International Journal of Advanced Research in Science Management and Technology
ISSN(Online)-2454-4159 Volume 4, Issue 2, February 2018

"Experimental investigation of lobe journal bearing with different grade oil"

Mr. Chandrakant B. Patil[1], Prof. Sunil V Yeole[2]

PG Scholar, Department of Mechanical Engineering, RCPIT Shirpur, Maharashtra, India[1]

Assistant Professor, Department of Mechanical Engineering, RCPIT Shirpur, Maharashtra, India[2]

Abstract— A journal bearing consist of a shaft (or journal) which rotates freely in a supporting metal sleeve or shell and has no rolling elements. Shell type journal bearings support radial loading only, therefore construction and design is simple whereas understanding of theory and operation may be complex. Its extensive applications are in the field of turbo machinery components such as pumps, compressors and gas turbines as well as in automobile suspension system (transmission and suspension system). The performance of journal bearing depends on pressure distribution, load carrying capacity, frictional force, and speed, viscosity of oil etc. In this thesis, an attempt has been made to present three different commercial lubricant oils such as SAE10W30,SAE 15W40 and SAE 20W40 and their role in determining condition of lubrication (hydrodynamic or hydrostatic), type of pressure distribution and the stresses developed in the bearing under varying loads and speeds. The maximum pressure and Change of viscosity (before and after test) were determined and compared. The maximum stress developed at maximum pressure was 2040 KPa. It is also noticed that, there is an increase in pressure from 0 Kpa to 800 Kpa and then decreases to minimum for remaining rotation of bearing. This clearly indicates the presence of hydrodynamic lubrication in the journal bearing and good to use at higher speeds. It has been found that SAE10W30 bears more load than SAE 15W40 and SAE20W40. Viscosity change shows that temperature has no or little effect. Results prove that an alternate and cheaper lubricant SAE graded oils can be used for improving performance of journal bearing.

Keywords— Hydrodynamic bearing, Viscosity, Pressure distribution, SAE graded oils, Sommerfield number, Lubrication.

I. INTRODUCTION

Journal or plain bearings consist of a shaft or journal which rotates freely in a supporting metal sleeve or shell. There are no rolling elements in these bearings. Their design and construction may be relatively simple, but the theory and operation of these bearings can be complex. This article concentrates on oil- and grease-lubricated full fluid film journal bearings; but first a brief discussion of pins and bushings, dry and semi lubricated journal bearings, and tilting-pad bearings. Plain journal bearings in hydrodynamic regime encounter instability problems of whirl and whip at high speed. Such instability may damage not only the bearings but also the complete machine. Lubrication reduces friction between two surfaces (such as sliding surfaces of a bearingand a shaft) in relative motion. It is typically categorized as boundary, mixed and hydrodynamic lubrication, for example by Heywood (1988), Becker (2004) and Gleghorn and Bonassar (2008). When a journal bearing operates under boundary lubrication, the sliding surfaces of the bearing and shaft are practically in direct contact and friction is at its highest level. Lower friction levels are achieved through the use of mixed lubrication, where the sliding surfaces are partially separated by the lubricant, and of hydrodynamic lubrication, where the sliding surfaces are completely separated by the lubricant. Fluid film bearings usually experience a considerable variation in temperature due to viscous heat dissipation. Thermal changes associated with viscous heat dissipation significantly affect the bearing performance since lubricant viscosity is a strong function of temperature. Moreover, excessive rise in temperature can cause oxidation of the lubricant and, consequently, lead to failure of the bearing. It has been reported in the past that the temperature rise is quite high in circular bearings as they operate with single active oil film. Researchers have tried to study the behavior of the different forms of lubricating film by adopting various approaches of simulation in accordance with the real conditions. This eventually encouraged the researchers to investigate fluid film bearings with non-circular profile, which operate with more than one active oil films. This feature accounts for the superior stiffness, damping and reduced temperature in oil-film as compared to the conventional circular journal bearings. The non-circular bearing geometry enhances the shaft stability and under proper conditions this will also reduce power losses and increases oil flow (as compared to an inscribed circular bearing), thus reducing bearing temperature. Out of non-circular journal bearings, elliptical, offset-halves and three-lobe configurations are the most common. Amongst these non-circular bearings the elliptical bearings are commonly used in turbo-sets of small and medium ratings. The so-called elliptical bearing is not actually elliptic in cross section but is usually made up of two circular arcs whose centers are displaced along a common vertical straight line from the centre of the bearing, as shown in Fig. 1. The thermo hydrodynamic (THD) model that considers thermal energy flow even across the thin lubricant film is well established for circular journal bearing. However, for non-circular bearings the published information is very limited, thus leading to the non-availability of the standard procedures for their design. Journal bearing finds an extensive application in the field of turbo machinery components such as pumps, compressors and gas turbines. It also finds application in automobile system such as transmission and suspension system. Journal bearing consists of the following parts as shown in Figure 1.

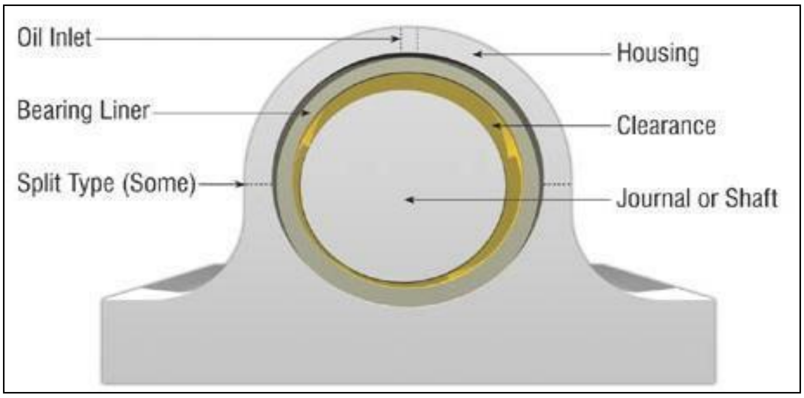

Fig.1 Different components of journal bearing

II. LITERATURE REVEW

The fluid film pressure and temperature distribution is one of the fundamental operating parameters to identify the operating conditions of journal bearing. The pressure distribution is crucial in load capacity estimation as well as dynamic analysis. In fluid film journal bearing, viscous shearing phenomenon occurs, that causes power loss and temperature rise. Rising temperatures lead to viscosity reduction of oil and bearing deformation. Hence it is needed to study pressure and temperature distribution in journal bearing. Experiments related to calculation of pressure and temperature inside the bearing of lubricating film gives the real scenario for calculation of performance parameters of the bearing. Only a limited experimental work has been reported for the thermal effects of elliptical bore journal which is two lobed bearing. This leads to unavailability of standard procedures for their design. Here, only few recent experimental and theoretical works has been reviewed.

Pinkus and Lynn [1] have theoretically presented an analysis of elliptical journal bearings based on the numerical solution of Reynolds equation for finite bearings with the assumption that there is no heat loss to the surroundings. The solution of the differential equation was supplemented by additional work on the nature of the oil flow, power loss, and eccentricity in elliptical bearings.

Fitzgerald and Neal [2] have presented some experimental data for 76 mm diameter two-axial groove circular bearings. The authors have observed that the axial temperature variation was negligible but the circumferential temperature variation could be very significant.

Hussain et al. [3] have predicted the temperature distribution in non-circular journal bearings: two-lobe, elliptical, and orthogonally displaced. The prediction of temperature by the authors is based on a two-dimensional treatment following McCallion's approach (an approach in which the Reynolds and energy equations in oil film are decoupled by neglecting all pressure terms in the energy equation).

Mishra et al. [4] have considered the non-circularity in bearing bore to be elliptical and made a comparison with the circular case to analyze the effect of irregularity. It has been 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.

Ostayen and Beek [5] have carried out a finite element analysis of a lemon-bore journal bearing. The thermo-hydrodynamic model presented by the authors is an inverse model, that is, a model in which the shaft eccentricity and attitude angle are calculated given a certain known and prescribed load and load angle. In analysis carried by the authors, care has been taken to accurately model the heat to and from the oil supplied and the model is used to check the design of the lemon bearings in a specific naval application.

Chauhan et al. [6] have presented a comparative theoretical analysis of three types of hydrodynamic journal bearing configurations namely: circular, axial groove, and offset-halves. It has been observed that the offset bearing runs cooler than an equivalent circular bearing with axial grooves.

AmitChauhan et al [7] have presented thermo-hydrodynamic analysis of plain journal bearing. During the analysis, deviation of pressure and temperature is considered on the fluid film. From result it is observed that putting constant viscosity during analysis may give incorrect prediction about the bearing. So the present paper gives future prediction of bearing performance

S. Baskar and G. Sriram [8] observed pressure distribution on hydrodynamic journal bearing using SAE20W40, rapeseed oil and soybean oil for loading condition such as 300N and 450N and speed ranges such as 1500rpm and 1750rpm. The Bearing is tested using journal bearing test rig. (JBTR) for different vegetable oils and compared the results with SAE20W40. And it is calculated that only 10% to 20% and 50% to 75% pressure distribution variation occurred in rapeseed oil and soybean oil. Hence Soybean oil generated more heat as compared to rapeseed oil, so the soybean oil is not suitable as a lubrication purpose for journal bearing applications.

NabarunBiswas and PrasunChakraborti [9] worked on unsteady transient analysis of 3lobe journal bearing. Author used physical properties SAE-50 for analysis purpose in journal bearing. They considered surface roughness as 0.9 and motion of shaft at 6000 rpm for analysis various flow parameters. Gambit is used for design and analysis is done with the help of fluent .They study six time steps 10, 30, 50, 70, 90, and 110 sec for unsteady analysis and found out after 110 sec the flow becomes steady. At minimum oil film thickness maximum pressure is observed with increasing value of roughness.

Peeyush vats et al [10] have done case study on heat transfer through journal bearing. This paper presented thermal analysis of journal bearing. The author has used FEM analysis to found out heat generated, temperature distribution and heat dissipation throughout the journal bearing. Theoretical as well as FEM analysis have done for journal bearing. From results it is showed that difference between heat dissipated and heat generated in oil film was very large, because of this temperature of the bearing rises and damaged the bearing pads.

K. M. Panday et al[11] have done unsteady analysis for thin film lubricated journal bearing with different L/D ratios such as 0.25, 0.5, 1, 1.5, and 2. During the analysis, author observed maximum pressure present at minimum oil film thickness. Also they found out that shear stress on surface of bearing and journal is reduced with increase in L/D ratio, but the turbulent viscosity of lubricant rises with increase in L/D ratio.

ArjunPanthi et al [12] have studied pressure and temperature effects on viscosity also found out effect of L/D ratio, rotational speed and eccentricity ratio on pressure distribution on bearing. The analysis has done using CFD tool and results obtained from the software validated with numerical results got from using Raimondi and Boyd chart method. From results, it is predicted that increasing temperature raises pressure but decreases of attitude angle.

Chaitanya K. Desai et al[13]have done experimental as well as theoretical analysis of pressure distribution for various loading conditions and working parameters in hydrodynamic journal bearing. From results it is concluded that maximum pressure generated where fluid film thickness is minimum and at cavitations zone zero pressure occurred, also increasing speed and load on bearing increases the pressure.

Hussain et al. [14] have predicted the temperature distribution in non-circular journal bearings: two-lobe, elliptical, and orthogonally displaced. The prediction of temperature by the authors is based on a two-dimensional treatment following McCallion's approach (an approach in which the Reynolds and energy equations in oil film are decoupled by neglecting all pressure terms in the energy equation).

Chun [15] examined the effect of Variable specific heat on maximum pressure, maximum temperature, bearing load, frictional loss journal. Now when the journal starts rotating, then at low speed and side leakage in high-speed journal bearing. Film pressure, load-carrying capacity attitude angle, end leakage flow rate, frictional coefficient and misalignment moment were calculated for different values of misalignment degree and eccentricity ratio. It was found that there are obvious changes in film pressure distribution, the highest film pressure, film thickness distribution, the least film thickness and the misalignment moment when misalignment takes place.

Sep [16] investigated the oil flow velocity, pressure and temperature distribution of a journal bearing with a two-component surface layer. After all such studies, still the case is far from closed. There are a limited number of studies that carry out an in-depth examination of the true operating conditions of bearings in actual experiments upon the test rig with different oils.

Baskar S and Sriram G [17]invstigated to analyze the pressure distribution on hydrodynamic journal bearing under different lubricants for various loading conditions and various operating parameters. The space between the shaft and the bearing is filled with different lubricants. Journal bearing test rig (JBTR) is used to test the 40 mm diameter and 40 mm long bearing. Test bearing is located between two antifriction bearings and loaded mechanically. At first the bearing is tested in JBTR under SAE20W40 of various load conditions such as 300N and 450N and speed ranges such as 1500 rpm and 1750 rpm and the pressure distribution results were observed and recorded. The vegetable oils such as rapeseed oil and soya bean oil were tested under the same operating conditions in the JBTR and the results are compared with the SAE20W40.

Mane RM, Soni S. [18]investigate the 3D model of hydrodynamic plain journal bearing using COMSOL Multi physics 4.3a software. Using 3D Model, pressure distribution in plain journal bearing is obtained by steady state analysis of plain journal bearing. Generalized Reynolds equation is used for analyzing hydrodynamic journal bearing by COMSOL as well as by analytical method by applying Sommerfeld boundary conditions. This Reynolds equation is solved for two theories of hydrodynamic journal bearing called infinitely short journal bearing and infinitely long journal bearing. Results obtained for pressure distribution simulation are compared with analytical results shows that the solutions are approximately similar to the analytical solutions. The comparison is quite decent for both version of the plain journal bearing, because the analytical and simulation results correspond to each other. Only the magnitude of the pressure build-up/drop differs, but this can be caused by the analytical assumptions and the software approximation.

Baskar. S., Sriram.G.[19] investigate the hydrodynamic journal bearings lubricated with synthetic lubricant and chemically modified rapeseed oil based bio lubricant dispersed with nanoCuO as environmental friendly vegetable oil based lubricants. The rapeseed oil was chemically modified via epoxidation, hydroxylation and

International Journal of Advanced Research in Science Management and Technology
IJARSMT
ISSN(Online)-2454-4159 *Volume 4, Issue 2, February 2018*
etherification process to improve its thermo oxidative stability and the pour point. The journal bearing test rig (JBTR) was used to measure pressure distribution developed in the circumference of journal bearing. Journal bearing test rig (JBTR) was used to test bearing diameter and length of 40 mm under the load and speed of 450 N &1000 rpm respectively under various lubricants. The experimental result obtained reveals that Nano added bio lubricant has good agreement with SAE20W40.

Y.Y. Wu et al. [20] examined tribological properties of lubricating oils an API-SF engine oil and base oil with CuO, TiO2 and Nano-Diamond nanoparticles Jadhav et al., International Journal of Advanced Engineering Technology E-ISSN 0976-3945 Int J AdvEngg Tech/Vol. V/Issue I/Jan.-March.,2014/01-04 used as additives. Friction and wear experiments were performed by using reciprocating Tribometer. CuO added in standard oil exhibit good friction reduction and anti-wear property. The additions of CuO nanoparticles in the API-SF engine oil & the base oil decreased the friction coefficient by 18.4 and 5.8% respectively, and reduced warn depth by 16.7 and 78.8% respectively as compared to the standard oils without CuOnano particles. The anti-wear mechanism is attributed to the deposition of CuOnano particles on the worn surface, which may decrease the shearing stress, thus improving the tribological properties.

III. EXPEREMENTATION

The pressure distribution of journal bearing is investigated with help of Journal Bearing Test Rig (manufactured by Ducom Industries Ltd, Bangalore) in a lubricant under load condition. The JBTR equipment consists of a vertically mounted shaft and driven by a variable speed motor. A metallic bellow connects brass bearing on the bottom and top is fixed to the frictional torque load cell. The bearing is made of brass material encloses the shaft at the lower end and is immersed in an oil sump. A stepper motor moves the bearing in the direction of the rotation of the shaft onto 180o in steps of 9o. A pressure sensor is fixed to the bearing, which measures the film pressure distributed in the oil film. The lever arm is used to apply radial load by dead weights. The assembly of the shaft and the bearing is immersed in oil, so as to provide continuous lubrication at all times. The equipment is connected to the controller, which displays the values of the angular position of pressure sensor with reference to the load line and the corresponding pressure values. The data obtained are transmitted to PC through data acquisition cable.

Pressure value for 800 rpm speed and 300N load condition for Oil 10W30.

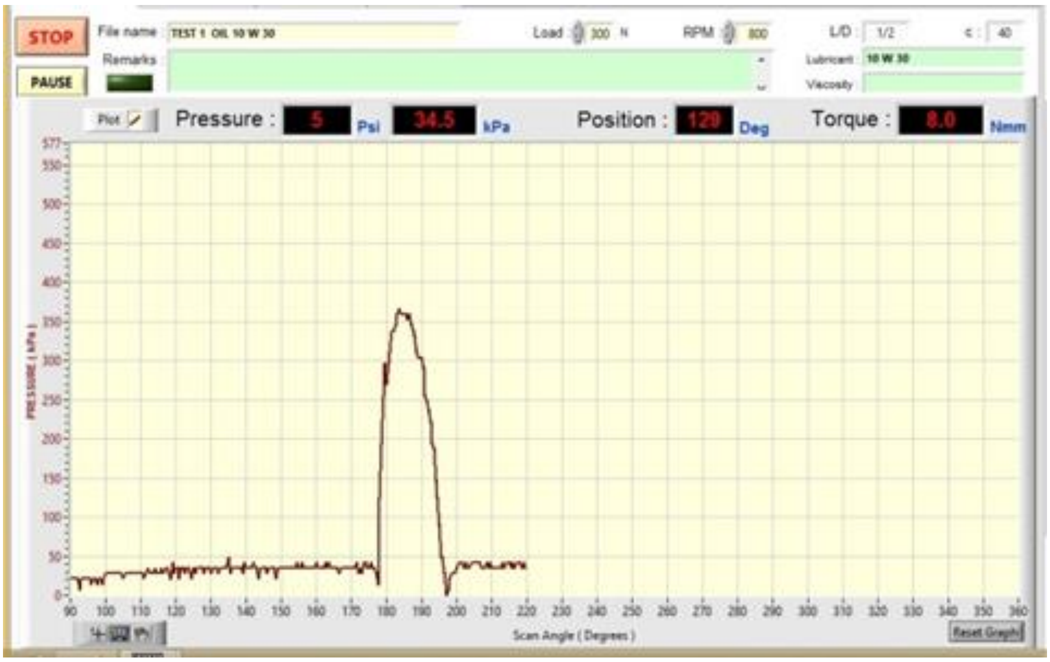

Fig.2 Variation of pressure with angular position and load condition at 800 rpm and 300N Load Grade Oil SAE 10W30

Pressure value for 1000 rpm speed and 450N load condition for Oil 10W30

Fig.3 Variation of pressure with angular position and load condition at 1000 rpm and 450N Load Grade Oil SAE 10W30

Pressure value for 1200 rpm speed and 600N load condition for Oil 10W30

Fig.4 Variation of pressure with angular position and load condition at 1200 rpm and 600 N Load Grade Oil SAE 10W30

Pressure value for 1000 rpm speed and 450N load condition for Oil 15W40

Fig.5 Variation of pressure with angular position and load condition at 1000 rpm and 450N Load Grade Oil SAE 15W40

Pressure value for 1200 rpm speed and 600N load condition for Oil 15W40

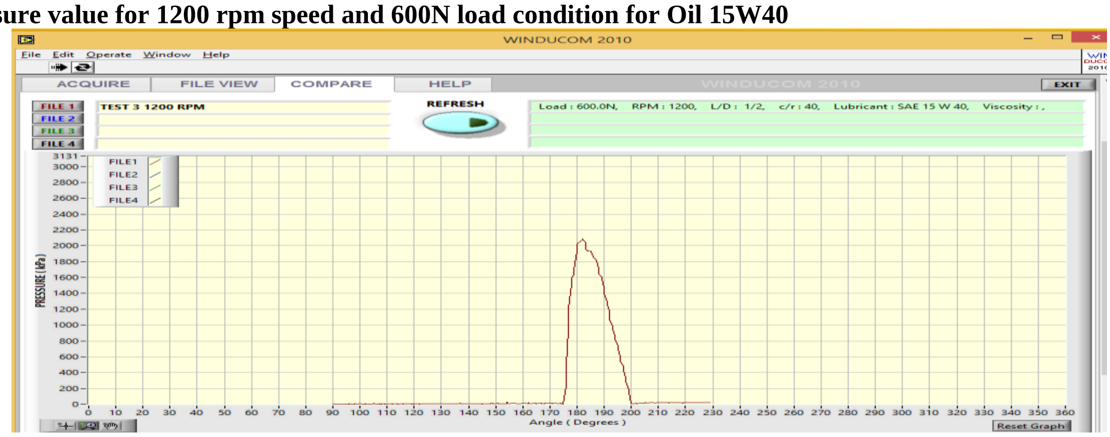

Fig.6 Variation of pressure with angular position and load condition at 1200 rpm and 600N Load Grade Oil SAE 15W40

International Journal of Advanced Research in Science Management and Technology

ISSN(Online)-2454-4159 *Volume 4, Issue 2, February 2018*

Pressure value for 800 rpm speed and 300N load condition for Oil 20W40

Fig.7 Variation of pressure with angular position and load condition at 800 rpm and 300N Load Grade Oil SAE 20W40

Pressure value for 1000 rpm speed and 450N load condition for Oil 20W40

Fig.8 Variation of pressure with angular position and load condition at 1200 rpm and 600N Load Grade Oil SAE 20W40

Pressure value for 1200 rpm speed and 600N load condition for Oil 20W40

Fig.9 Variation of pressure with angular position and load condition at 1200 rpm and 600N Load Grade Oil SAE 20W40

IV. FAST FOURIER TRANSFORM (FFT)

The Fast Fourier Transform (FFT) is an algorithm for transformingdata from the time domain to the frequency domain. Since this is exactly what we want a spectrum analyzer to do, it would seem easy to implement a Dynamic Signal Analyzer based on the FFT. However, we will see that there are many factors which complicate this seemingly straight forward task first, because of the many calculations involved in transforming domains, the transform must be implemented on a digital computer if the results are to be sufficiently accurate. Fortunately, with the advent of microprocessors, it is easy and inexpensive to incorporate all the needed computing power in a small instrument package. Note, however, that we cannot now transform to the frequency domain in a continuous manner, but instead must sample and digitize the time domain input. This means that our algorithm transforms digitized samples from the time domain to samples in the frequency domain.

International Journal of Advanced Research in Science Management and Technology
IJARSMT
ISSN(Online)-2454-4159 *Volume 4, Issue 2, February 2018*

Graph of velocity for 800 RPM speed and 300 N load condition for Oil 10W30

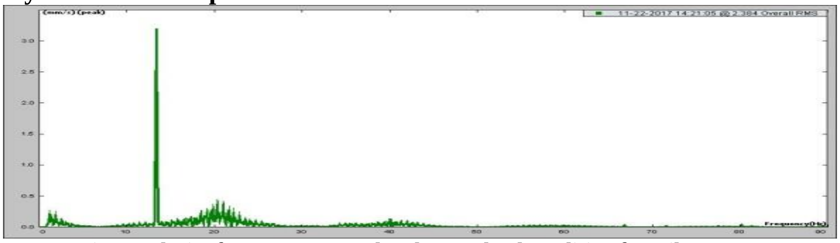

Fig.10 **velocity for 800 rpm speed and 300 N load condition for Oil 10W30**

Graph of Displacement for 800 RPM speed and 300 N load condition for Oil 10W30

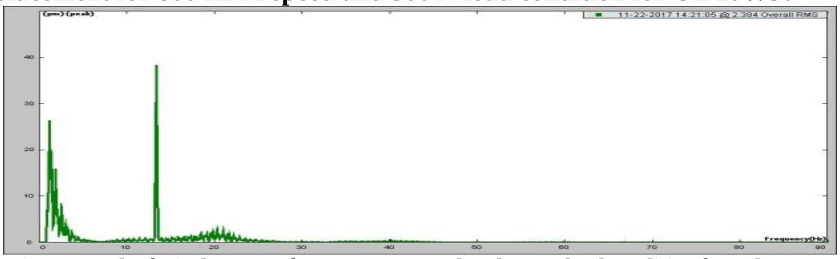

Fig.11 Graph of Displacement for 800 RPM speed and 300 N load condition for Oil 10W30

Graph of velocity for 1000 RPM speed and 450 N load condition for Oil 10W30

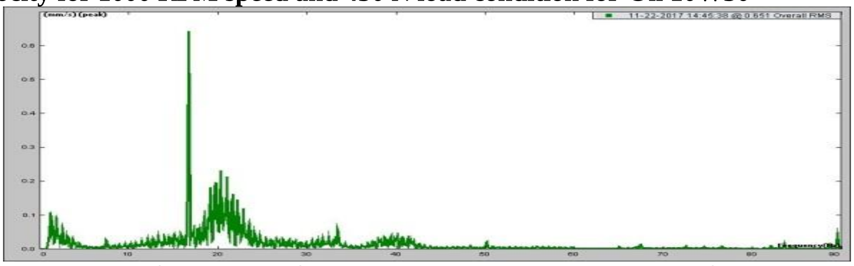

Fig.11 Graph of Displacement for 800 RPM speed and 300 N load condition for Oil 10W30

Graph of Displacement for 1000 RPM speed and 450 N load condition for Oil 10W30

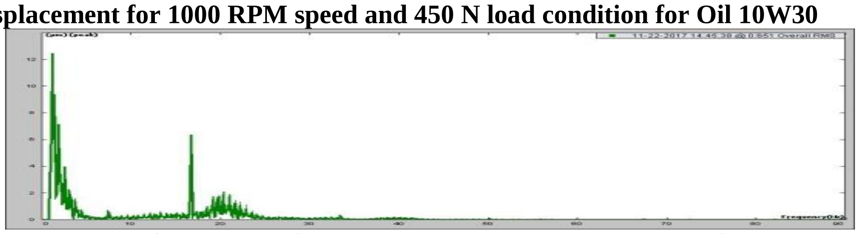

Fig.12 Graph of Displacement for 1000 RPM speed and 450 N load condition for Oil 10W30

Graph of velocity for 1200 RPM speed and 600 N load condition for Oil 10W30

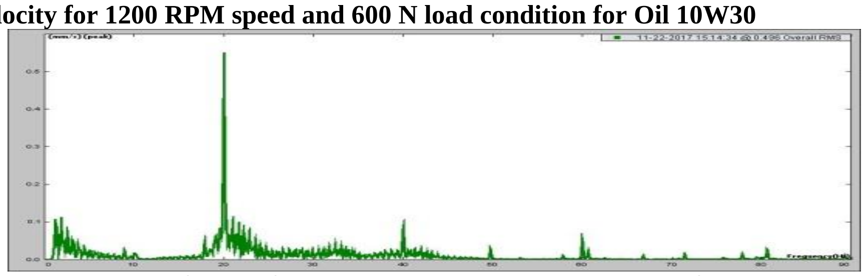

Fig.13 Graph of velocity for 1200 RPM speed and 600 N load condition for Oil 10W30

Graph of displacement for 1200 RPM speed and 600 N load conditions for Oil 10W30

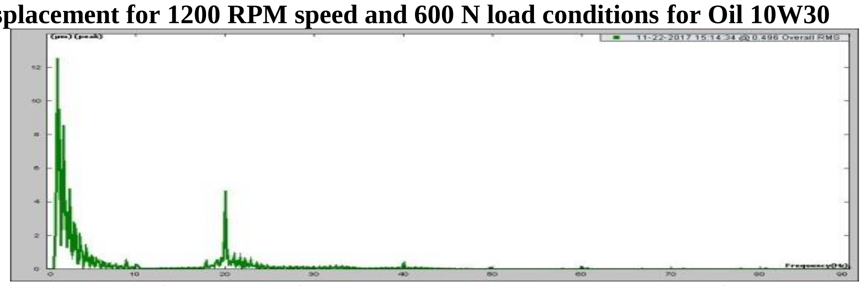

Fig.14 Graph of displacement for 1200 RPM speed and 600 N load condition for Oil 10W30

International Journal of Advanced Research in Science Management and Technology

ISSN(Online)-2454-4159 *Volume 4, Issue 2, February 2018*

Graph of velocity for 800 RPM speed and 300 N load condition for Oil 15W40

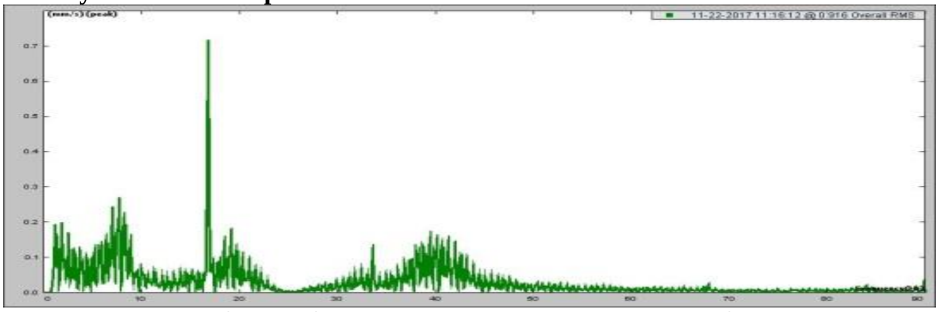

Fig.15 Graph of velocity for 800 RPM speed and 300 N load condition for Oil 15W40

Graph of displacement for 800 RPM speed and 300 N load condition for Oil 15W40

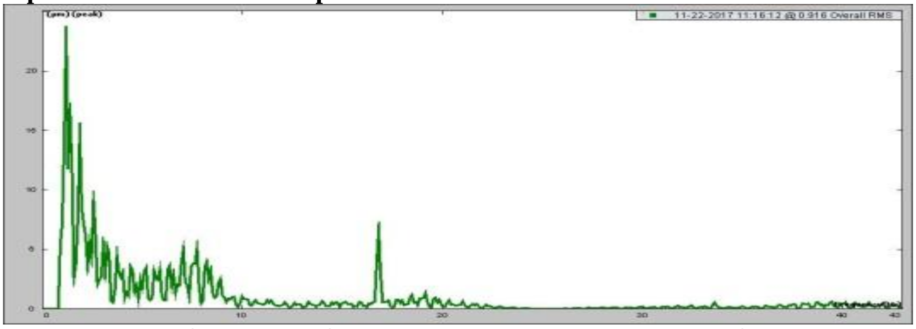

Fig.16 Graph of displacement for 800 RPM speed and 300 N load condition for Oil 15W40

Graph of velocity for 1000 RPM speed and 450 N load conditions for Oil 15W40

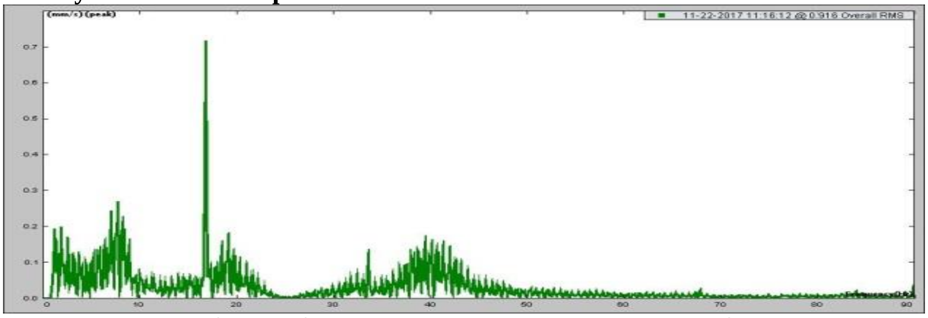

Fig.17 Graph of velocity for 1000 RPM speed and 450 N load condition for Oil 15W40

Graph of displacement for 1000 RPM speed and 450 N load condition for Oil 15W40

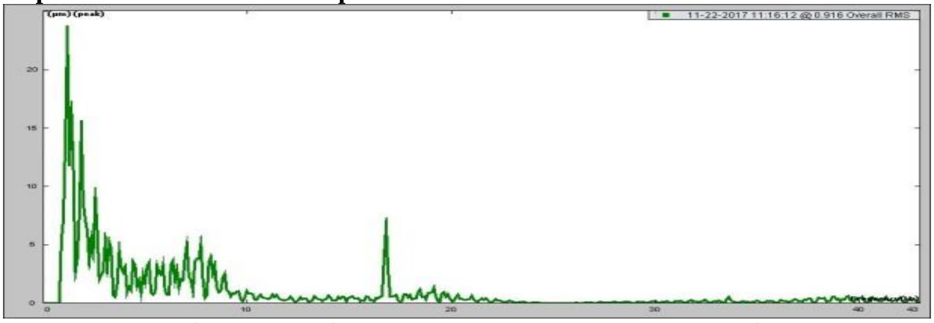

Fig.18 Graph of displacement for 1000 RPM speed and 450 N load condition for Oil 15W40

Graph of velocity for 1200 RPM speed and 600 N load condition for Oil 15W40

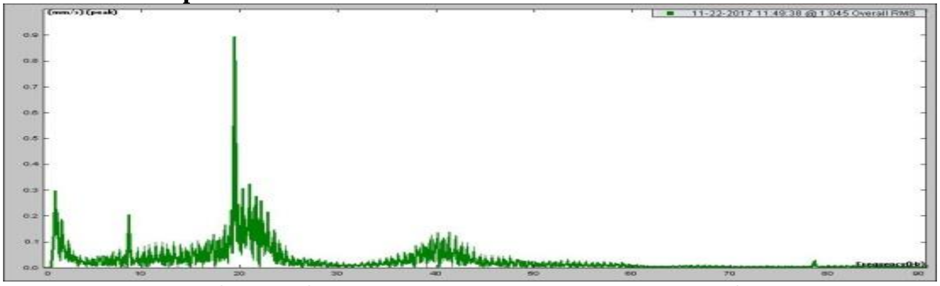

Fig.19 Graph of velocity for 1200 RPM speed and 600 N load condition for Oil 15W40

International Journal of Advanced Research in Science Management and Technology
ISSN(Online)-2454-4159 Volume 4, Issue 2, February 2018

Graph of displacement for 1200 RPM speed and 600 N load condition for Oil 15W40

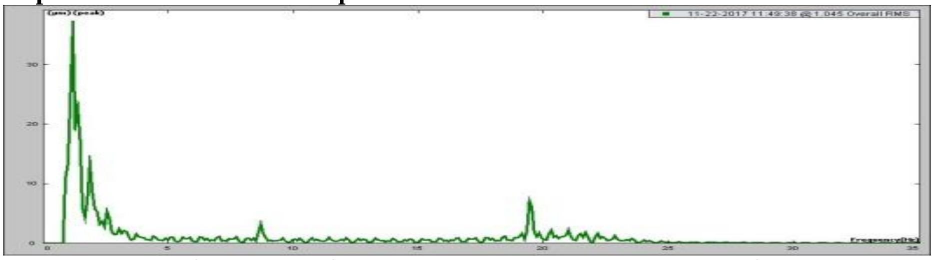

Fig.20 Graph of displacement for 1200 RPM speed and 600 N load condition for Oil 15W40

Graph of velocity for 800 RPM speed and 300 N load condition for Oil 20W40

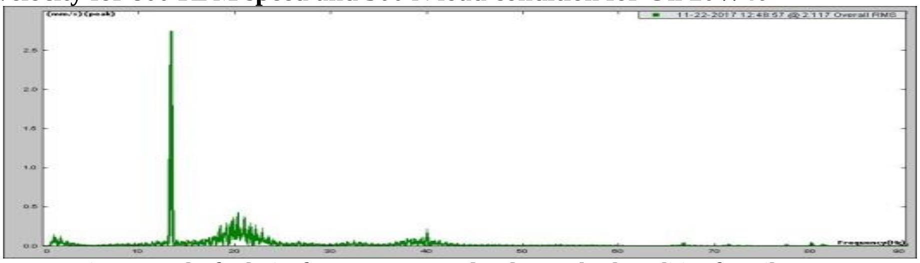

Fig.21 Graph of velocity for 800 RPM speed and 300 N load condition for Oil 20W40

Graph of displacement for 800 RPM speed and 300 N load condition for Oil 20W40

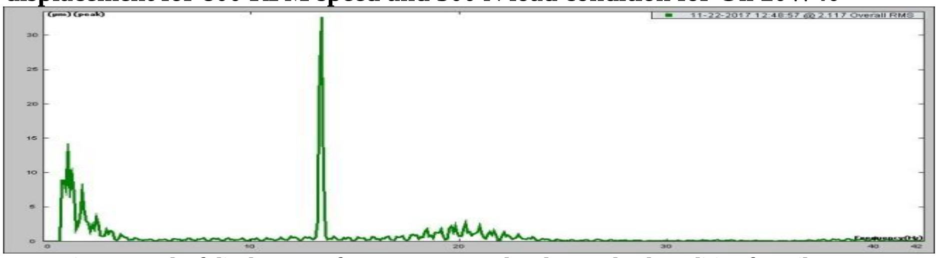

Fig.22 Graph of displacement for 800 RPM speed and 300 N load condition for Oil 20W40

Graph of velocity for 1000 RPM speed and 450 N load condition for Oil 20W40

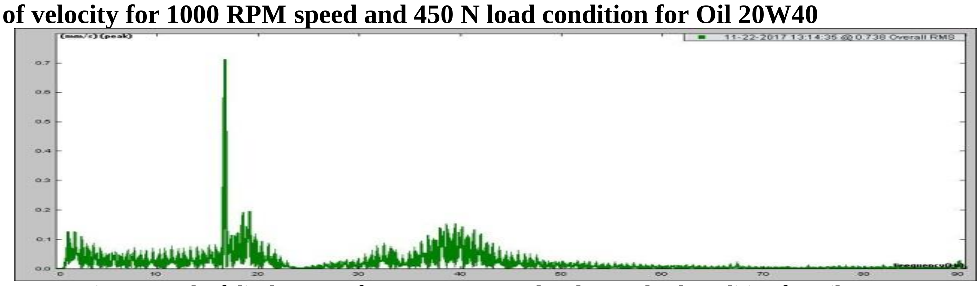

Fig.23 Graph of displacement for 1000 RPM speed and 450 N load condition for Oil 20W40

Graph of displacement for 1000 RPM speed and 450 N load condition for Oil 20W40

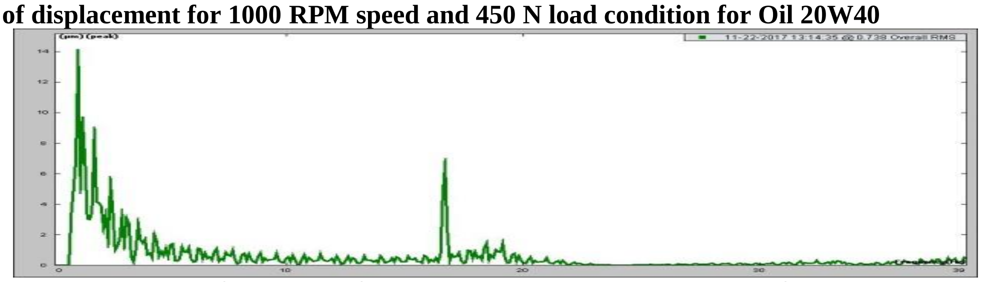

Fig.24 Graph of displacement for 1000 RPM speed and 450 N load condition for Oil 20W40

Graph of velocity for 1200 RPM speed and 600 N load condition for Oil 20W40

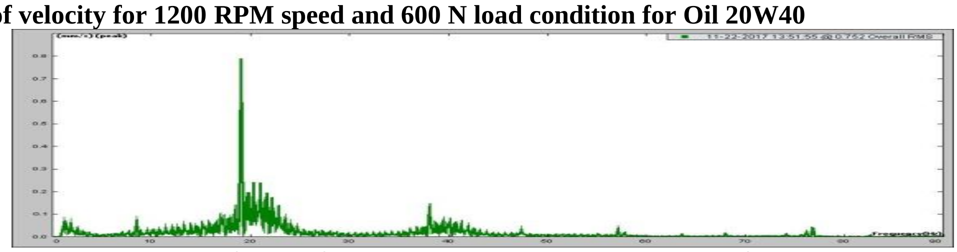

Fig.25 Graph of velocity for 1200 RPM speed and 600 N load condition for Oil 20W40

International Journal of Advanced Research in Science Management and Technology

ISSN(Online)-2454-4159 Volume 4, Issue 2, February 2018

Graph of displacement for 1200 RPM speed and 600 N load condition for Oil 20W40

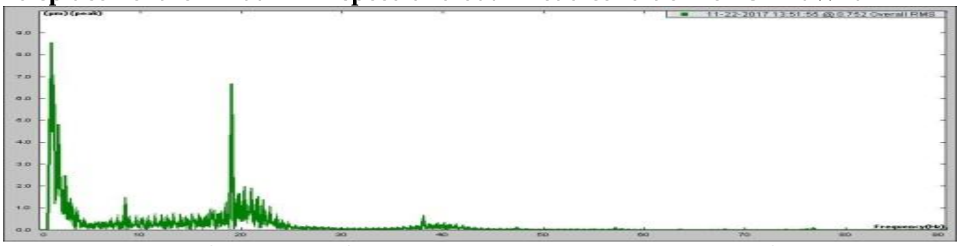

Fig. 26 Graph of displacement for 1200 RPM speed and 600 N load condition for Oil 20W40

V. RESULTS AND DISCUSSIONS

Comparison of Journal Bearing for different load and speed:

Table 6.1 gives the value of nature of pressure distribution of journal bearing

OIL GRADE	SPEED(RPM)	LOAD (N)	MAXIMUM PRESSURE(kPa)
SAE 10W30	800	300	370
	1000	450	1100
	1200	600	1800
SAE 15W40	800	300	450
	1000	450	1200
	1200	600	2100
SAE 20W40	800	300	525
	1000	450	1350
	1200	600	2200

- It shows the value of circular journal bearing oil SAE 10W30 for 800 speed and load 300 N the pressure obtained is 370 kPa, for 1000 speed and load 450 N the pressure obtained is 1100 kPa, 1200 speed and at load 600 N the pressure obtained is 1800 kPa.
- SAE 15W40 is for 800 speed and at load 300 N the pressure obtained is 450 kPa , 1000 speed and at load 450 N the pressure obtained is 1200 kPa, 1200 speed and at load 600 N the pressure obtained is 2100 kPa.
- SAE 20W40 for 800 speed and load 300 N the pressure obtained is 525 kPa , 1000 speed and at load 450 N the pressure obtained is 1350 kPa, 1200 speed and at load 600 N the pressure obtained is 2040 kPa as shown in table 7.1
- It shows the maximum value of journal bearing oil SAE 20W40 is 2200 kPa for higher speed(i.e 1200rpm) and at higher load(600N) and minimum value of pressure for oil SAE 10W30is 370at lower speed and load(i.e 800 rpm and 300N)

VI. CONCLUSION

This study concludes that pressure distribution of a hydrodynamic journal bearing predicted experimentally. It was observed that pressure variation in SAE20W40 oil is maximum as compared to SAE 10W30 and SAE15W40.

The fluid film pressure developed in hydrodynamic journal bearing increases from 00 up to certain angle, then again decreases as an angle reaches to 1800.

The following conclusions are drawn from the experimental study carried out:

1. Variation of pressure distribution with respect to angle of rotation (degrees) for SAE 10W30, 15W40 and SAE20w40 oils revealed a pressure increases from 0 to 800 of rotation and then decreases for the remaining rotation of bearing. This clearly indicates the presence of a hydrodynamic lubrication which is the need of high speed journal bearing.

2. The maximum value of pressure was found to be 2200 KPafor the oils 20 W40. This variation is due to the rotational speed and the viscous nature of an oil and viscous nature of the oil, change of eccentricity etc. but shown good performance under various loading and speed.

ISSN(Online)-2454-4159 *Volume 4, Issue 2, February 2018*

3. From pressure distribution from JBTR and displacement from FFT it conclude that 20W40 is the best oil for bearing.

REFERENCES

[1] Pinkus, O. and Lynn, M. Analysis of Elliptical Bearings. Trans. ASME 1956; 55-LUB-22: pp965-973.

[2] Fitzgerald, M. K. and Neal, P. B. Temperature Distributions and Heat Transfer in Journal Bearings. Trans. ASME, J. Tribol 1992; 114: pp122-130.

[3] Hussain, A., Mistry, K., Biswas, S. and Athre, K. Thermal Analysis of Non- Circular Bearing. Trans. ASME, J. Tribol 1996; 118:pp 246-254.

[4] Mishra, P. C., Pandey, R. K. and Athre, K. Temperature Profile of an Elliptic Bore Journal Bearing. Tribology International 2007; 40(3): pp453-458.

[5] Ostayen, R. A. J. V. and Beek, A. V. Thermal Modelling of the Lemon-Bore Hydrodynamic Bearing. Tribology International 2009; 42: pp23-32.

[6] Chauhan, A., Sehgal, R. and Sharma, R. K. Thermohydrodynamic Analysis of Elliptical Journal Bearing with Different Grade Oils. Tribology International 2010; 43: 1970-1977.

[7] AmitChauhan, AmitSingla, NarenderPanwar and Prashant Jindal, "CFD Based Thermo-Hydrodynamic Analysis of Circular Journal Bearing," International Journal of Advanced Mechanical Engineering, ISSN 2250-3234 Volume 4, Number 5 (2014), Pp. 475-482

[8] S. Baskar,G. Sriram, "Experimental Analysis Of Hydrodynamic Journal Bearing Under Different Biolubricants," Ieee-International Conference On Advances In Engineering, Science And Management (ICAESM -2012) March 30, 31, 2012

[9] NabarunBiswas,PrasunChakraborti, "Transient Analysis Of 3 Lobe Bearing Considering Surface Roughness Effect For Gas Turbine" 6th BSME International Conference On Thermal Engineering (ICTE 2014)

[10] Peeyush Vats, B.C. Sharma, Sumit Sharma, "Heat Transfer through Journal Bearing: A Case Study," IJRET: International Journal of Research in Engineering and Technology Eissn: 2319-1163 | Pissn: 2321-7308

[11] K. M. Panday, P. L. Choudhury, And N. P. Kumar, "Numerical Unsteady Analysis of Thin Film Lubricated Journal Bearing", IACSIT International Journal of Engineering and Technology, Vol. 4, No. 2, April 2012

[12] ArjunPanthi, Prof. Jai Balwanshi, Prof. Ajay Chandravanshi, Prof. Gourav Gupta, " Design And Analysis Of Hydrodynamic Journal Bearing Using Raimondi And Boyd Chart" , International Journal Of Core Engineering & Management (IJCEM) Volume 2, ISSN: 2348 9510 Issue 3, June 2015

[13] Chaitanya K Desai AndDilip C Patel, "Experimental Analysis Of Pressure Distribution Of Hydrodynamic Journal Bearing: A Parametric Study", Proceedings Of The International Conference On Mechanical Engineering 2005 (ICME2005) 28- 30 December 2005, Dhaka, Bangladesh

[14] Hussain, A., Mistry, K., Biswas, S. and Athre, K. Thermal Analysis of Non- Circular Bearing. Trans. ASME, J. Tribol 1996; 118: 246-254.

[15] SM Chun, "Thermo hydrodynamic lubrication analysis of high-speed journal bearing considering variable density and variable specific heat," Tribology International, 37, pp. 405-413, 2004

[16] J Sep, "Three dimensional hydrodynamic analysis of a journal bearing with a two component surface layer, "Tribology International, 38, pp. 97-104, 2005

[17] Baskar S and Sriram G Experimental analysis of hydrodynamic journal bearing under different bio lubricants. 2012, pp.132–135.

[18] Mane RM, Soni S. Analysis of Hydrodynamic Plain Journal Bearing, Proc. 2013 COMSOL Conference, Bangalore, 2013.

[19] Baskar. S., Sriram.G. "Pressure Distribution Analysis Of A Hydrodynamic Journal Bearing Using Vegetable Oil Based Nano Lubricant", International conference on advances in tribology, 2014:14-15

[20] Y.Y. Wu & W.C. Tsui "Experimental analysis of tribological properties of lubricating Oils withnanoparticles additives" Wear 262 (2007) 819–825, 10 October 2006.