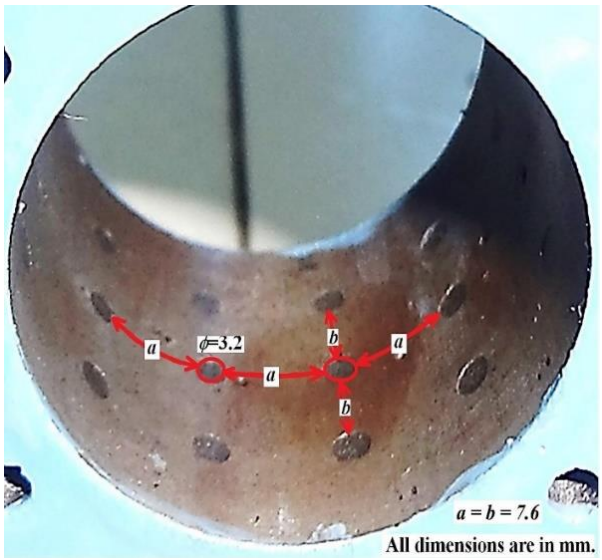


Fig. 3. Enlarged photographic view of the inside surface of a textured test bearing's bore.

2.3 Lubricating oil

SAE-30 lubricating oil (without any additives) was used to lubricate the test bearings. The measured viscosity and density of oil are: $\eta_{@40^0C}$= 99.40 cSt, $\eta_{@100^0C}$= 12 cSt and $\rho_{@25^0C}$ =860 kg/m^3. A small sample of the lubricating oil is drawn into a calibrated capillary tube in a constant temperature bath. Once the sample has come to desired temperatures (40°C and 100^{0}C), it is allowed to flow down the tube for predetermined distance. Thus, the viscosity is computed as the product of flow time and tube calibration factor. However, the density of lubricating oil is measured using a standard weighing procedure with density bottle.

2.4 Experimental procedure

The experimental procedure mainly involves the following steps:

1. Three litres of clean lubricant (SAE-30) is poured in oil tank for supply of lubricant to the test bearing. The feeding pressure of lubricant inside the bearing about 2 times of the atmospheric pressure (0.2 MPa). Temperature of lubricant in oil tank is recorded. The feeding temperature of lubricant is maintained constant equal to 40°C for all the tests by using heat exchanger.

2. The load and speed is selected through the software. The speed of the journal is increased slowly. The desired load is also applied at the loading lever in a stepwise manner. After 40 minutes of running operation (for achieving steady state condition), readings of friction torque and temperature were recorded. Frictional force measuring concept has been shown in Figs. 4(a) and 4(b). The accuracy in friction force measurement is about ± 1.5%. The friction force sensor is pre-loaded. Each pre-loaded reading is deducted from the corresponding final load reading. Pre –load was different (with ± 5% variation) for each case.

3. Each set of experiment was carried out three times for checking the repeatability of the results. The largest difference observed is less than 1 ^{o}C for maximum temperature that are about 1% of the maximum value.

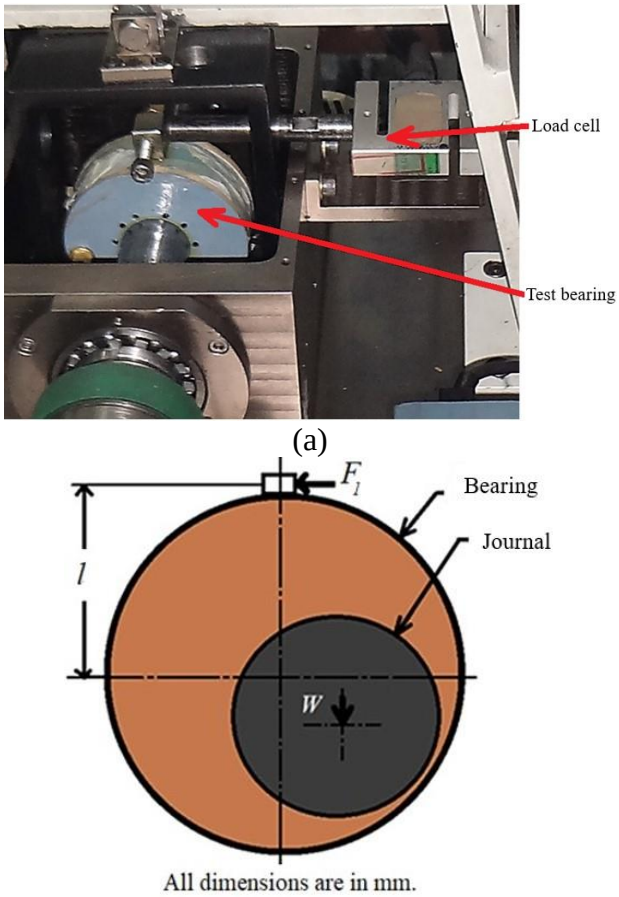


Fig. 4. (a): Position of load cell in the test setup and (b) schematic representation of measurement of friction torque.

2.5 Calculation of temperature

K-type thermocouples have been used to measure the temperature of bearing with accuracy of ±1^{0}C. The locations of temperature measurement are at the mid-plane of bearing close to bore. Moreover, the additional information related to locations can also be seen in Fig. 1(b). Total ten holes close to bore of bearing have been numbered where temperatures were measured. Angle made by each hole at centre of bearing w.r.t. vertical line, moving in clockwise direction have also been indicated along with numbers in Fig. 1(b) for adding more clarity. The distance between the temperature pick-up and bore surface is equal to 1 ± 0.05 mm.

2.6 Calculation of friction force

The load cell measures the friction force exerted by the free bearing. The position of the load cell is shown in Fig. 4 (a), whereas Fig. 4 (b) displays the friction torque measurement scheme. The value of torque was displayed on the computer screen after software based calculations. The concept employed in test rig to find the coefficient of friction is as follows:

Coefficient of friction (μ) = F/W

Where,

$T = F_1 \times l$ N-mm
l = Distance between the load cell and bearing centre = 62 mm
$F = T / (D/2)$
D= Bearing bore (mm)

3. RESULTS AND DISCUSSION

The experiments were performed using all four test bearings as per the procedure mentioned in the previous section - 2.4. The temperature rise and friction were recorded to assess the effects of segmental textures present at bore of test bearings. It has been shown in Fig.5 that at constant external applied load of 0.3 MPa (500 N), all test bearings with micro-texture shows higher temperature as compared to the conventional bearing case. However, out of various considered texture locations, bearing with presence of texture in entrance region yielded least value of the maximum temperature rise. Previous published studies [34-36] suggested that the rise in lubricant temperature in hydrodynamic journal bearing is highly dependent on the value of lubricant film thickness. It has been observed that the maximum temperature in bearing is increasing with reduction in the lubricant film thickness and vice versa behaviour is also observed with the increment in the film thickness value [36]. Tala-Ighil et al. [2,3], Brizmer and Kligerman [37], and Fowell et al. [38] in their theoretical investigation found that the presence of micro-texture at the inlet region of sliding contact bearings is causing significant increment in the volumetric inflow rate of the lubricant. They further mention that bearing with texture in the diverging or cavitation zone is not effective in improving the tribological performances and these cause reduction in the lubricant film thickness. This improvement in volumetric inflow rate with partial texture may be responsible for the minimum temperature rise as compared to the full and diverging zone textured bearing. Moreover, the decrease in lubricant film thickness due to presence of textures in full and

diverging zones, is one of the reasons behind the higher temperature rise of the lubricant film [24]. Similar behaviour has been observed in the mid-plane temperature rise variation at the load of 1.5 MPa (2500 N) and speed of 2 m/s as shown in Fig. 6. It may also be noted that the difference of temperature rise curves among the considered textured locations becomes very less as film thickness get reduced due to the application of high loads. After examining the effects of low and high loads on temperature rise at low speed (2 m/s), the speed has also been increased to 5 m/s to see its effect. Figures 7 and 8 present the temperature rise variations along the circumferential direction at low load (0.3 MPa) and high load (1.5 MPa), respectively, at speed of 5 m/s. Similar trends of temperature rise have been found as of at speed of 2 m/s, but magnitude of temperature rise has increases more than 2 times. It is understood that it has happened due to increased viscous heat generation at 5 m/s as compared to case of 2 m/s. It is essential to mention here that relatively low temperature rise in textured bearing (texture lying in entry zone) in hydrodynamic lubrication regime (since present experimental study has been done in hydrodynamic lubrication regime only) has happened not due to storage of lubricant in micro-dimples, rather due to enhanced side leakage. In presence of certain textures (having micro-grooves and dimples) at certain locations in the bearing, the film thickness increases as compared to conventional bearing case at the same operating condition (i.e. at identical load and speed). This increase in film thickness yields more side leakage (which carries heat from bearing in convection mode), hence, the reduction in temperature rise in bearing or effective lubricant cooling inside the bearing.

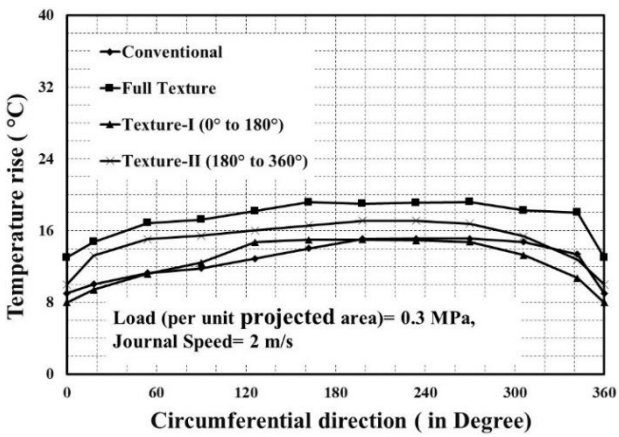


Fig. 5. Mid-plane temperature rise in test bearings vs. circumferential direction (0.3 MPa, 2m/s).

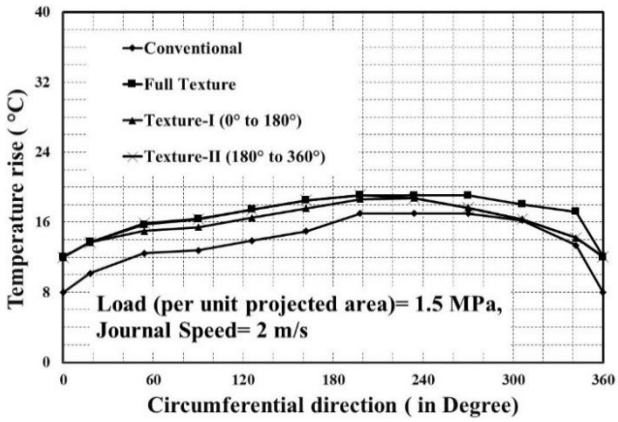


Fig. 6. Mid-plane temperature rise in test bearings vs. circumferential direction (1.5 MPa, 2 m/s).

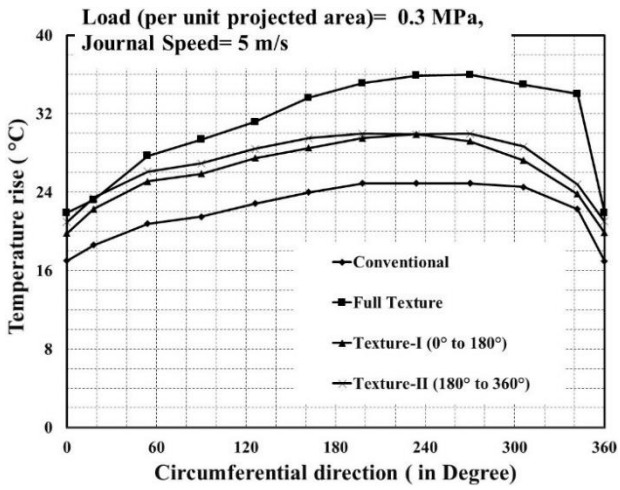


Fig. 7. Mid-plane temperature rise in test bearings vs. circumferential direction (0.3 MPa, 5 m/s).

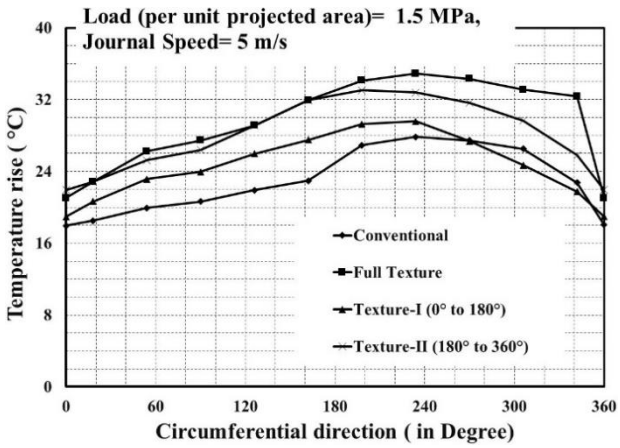


Fig. 8. Mid-plane temperature rise in test bearings w.r.t. circumferential direction. (1.5 MPa, 5 m/s).

Figures 9 and 10 show the comparison of coefficient of friction (COF) values at loads of 500 N (0.3 MPa) and 2500 N (1.5 MPa) at two speeds (2 and 5 m/s). It can be seen in Fig. 9 that at low load (0.3 MPa) and low speed (2 m/s), the COF values are almost same. This has happened due to large

film thickness at lightly loaded condition. The large film thickness diminishes the effect of surface texture, however, at high speed (5 m/s) the value of COF is found minimum for test bearing (with texture I (0^0 to 180^0)) as compared with plain and other two textured cases. It is worth noting here that at high speed (5 m/s) more viscous heat dissipation has taken place as compared to case of 2 m/s, this caused more temperature rise hence reduction in viscosity of oil.

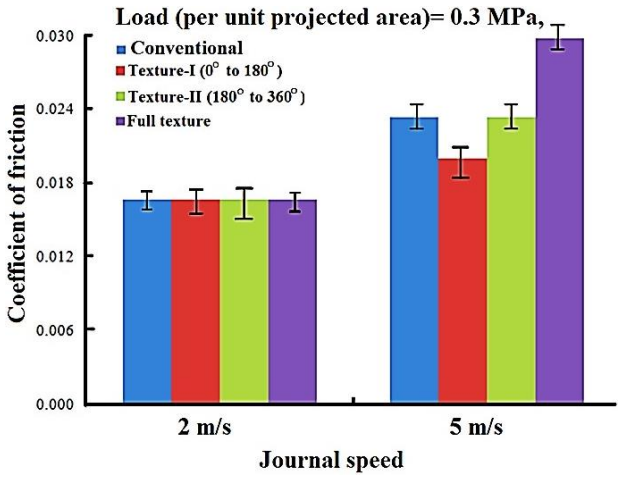


Fig. 9. Comparison of coefficient of friction of different bearings (0.3 MPa).

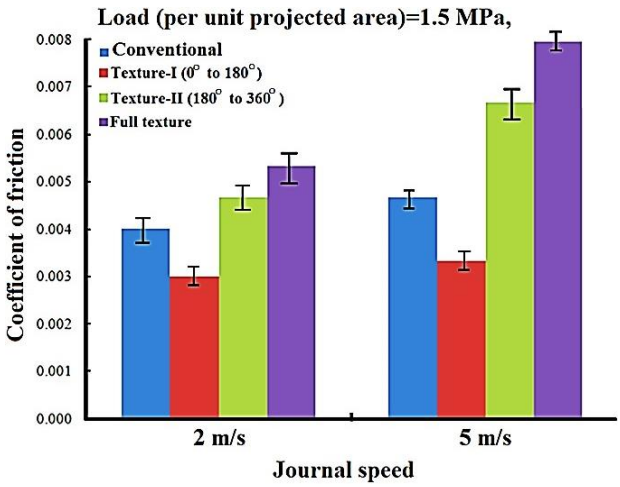


Fig. 10. Comparison of coefficient of friction of different bearings (1.5 MPa).

This reduced viscosity of oil has caused reduction in film thickness as well. Thus, the synergistic effect of low viscosity and textures led to variation in friction coefficient in Fig. 9 for case of 5 m/s. Moreover, Figure 10 shows variation in COF at both the speeds (2 and 5 m/s). It is due to high load (1.5 MPa), that has caused lower film thickness, hence effect of surface texture has taken place.

4. CONCLUSIONS

Based on the experimental investigations presented herein, the following conclusions have been drawn:

- Partial textures provided in entry converging zone (Texture I: 0° to 180°) yields significant reduction in coefficient of friction as compared to other cases adopted.
- Bearing with Texture-I has also reduced temperature as compared with other textured test bearings.

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