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RESEARCH

Evaluation of Oil Film Pressure and Temperature of an Elliptical Journal Bearing - An Experimental Study

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A B S T R A C T

The present study is aimed at experimental evaluation of both oil film pressure and temperature at the central plane of finite elliptical journal bearing configuration. These parameters have been obtained by running the machine at various speeds under different applied loads ranging from 500 N to 2000 N using three different grades of oil (HYDROL 32, 68 and 150). The data has been obtained through a test rig which is capable of measuring both pressure and temperature at the same location on the elliptical bearing profile. An elliptical journal bearing with journal diameter=100 mm, L/D ratio=1.0, Ellipticity Ratio=1.0 and radial clearance=0.1 mm has been designed and tested to access the pressure and temperature rise of the oil film at the central plane of the bearing. Two different lobes of positive pressure have been obtained for elliptical bearing which results in smaller area for cavitation zone and accounts for better thermal stability. Also, with the increase in load both pressure and temperature of an oil film increases for all the three grades of oil. Experimentally, it has been established that the HYDROL 68 is suitable grade of lubricating oil which gives the optimum rise of pressure and temperate under all operating conditions among the lubricating oils under study.

1. INTRODUCTION

The current trend of industries demands for those machinery which carry both high speed and load. Hydro-dynamic journal bearings are extensively used in high speed machines to support the shaft which runs in these machineries. The conventional circular shape of journal bearing results in large heat generation in the lubricant film when operated at high speeds and loads. This affects the performance parameters of the bearing like stability and load capacity of the bearing. Researchers are continuously investigated the various alternative solution to overcome this problem and have suggested many design modification in the shape of the circular bearing for not only to increase the stability but also for less thermal degradation of lubricating oil under high speed and load. One such design modification is to operate the machines with non-circular journal bearing such as elliptical, offset, two lobes, three lobes etc instead of circular journal bearing. These non-circular bearings works with more than one active oil film as compare to circular bearing. The number of active oil film signifies the number of lobes of positive pressure zone around the circumference of the bearing. This features reduces the area of the cavitation zone and hence accounts for better stability and lesser rise in oil film temperature when compared with circular bearings which works on single active oil film. Experiments related to measurement of both pressure and temperature of lubricating oil film, at the same location, inside the bearing gives the real scenario for calculation of performance parameters of the bearing. It has been observed by the authors that very limited experimental work for the evaluation of both pressure and temperature has been reported for non-circular journal bearing. This unavailability of data leads to hindrance for the designers to choose the proper non-circular journal bearing based on the requirement. A few experimental and theoretical works related to thermal behaviour of non-circular journal bearings are reviewed here.

Pinkus and Lynn [1] have firstly investigated the performance parameters of non-circular elliptical bearing theoretically and have found better steady state load capacity and power losses when compared to circular bearing. Tayal et al. [2] and Prabhakaran et al. [3] have investigated the performance characteristics of finite width elliptical journal bearing through finite element method. Singh et al. [4] have studied the effects of bearing linear elastic deformation and lubricant viscosity with pressure and temperature for an elliptical bearing. Hussain et al. [5] have predicted the temperature distribution in different non-circular journal bearings (namely two-lobe, elliptical, and orthogonally displaced) based on a two-dimensional treatment following McCallion's approach. Ma and Taylor [6] have experimentally evaluated the temperature for two-axial groove circular and an elliptical (lemon bore) bearing. Mishra et al. [7-8] have considered the non-circularity in bearing bore to be true elliptical and made a comparison with the circular case to analyze the effect of manufacturing irregularity in the bearing and have observed that with increasing non-circularity the pressure gets reduced and temperature rise is less in case of journal bearing with higher non-circularity value using finite difference method. Ostayen and Beek [9] have also presented a thermo-hydrodynamic model for analyzing a lemon-bore journal bearing through finite element method and the presented model is used to check the design of the lemon bearings in a specific naval application. Chauhan et al. [10-11] and Sehgal et al. [12-14] evaluated thermal performance parameters theoretically and experimentally for offset and elliptical bearing (also named as two lobes) considering three different grades of oils. The authors have evaluated only the temperature of oil film along the circumference of the bearing at various speeds and loads. Huang et al. [15] have presented the thermo-elasto-hydrodynamic performance analysis for two lobe elliptical journal bearing through finite element analysis commercial available software. Mishra [16] has studied the performance characteristics of an elliptical profile journal bearing considering isotropic surface roughness orientation by stochastic function. Bhagat and Roy [17] have done steady state thermo-hydrodynamic analysis for two axial groove and multi lobed hydrodynamic journal bearing and found that oil film temperature is highest for three lobed journal bearing as compared to other bearings under study.

The literature reveals that very little experimental work has been done for evaluation of performance parameters related to non-circular journal bearing. The most of the experimental work is related to the elliptical journal bearing is for evaluation of oil film temperature along the circumference of the bearing. Also, the available work is limited to those elliptical bearing which was formed by displacing the two circular arcs of the bearing towards the center and also named as two lobe bearing. The authors find no paper which evaluates both the pressure and temperature simultaneously at the same location of non-circular journal bearing. In the present work, customized journal bearing test rig has been designed which is capable of measuring the pressure as well as the temperature of the oil film along the central plane of the bearing at different combinations of the journal speed and the load. The test bearing is a non-circular elliptical journal bearing made up of bronze. The non-circularity in the elliptical journal bearing is made by cutting a elliptic slot having ellipse profile defined by major and minor axis inside the cylinder, named as elliptical bearing. The schematic diagram of elliptical bearing has been shown in Fig 1.

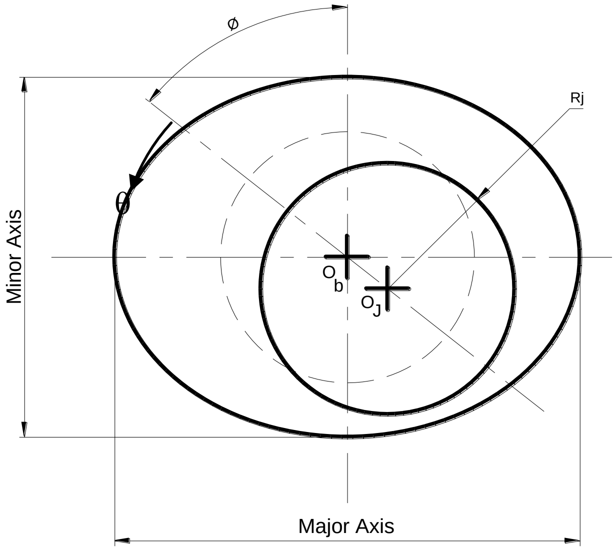

Fig. 1. Schematic diagram of elliptical profile journal bearing.

2. EXPERIMENTAL SET UP

The bearing test rig (Fig. 2) consists of a journal, 100 mm in diameter, mounted horizontally on a housing supported on self-aligned bearings. The shaft/journal is driven by AC motor with timer belt having 1:2 pulley ratios which is capable to attain the maximum speed of 6000 rpm. The one side of the journal is connected to the belt drive and on the other side the test bearing is fitted. The elliptical bearing made of brass material is centrifugally casted and machined on Computerized Numeric Controlled machines to generate the slot having ellipse profile having minor axis = 100.2 and major axis = 100.4 mm. The radial load is applied on the elliptical bearing by pulling upwards with a loading lever operated through pneumatic cylinder. The test bearing moves relative to the shaft as the load and speed changes.

The pressure sensors (9 No's) are mounted on the back plate and tightened to the front face of the bearing at the middle position. O-rings are provided to ensure the leakage of lubricating oil. The cable ends of each pressure sensor is terminated with plug and connected to junction box. The temperature sensors (9 No's) are fitted on the circumference of the bearing with individual cable ends and also terminated with plug for connection to junction box. The loading lever is vertical and attached at the top portion of the bearing. The lubricating oil is supplied horizontally and two oil groves are provided on each side of the bearing. The used oil flows back to the tank from bottom of the housing. The oil flow is regulated by pressure gauge to flow between 0.5 to 2.0 litres per minute. The flow rate is acquired by digital flow meter. The oil pumped enters the heat exchanger to cool before coming into the bearing housing. The spindle housing is mounted above the frame for spindle to rotate in horizontal axis. The rear end of the spindle is fitted with reducer pulley rotated by AC motor through belt drive. The reducer pulley used to doubles the motor base speed. The other end of the spindle is used for test bearing. It is hard chrome plated and ground to 1.6 Ra for 100 mm long to seat the test bearing. The radial loading on the test bearing is applied by the pneumatic cylinder through a lever mechanism.

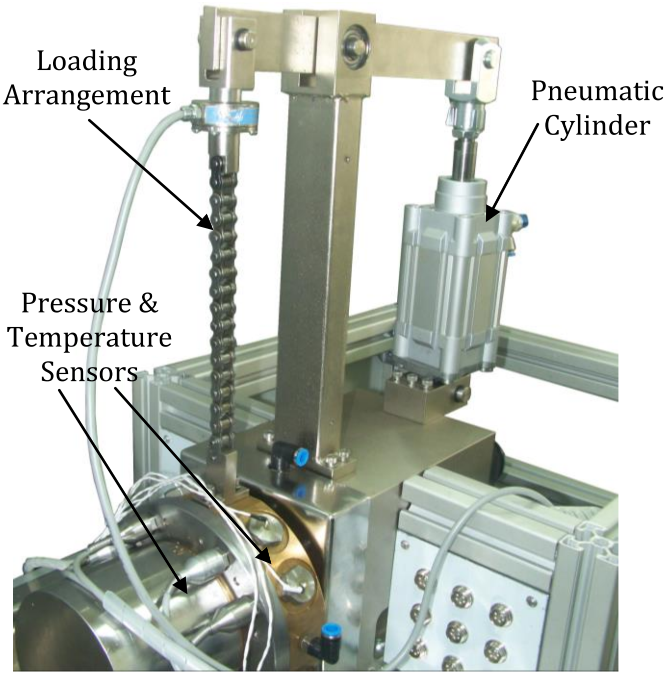

Fig. 2. Non-Circular journal bearing test rig for measurement of pressure and temperature at different load.

The lever is pivoted horizontally with 1:2 mechanical advantages on a vertical bracket. The loading end is connected to the top portion of the test bearing through a chain and the load cell. The free end of the chain is connected to the outside diameter of the bearing. When the load is applied on the bearing, it is pulled upwards while the pneumatic cylinder coupled at other end of lever pulls it downwards. The force required to pull the test bearing is measured by a load cell and is displayed as radial load on the software. The loading distance from the pivot is half the pneumatic cylinder distance to get a mechanical advantage of 1:2.

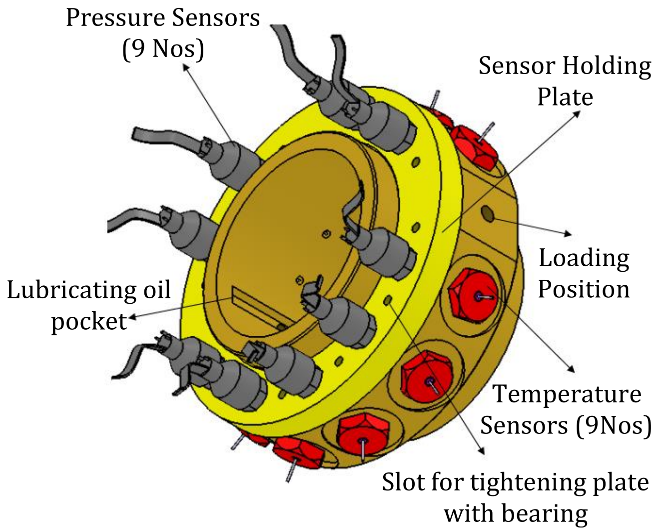

Fig. 3. The Elliptical test bearing equipped with pressure and temperature Sensor.

The controller of the test rig is software driven and is housed within the frame below the loading lever. The electrical incoming supply of 415 V, 3 ϕ and 25 A is sent to variable frequency drive (VFD) to control the spindle motor speed and another line of Liberating oil pump motor of re-circulating unit. 15 V supply is given to 4 No's of I-das cards used for excitation of all pressure and temperature sensors along with normal load and speed sensor. The output voltage from sensors are processed in the instrumentation portion of the I-das card and transmitted to the NI card. Therefore, the measured parameters by sensors which are processed in NI card are displayed on the PC screen through data acquisition system WINDUCOM 2010. The software is in lab view format and used for acquiring the data of the test result and also for the post processing of the data for graphical representation of various parameters.

The elliptical test bearing (Fig – 3) is provided with 12 numbers of ports at equal distance on circumference to clamp temperature sensor and also for clamping pressure sensor with help of back plate at the same location. Out of the 12 ports, 3 ports at 0°, 90° and 270° are used for applying load and oil inlet-1 and oil inlet - 2 respectively. All other ports are used for detecting the pressure and temperature of lubricating oil along the circumference of bearing.

The sensor used for pressure measurements is of that series of pressure transducers which are designed for high pressure and high accuracy and can be used in corrosive medium. The pressure sensor uses ultra stable technology that provides stability over a wide range of temperature. The ultra stable technology employs silicon based strain gauge isolated by an oil filled capsule and stainless steel diaphragm. The high stability rating is provided through MEMS based technology. 9 No's of digital stainless steel covered pressure sensor are fixed on the adaptor plate so that the same sensors can be used to test any type of bearing at the same location where the temperature sensor is employed. The adaptor plate is tightened to the face of the bearing with cables of individual sensors plugged to the panel of controller. A hole of diameter 5 mm connects test bearing inner circumference to each pressure sensor location. Fig 4 shows the position of the pressure and temperature sensor on the bearing.

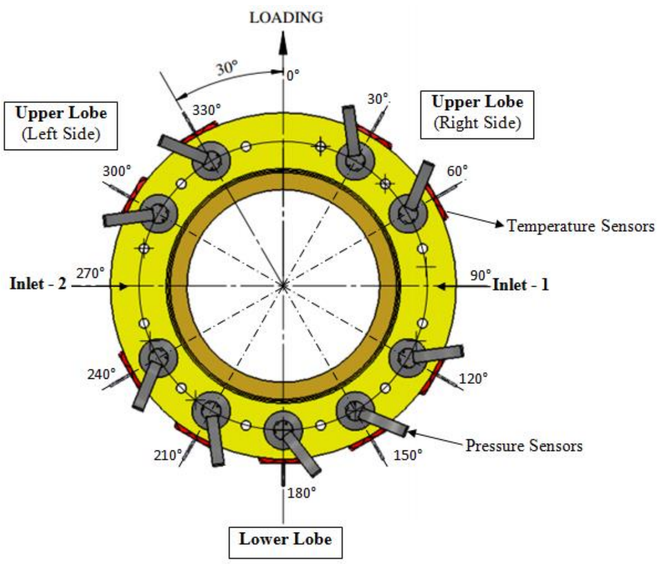

Fig. 4. Location and position of pressure and temperature sensor along the circumference of bearing.

The pressure sensor is tightened with oil filled inside the hole for a column of oil is retained between the inner circumference of the bewaring and tip of the pressure sensor. The signals generated during the test are transmitted to the instrumentation of the controller for processing the data. PT-100 temperature sensors are used for the measurement of temperature. 9 No's temperature sensors are fitted on the circumference of the bearing and one sensor is mounted to measure the inlet of lubricating oil. The temperature sensors are screwed into the test bearing from the circumference so that the tip of sensor is in contact with the internal lubricating oil of the bearing. The end cables of all sensors are plugged to the instrumentation controller for processing the data. A 100 kg tension-compression load cell with threads on either faces is screwed between loading lever and chain. The load cell is screwed to loading lever with a hinge and to adaptor on chain and the other end of the chain is tightened to the outer diameter of bearing. When the compressed air is allowed to enter into the pneumatic cylinder, the loading lever is pulled upwards and it pulls the bearing in the radial direction towards the journal. The pulling of lever exerts force on the bearing and this force is sensed by load cell and displayed as the normal load on the software screen.

3. EXPERIMENTAL PROGRAMME

During experimentation, the pressure and temperature of the lubricating oil has been evaluated along the circumference of an elliptical journal bearing at the same location for three different grades of oils named HYDROL 32, 68 and 150. These oils have been chosen based on the availability in the market. Also, Chauhan et al. [13] and Sehgal et al. [14] have also taken some of these rated oils for determination of temperature in offset and two lobe elliptical bearing profiles. For measurement of both pressure and temperature at the same location, all the pressure sensors have been mounted on separate holding plate which fastened along the circumference of the bearing. This also facilitates the changing of the bearing easily from the test rig. An elliptical bearing in which the slot was made based on the ellipse profile by specifying the major and minor axis was tested at various speeds and load for different grades of lubricating oil. The supply pressure was kept constant and the spillage of the oil is re-circulated to the oil tank was done during experimentation. The oil film pressure and temperature have been evaluated for elliptical journal bearing at different journal speed=2000, 3000, 4000 and 5000 rpm. The tests were conducted at varying load from 500 N to 2000 N in the steps of 500 N for the various journal speeds. The lubricating oil was supplied at 0.2 M Pa pressure and passed through two horizontal ports inside the bearing at an angle of 90° to the vertical loading line. The radial pneumatic load has been applied. Table 1 represents the dimensions of the test bearing under study and various input parameters of the study.

Table 1. Test Bearing Dimensions.

Outer Diameter of Bearing	120 mm
Internal elliptical hole of Bearing Major Axis (Horizontal Dimension) Minor Axis (Vertical Dimension)	 100.4 mm 100.2 mm
Journal Diameter, D	100 mm
Length of Bearing, L	100 mm
Radial Clearance, C	200 μ m
Minimum Clearance, C_{min}	100 μ m
Rotational Speeds, N	2000, 3000, 4000, 5000 rpm
Radial Load Applied, W	500, 1000, 1500, 2000 N

The test was conducted to obtain the pressure and temperate profile of lubricating oil inside the bearing for various rotational speed vary from 2000 rpm to 5000 rpm in the steps of 1000 rpm. These results have been obtained for three different lubricating oils named oil - 1, oil - 2 and oil - 3. The properties of these oils have been represented in Table - 2.

Table 2. Properties of Lubricating Oil*

	Oil – 1	Oil - 2	Oil - 3
Density at 15 °C	852	860	874
Kinematic Viscosity at 40 °C (centiStokes)	32.6	68.1	150.5
Viscosity Index	101	99	95
Flash Point, (COC), °C	220	238	256

* www.maklubes.com

4. RESULTS AND DISCUSSION

The pressure and the temperature of the oil film has been obtained for elliptical journal bearing at different loads 500 N, 1000 N, 1500 N and 2000 N for the three oils under study at various journal speeds. The effect of load on the pressure and temperature of the lubricating oil at fixed journal speed of 5000 rpm for different lubricating oil under study at different combination of load as specified in table - 1 have been represented in Figs. 5-10. Figures 5–7 represents the pressure profile of lubricating oil of the bearing at different loads at journal speed=5000 rpm for three different grades of lubricating oil.

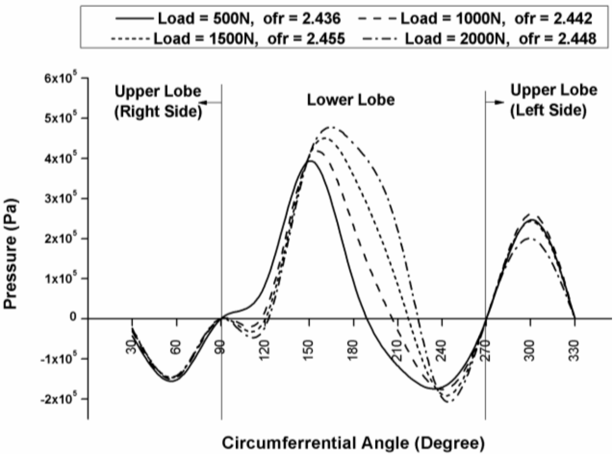

Fig. 5. Variation of pressure along the circumference of elliptical bearing at the central plane at 5000 rpm for different load = 500, 1000, 1500, 2000 N load for lubricating oil – 1.

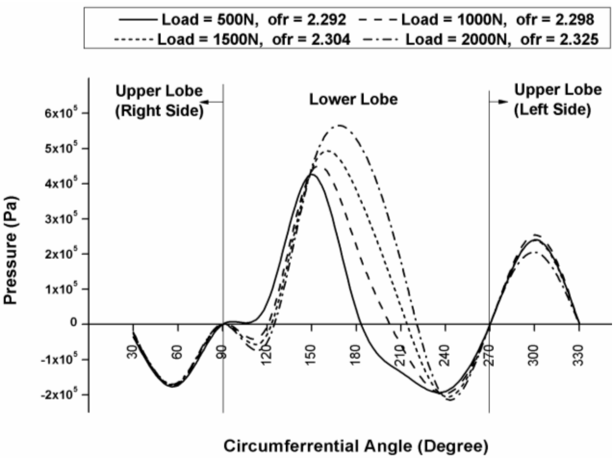

Fig. 6: Variation of pressure along the circumference of elliptical bearing at the central plane at 5000 rpm for different load = 500, 1000, 1500, 2000 N load for lubricating oil – 2.

It has been observed from the Figs. 5-7 that with the increase in load at constant speed the maximum pressure of the bearing rises for all the three grades of oil. Also, two positive pressure lobes has been observed based on the configuration of the elliptical bearing having active oil film in each lobe corresponding to two converging zone along the circumference of the journal.

The profile of film thickness from simulation analysis inside the clearance space of an elliptical journal bearing has been represented in Fig. 8. The significance of the film thickness provides the accurate variation of the pressure profile along the bearing. The maximum value of pressure would be occurred at the point of minimum film thickness. The similar variation has been obtained in the pressure plots obtained experimentally. It has also been observed that the range of positive pressure increases with the increase in load for all the three grades of oils. From the Fig. 5-7, it has been found that the negative pressure area decreases which implies reduced cavitation zone inside the bearing and accounts for better stability. The similar trends of pressure have been observed in the both upper and lower lobe for all the three oils under study and also for other journal speeds of rotation.

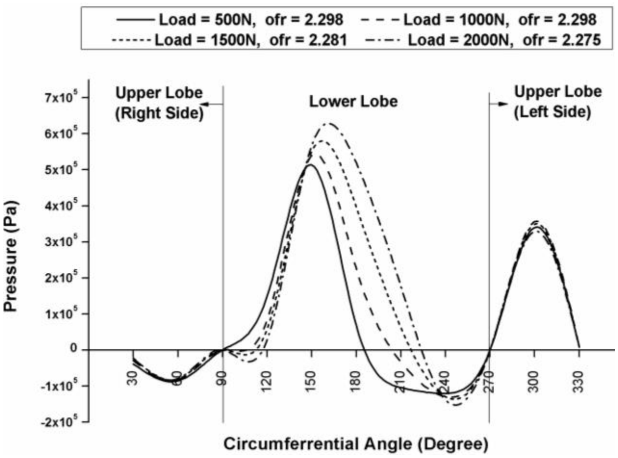

Fig. 7. Variation of pressure along the circumference of elliptical bearing at the central plane at 5000 rpm for different load = 500, 1000, 1500, 2000 N load for lubricating oil – 3.

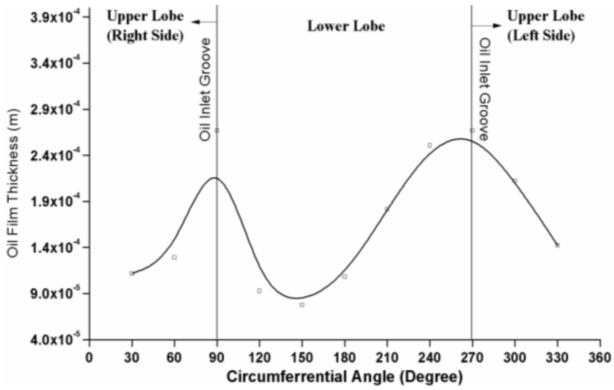

Fig. 8. Variation of oil film thickness plot for an elliptical journal bearing under study.

Figures 9-11 represents the variation of circumferential temperature of the lubricating oil at the central plane of elliptical bearing at journal speed = 5000 rpm. It has been observed that the temperature of the lubricating oil increases with the grade of oil – 1 and oil – 2 but a slight fall in temperature has been obtained for the grade of oil – 3 (Fig. 11). This fall in temperature with the increase in radial load may be due to very high viscosity of oil – 3 as compared to oil – 1 and oil – 2 also due to lower oil flow rate for the grade of oil – 3 as compared to lubricating oil of grade 1 and 2. The lubricating oil temperature is highest for oil - 3 at same speed and load and lowest for the oil - 1. This is due to the fact the oil - 3 is highly viscous as compare to oil – 1. The viscosity results in high frictional forces between the layers due to rubbing velocity which rises the temperature.

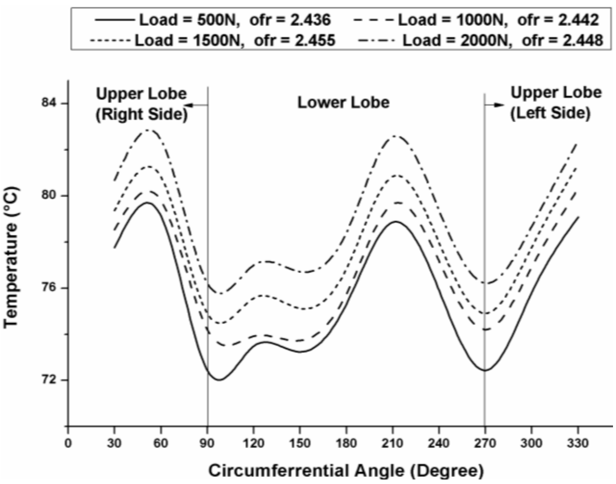

Fig. 9. Variation of temperature along the circumference of elliptical bearing at the central plane at 5000 rpm for different load = 500, 1000, 1500, 2000 N load for lubricating oil – 1.

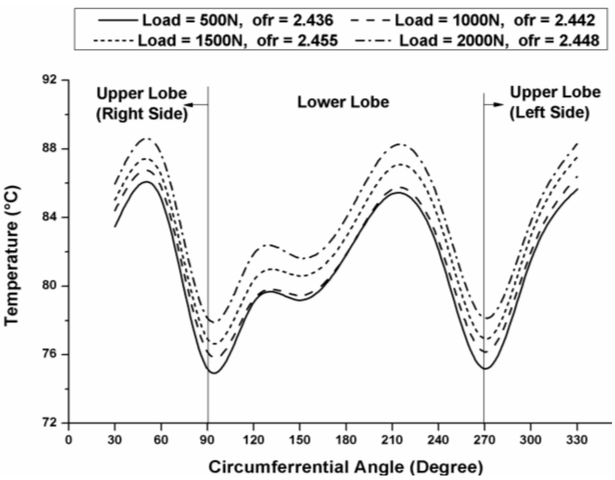

Fig. 10. Variation of temperature along the circumference of elliptical bearing at the central plane at 5000 rpm for different load = 500, 1000, 1500, 2000 N load for lubricating oil – 2.

The configuration of elliptical bearing has two active oil films which minimizes the cavitations zone (negative pressure) along the circumference of the bearing. This attribute accounts to more oil pump into the system. Therefore, both the lobes have converging and diverging zone which leads to rise in the temperature in both the zones. The similar trends have been observed in all the Figs. 5-7. Also the maximum temperature rise has been obtained in the lower lobe for all the grades of oil.

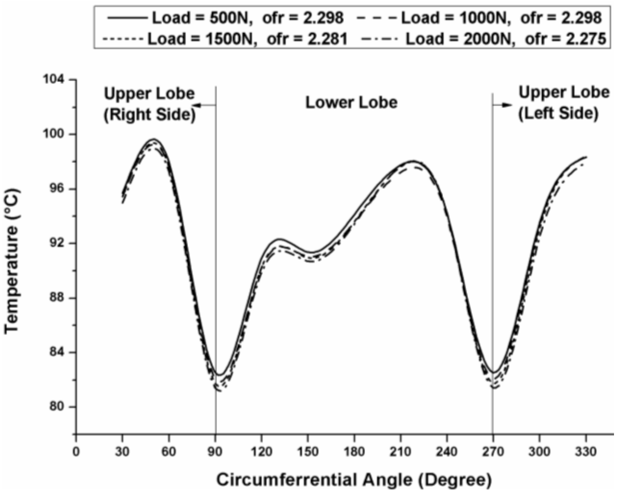

Fig. 11: Variation of temperature along the circumference of elliptical bearing at the central plane at 5000 rpm for different load = 500, 1000, 1500, 2000 N load for lubricating oil – 3.

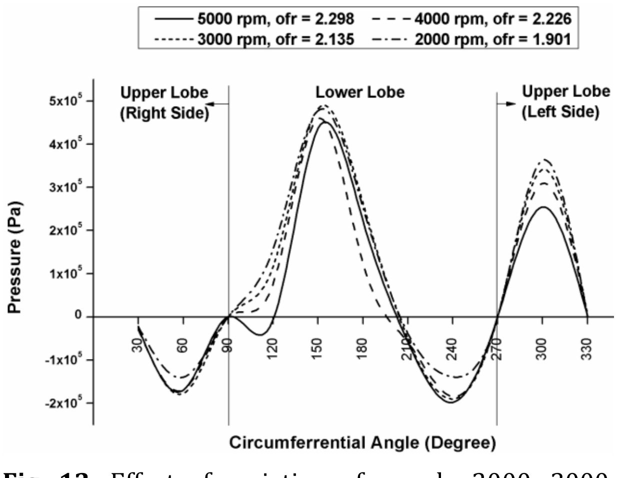

Fig. 12. Effect of variation of speed= 2000, 3000, 4000 and 5000 rpm on the lubricating oil pressure at the central plane of elliptical bearing for oil - 2 only.

The effect of variation of speed on the oil pressure at the central plane of elliptical bearing has been presented in Fig. 12 for lubricating grade of oil - 2 at fixed load of 1000 N. It has been observed that with the increase in speed from 2000 rpm to 5000 rpm there has been slight decrease in the oil pressure has been observed. This is because of the reason that there was increase in the oil flow rate due to increase in temperature of the lubricating oil with the increase in the journal speed.

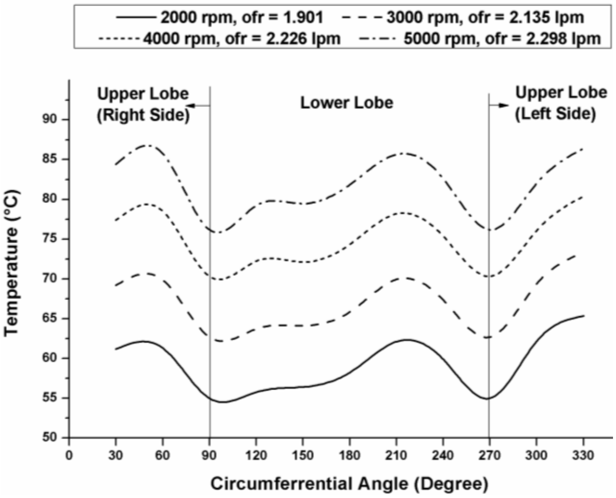

Fig. 13: Effect of variation of speed= 2000, 3000, 4000 and 5000 rpm on the oil film temperature at the central plane of elliptical bearing for oil - 2.

Figure 13 represents the effect of variation of speed on the oil film temperature for lubricating grade of oil - 2 at journal speed 2000, 3000, 4000 and 5000 rpm at 1000 N load. It has been observed from the experimentation that the temperature of the lubricating oil rises with the increase in speed in both the lobes of the bearing. The inlet temperature of the oil sump also increases with increase in journal speed. This is because the test rig is designed in such a way that the spillage of lubricating oil from the bearing is again re-circulating to the oil sump. Therefore the oil sump temperature increases correspondingly, the temperature of the oil at the inlet also increases.

4. CONCLUSIONS

A test rig has been developed to measure simultaneously both oil film pressure and temperature at the same location along the circumference of non-circular journal bearing. The pressure and temperature has been measured with the direct contact type sensors fitted on the bearing. The following conclusions were made from the various conducted experiments during the study.

i. The oil film pressure and temperature increases with the increase in load from 500 N to 2000 N at the fixed speed for all the three grades of lubricating oil. With the increase in load from 500 to 2000 N at constant journal speed=5000 rpm for lubricating oil - 2, there has been 11 % rise in pressure and 4 °C rise in temperature has been observed.

ii. The pressure and temperature of the oil film increases with the change of grade of lubricating oil at 5000 rpm of journal speed. The maximum rise in temperature of 8 °C has been observed for oil – 1 having kinematic viscosity 0.032 kg/m-sec; 11 °C rise for oil – 2 with kinematic viscosity=0.068 kg/m-sec and 17 °C rise in temperature has been obtained for oil – 3 having kinematic viscosity=0.15 kg/m-sec.

iii. Two positive pressure lobes have been found along the circumference of an elliptical bearing which reduces the cavitation zone and hence accounts for better stability. The trend of the pressure and the temperature profile is similar with the findings of the theoretical results presented by Hussain et al. [5].

In the nutshell, the results presented for evaluation of pressure and temperature profile of oil film inside the elliptical bearing will help the designer to choose the elliptical bearing as per the requirement of their data. Also, the results showed that the elliptical bearing will be an alternative to the circular bearing for better stability due to two positive zones of pressure along the circumference of the bearing.

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