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Vibrational and Tribological Analysis of Bearings with Experimental Method

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***Abstract*: The bearings are normally used components in machinery for a wide range of applications. Bearing is a mechanical element which permits relative motion between two parts, also used to support the shaft and takes up forces acting on shaft. It is crucial part of any rotary components and its failure causes disastrous failure of machinery. In this work, performance of hydrodynamic journal bearings with different groove geometry is investigated. This work carried out with two helical, three helical grooves and non-groove (plain) bearings, tested at radial load and RPM are mottled within specified range an experimental vibration signals were obtained. The bearing shows different behavior for different loading and speed conditions with same groove locations. Vibrational analysis is the effective tool for monitoring the conditions of bearing before they get failed. The vibration signal analysis has been carried out in Frequency domain by FFT analyzer. Results obtained by experimentation were in amplitude versus speed.The tribological performances of plain, two groove and three groove journal bearings were investigated in this dissertation work. Several experiments were performed under different static loads by using pin on disc tribometer. The tests were conducted at different speed and different loading conditions. The results were in the form of coefficient of friction.**

***Index Terms*-Coefficient of Friction, Frequency Domain, FFT Analyzer, Helical grooves, Pin on Disc Tribometer**

I. INTRODUCTION

THEhydrodynamic bearings are common components of rotating machinery. They are frequently used in applications involving high loads and high speeds between two surfaces that have relative motion. Hydrodynamic journal bearings are specific to surfaces that mate cylindrically with the applied load in the radial direction.In hydrodynamic lubrication small clearances between the journal and the bearing are present, and the smallest change in bearing or journal dimensions has an effect on many properties such as eccentricity, oil supply pressure, temperature rise etc. Vibration monitoring is one of the best techniques for checking condition of bearings. In this work, the vibration analysis of different grooves bearing with varying speed and load is studied. The oil hole and groove are provided for continuous supply of oil for avoiding the metal to metal contact between the bearing and journal. The three types of bearings are used, plain bearing (non-grooved), double groove and triple groove journal bearing.

For vibration analysis of bearing FFT was used. The amplitude versus time, amplitude versus frequency plots is drawn to study the vibration characteristics of bearings. The bearings used for testing has following dimensions, inner diameter of bearing 30 mm, outer diameter of bearing 46 mm, length of bearing 15 mm, clearance 0.02 mm. In general the groove length to bearing length ratio kept less than unity to avoiding leakage of oil. The lubricating oil used was SAE 20 W 40 engine oil. The tribology test was conducted on pin on disc tribometer at different speed and loading conditions.

II. LITERATURE REVIEW

V. N. Patel et al. investigated theoretical and experimental vibration study of dynamically loaded deep groove ball bearings. The defect of 60 μm diameter on either side of races is used for test. The deep groove ball bearings consist of local circular shape defects on either race. The shaft, housing, raceways and ball masses are combined in the proposed mathematical model. The model delivers the vibrations response for the balls, shaft and housing in time and frequency domains. The validation for proposed mathematical model is provided by experimentation. The experiment were conducted on testing bearing, test bearing is a deep groove ball bearing (Designation: SKF BB1B420205), which was mounted at the free end of the shaft towards right hand side of bearing. The electro-mechanical shaker is used to provide dynamic loading varying from 10-1000 Hz. The displacement of the ball changes from zero to maximum, when a ball approaches to the inner race defect, while, it grasps to zero from its extreme value, when ball reaches from the centre of the defect to the another end of the defect. In defective inner race, characteristic defective frequency along with the side bands at shaft rotation frequency is noticed. The further vibrations are also observed due to the noise added by the electro-mechanical shaker [1]. The hydrodynamic journal bearings are used widely in high speed rotating machine such as compressors, gas turbines, water turbines, steam turbines, alternators etc. As rotor rotates at high speed, the flow between journal and bearing does not remain laminar. The analysis is carried out for the case of short bearing approximation aspect ratio (L/D<0.5) under different flow regime i.e. laminar, transition and turbulent flow condition assuming the perfectly rigid journal and bearing. Eccentricity ratio decreases when Sommerfeld number increases for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and for constant sommerfeld number eccentricity ratio decreases, when flow changes from laminar to transient, transient to turbulent. The minimum oil film thickness is directly proportional to sommerfeld number for laminar, transition and turbulent flow regime and at constant sommerfeld number minimum film thickness increases when flow regime changes from laminar to turbulent flow. The attitude angle increases with rise in Sommerfeld number for laminar, transition and turbulent flow regime of hydrodynamic journal bearing and at a constant Sommerfeld number attitude angle rises whilegoing from laminar to transition to turbulent flow. From this study it has been seen that, as the fluid flow from laminar to turbulent the minimum film thickness, attitude angle, direct and cross coupled damping coefficients increases for a constant Sommerfeld number. However, eccentricity ratio decreases as the fluid moves in laminar to turbulent flow [2]. Mohamad Ali Ahmad et al. investigated the effect of oil supply pressure on circumferential pressure profile in hydrodynamic journal bearing. For hydrodynamic lubrication, the pressure condition of the fluid is difficult to ensure good performance of the lubricated machine elements such as journal bearings. In this work, an experimental work was conducted to determine the result of oil supply pressure on pressure profile around the circumference of a journal bearing. A bearing with journal diameter of 100 mm and length to diameter ratio (L / D) of 0.5 was used. The three different values of supply oil pressure (0.3, 0.5, 0.7 MPa) are used and the results are taken at three different speed (400, 600, 800 RPM) conditions by changing radial load also. The short angle groove is used in present work. The bearing used for testing of specifications diameter 100 mm, length 50 mm, radial clearance 52 µm. The oil supply pressure affects the pressure distribution for the case of lower loads, 5 kN and 7kN. The pressure distribution is does not affected by journal speed. The maximum pressure values increases with increase in load. For higher load (10 kN) pressure profile is constant for different speed and oil supply pressure values [3]. Salmiah Kasolang et al. studied effect of oil supply pressure at different groove position on friction force and torque in journal bearing was studied. An axial groove is a mostly used method supply for distributing lubricant within a journal bearing. The lubricant is fed at certain supply pressure to ensure that journal is separated from bearing. The shearing action between lubricant and bearing parts produces friction which contributes to power loss in the bearing. In this study, experimental work was accompanied to determine the effect of oil supply pressure at different oil groove places on torque and frictional force in hydrodynamic journal bearing. The journal bearing with 100 mm journal diameter and 0.5 mm length to diameter ratio was used. The supply oil has pressure values of 0.2, 0.5, 0.7 MPa. The groove was positioned at 7 different locations of -450, -300, -150, 00, +150, +300 and +450. The friction force, coefficient of friction and torque was measured for speed values of 500 and 800 rpm at 10 and 15 kN radial loads. In hydrodynamic analysis, the oil supply was expected to flow into the bearing at least as fast as it leaks out. The 1000 rpm was the maximum speed used for testing. Torque and frictional force tend to changes with change in groove position takes place. At certain positions increasing oil supply pressure will increase torque and frictional force of the bearing. Fluid friction coefficient of 15 kN load is higher compared to that of 10 kN. The groove position of - 300 has a lower friction coefficient value for speed values of 500 and 800 rpm [4]. The conventional hydrodynamic journal bearing performance tools cannot suitably measure the effect of lubricant feeding conditions on bearing performance, although these conditions affect important performance parameters such as eccentricity and power loss. In this work, lubricant supply pressure and temperature groove length ratio and number of grooves are analysed. The increase in lubricant feeding temperature proved to be beneficial under low loads (it decreased power loss) butespecially harmful under high loads as it strongly increases the eccentricity, T_{max}, P_{max}, and the thermal and mechanical distortions, while reducing the critical load for which hot oil reflux starts occurring. As smaller length grooves used (small groove length to bearing length ratio), it decreases in power loss around 35%, but at the expense of a less efficient bearing cooling. The increase of w/d (groove width to bearing diameter ratio) induced a reduction in power loss and maximum bush temperature without a significant decrease in load carrying capacity. As single groove and double groove bearings are compared, it was found that in double groove bearings, temperature level and eccentricity get decreased for higher load range [5]. Hakan Adatepe *et al*. investigated frictional behavior of statically loaded micro-grooved journal bearing. The effects of interface friction on frictional behavior of plain and micro-grooved bearing were studied. The coefficient of friction reduces with increased bearing parameters in the boundary lubrication regime. It increased with increase in bearing parameter in full film lubrication regime. The transverse groove bearing shows highest value of coefficient of friction and then circumferential and non-grooved. The higher value of coefficient of friction obtained in transverse grooved bearing due to interruption of oil flow offered by transverse cut grooves [6]. The theoretical calculation method and performance of micro-grooved bearing was studied by Kenji Watanabe *et al*. The minimum oil film was thicker in micro-grooved bearing than in plain bearing. The oil flow in micro-groovedbearing is higher than that in the normal plain bearing and lesser frictional torque obtained in micro-grooved bearing. The durability of micro-grooved bearing was affected by minimum oil film thickness [7].

III. Objectives

As the system shows different vibrating nature at different loading and speed conditions,

- To investigate the effect groove geometry on mechanical vibrations of journal bearing.
- To investigate tribological behaviour of groove geometry on journal bearing.

IV. Methodology

A. Experimental Setup

To carry out the proposed work different tests are conducted on Pin on Disc Tribometer and FFT. The layout of experimental setup is as shown in Fig. 1.

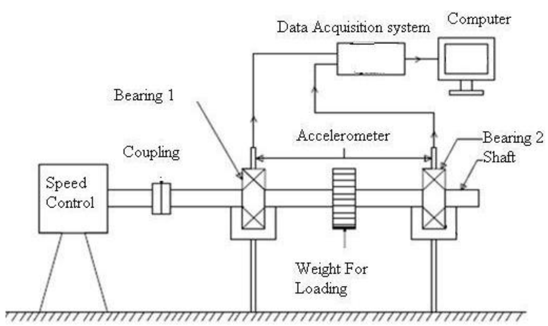

Fig. 1. Experimental setup

It mainly consist of, two journal bearings, motor for driving, coupling, pedestal bearing at the centre of shaft for applying load.The Fig. 2 shows setup used for tribological testing.

Fig. 2. Pin on disc tribometer

It consist of loading assembly, Display, Pump for oil supply, Winducom software etc.

B. Experimentation

The three different bearings are used for this work, plain (non-grooved), two grooved and three grooved. Bearings of outside diameter 46 mm, inside diameter 30 mm, radial clearance 0.01 mm and length of bearing 15mm were used. The non-grooved bearings were tested for different speed and loading conditions. The accelerometer was connected on bearings for taking readings.The first reading was taken for 3 kg load and 600 rpm, continuous oil supply is provided for bearings.Again for same load, readings are taken by changing speed to 900 rpm and 1200 rpm.The same procedure is repeated for 5 kg and 8 kg load. The speed can be varied by rheostat and for changing load directly load applied on pedestal bearing. The amplitude and frequency for each reading is note down and graph of amplitude versus frequency was plotted.The FFT used for this work has vibration level of 20-2000 Hz. It has input accuracy of $\pm$ 2 % at 1 kHz. The measuring accuracy of amplitude is 1 % for frequency range of 5 Hz to 20 kHz and sampling frequency is greater than 200 kHz.

The same bearings are used for tribological testing and Pin on Disc Tribometer is used for such analysis. Firstly the disc of material EN24 and pin of material Bronze are prepared and mounted on tester. The dimensions of disc, diameter 165 mm and 12 mm thick and that of pin is 12 mm. The maximum loading capacity of machine is 200 N. After it wear and frictional force reading on display set to zero by proximity sensors and required load is applied. The lubrication started by starting lubricating pump. By applying 3 kg load and 600 rpm reading was taken. The display shows corresponding value of frictional force in newton and wear in micrometer. The same procedure is repeated for 5 kg and 8 kg load.

V. Results And Discussion

The Table I show results of tribological test on different groove bearings.The Fig. 3 shows graph of coefficient of friction versus speed at 3 kg load. The value of frictional force is directly taken from display and coefficient of friction is calculated as follows,

$$COF = FF \div Load \tag{1}$$

TABLE I

COEFFICIENT OF FRICTION

Load(kg)	Speed (RPM)	Frictional Force (N)	Coefficient of Friction (COF)
	600	1.38	0.047
3 kg	900	0.79	0.027
	1200	1.70	0.058
	1500	1.85	0.063
	600	2.30	0.047
	900	2.20	0.045
5 kg	1200	1.22	0.025
	1500	2.20	0.045
	600	3.29	0.042
	900	2.11	0.027
8 kg	1200	2.90	0.037
	1500	2.82	0.036

The Fig. 4 shows graph of coefficient of friction versus speed at 5 kg load. The maximum value of coefficient of friction, 0.063 is observed at 1500 rpm and minimum value of 0.027 observed at 900 rpm.

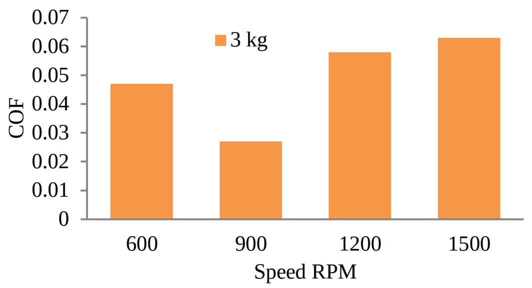

Fig. 3 Speed vs coefficient of friction at 3 kg load

As shown in the graph represented in Fig. 4 the coefficient of friction was more for 600 rpm and that is less for 1200 rpm at 5 kg load. The maximum and minimum value observed was 0.047 and 0.027.

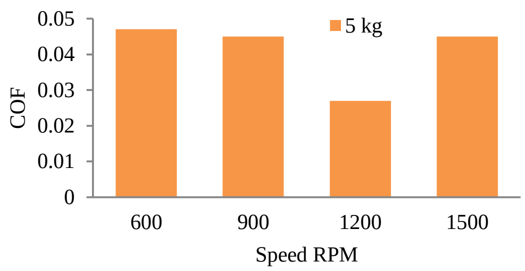

Fig. 4 Speed vs coefficient of friction at 5 kg load

The Fig. 5 shows graph of coefficient of friction versus speed at 8 kg load.The value of coefficient of friction at 600 rpm is maximum at 600 rpm and it was 0.042 . The minimum value of that was at 900 rpm and it was 0.027.

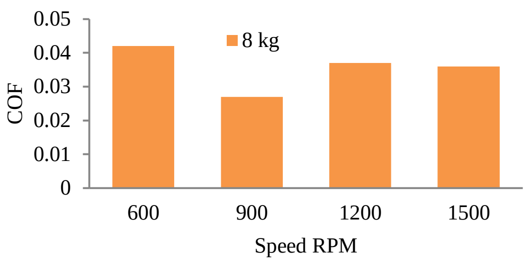

Fig. 5 Speed vs coefficient of friction at 8 kg load

The results obtained for vibration analysis at different speeds and 3 kg load for plain bearing was tabulated in Table II.

TABLE II

PLAIN BEARING AT 3 KG LOAD

Speed (rpm)	Drive		Drive End	
	Amplitude (µm)	Frequency (Hz)	Amplitude (µm)	Frequency (Hz)
600	16.5	5	5.60	4
900	24.1	7	2.47	6
1200	10.5	4	4.17	5
1500	8.0	3	3.50	4

The Table III shows results obtained for two groove bearing at 3 kg load and various speeds.

TABLE III

TWO GROOVE BEARING AT 3 KG LOAD

Speed (rpm)	Drive		Drive End	
	Amplitude (µm)	Frequency(Hz)	Amplitude (µm)	Frequency(Hz)
600	2.70	7	1.16	10
900	2.80	8	1.92	9
1200	4.31	6	1.65	14
1500	7.65	8	2.57	22

The results obtained for vibration analysis at different speed and 3 kg load for three groove bearing was tabulated in Table IV.

TABLE IV

THREE GROOVE BEARING AT 3 KG LOAD

Speed (rpm)	Drive		Drive End	
	Amplitude (µm)	Frequency(Hz)	Amplitude (µm)	Frequency(Hz)
600	2.27	16	1.40	40
900	5.90	20	2.80	20
1200	8.00	18	4.40	25
1500	11.40	24	5.70	24

Graph of result tabulated in above tables is shown below in Fig. 5. The maximum amplitude of vibration is observed at 900 rpm for plain groove bearing and the minimum value of amplitude of vibration was observed at 900 rpm for three groove bearing.

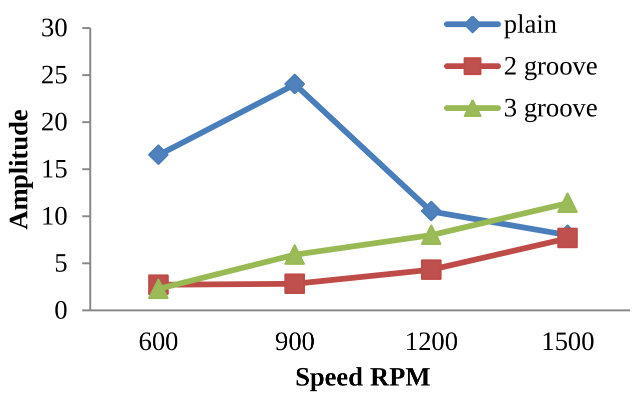

Fig. 5 Speed vs amplitude at 3 kg load

The maximum amplitude of vibration was 24.1 µm and minimum value observed is 2.27 µm. The amplitude of vibration for plain bearing and three groove bearing fluctuates more as speed and load is changed. But the two groove bearing gives nearly linear results with change in speed and load. The two groove bearing gives optimum results at 3 kg load.

VI. CONCLUSION

A comparison of performanceof plain (non-grooved), two grooved and three grooved hydrodynamic journal has been carried out. This work tried to evaluate the effect of different grooves, load and speed on tribological characteristics and vibration of bearing.

- The amplitude of vibration for non-grooved bearing decreases with increase in speed as compared with other two bearings.
- At speed up to 1000 rpm two helical groove journal bearing has less amplitude of vibration than other bearings.
- As the speed is limited to 1000 rpm and load goes on increasing three groove helical bearing shows better performance.
- The coefficient of friction is more at the start and decrease with increase in speed and again increases with further increase in speed takes place.

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