Pressure Distribution Analysis of Plain Journal Bearing with Lobe Journal Bearing

Mahesh Aher[1], Sanjay Belkar[2], R. R. Kharde[3]

[1,] P.G Student, Mechanical Engineering Department, University of Pune
[2,] Associate .Professor, Mechanical Engineering Department, University of Pune
[3]Head of Department, Mechanical Engineering Department, University of Pune
Pravara Rural College of Engineering, Pravaranagar.

***Abstract** - The cylindrical (plain) hydrodynamic journal bearing is the most basic hydrodynamic bearing with cylindrical bore. This bearing has a high load capacity, and the simple design is compact, bi-rotational, and easy to manufacture However, as the design speeds of machines increased, it is found that this bearing had limitations due to oil whirl. Oil whirl is very undesirable because of high vibration amplitudes, forces, and cyclic stresses that are imposed on the shaft, bearings and machine creating the instability of bearing. This resulted in a variety of fixed geometry bearings which are modifications to the profile of the bearing. It is found that the stability of bearings can be increased by the use of lobe bearing. Hence, an attempt is made to find the load carrying capacity of lobe bearing.*

A Comparative study and experimental analysis is carried out for getting the pressure distribution and load carrying capacity of plain and lobe bearing at different speed and load. The result obtained reveals the higher load carrying capacity of lobe bearing with good stability.

Keywords - Journal bearing, pressure distribution, stability, synchronous whirl.

Nomenclature :

c	Radial clearance, mm
C	Diametrical clearance, mm
e	Eccentricity, mm
g	Acceleration due to gravity, m/sq.sec
h	Film thickness, mm
ε	Eccentricity ratio
h_m	Minimum oil film thickness, mm
D	Diameter of bearing, mm
d_j	Diameter of journal, mm
lb	Length of journal bearing, mm
N_j	Rotational speed of journal, rpm
ωj	Angular velocity , rad/sec
P	Bearing pressure, kPa
p	Pressure on fluid film, kPa
W	Load, N
μ	Dynamic viscosity, N-s/sq. m
S	Sommerfeld Number

1 .INTRODUCTION

Hydrodynamic journal bearing are considered to be a vital component of all the rotating machinery. It is used to support radial loads under high speed operating conditions. In a hydrodynamic journal bearing pressure of hydrodynamic lift is generated in thin lubricating oil film that separates the shaft and the bearing thus preventing metal to metal contact.

The recent demands for higher performance of automotive engines and turbo machinery require the crankshaft and connecting rod bearing to operate under more severe condition like higher speeds, higher loads and higher temperature. To meet these requirements plain bearing with symmetrical lobe has been developed. Various tests has been conducted to improve performance of plain bearing under these condition in both materials and design and possibility has been formed in modifying the geometry of bearing. Although it is generally believed that surface unevenness impairs hydrodynamic performance.

Theoretical analysis has been revealed a multilobe bearing has found to be more stable than circular bearings. A three lobe bearing possesses good stability characteristics as turbomachinery works on higher speed and load, hence would act has better replacement for plain journal bearing.

2. LITERATURE SURVEY

The current trend in industry is to run turbomachines at high speeds in order to make them compact and reduce mass. It is found that the performance of ordinary circular bearings is not very satisfactory. To improve the stability of these bearings, pressure dams are incorporated in these bearings. The analytical dynamic analysis has shown that the cylindrical pressure-dam bearings are found to be very stable. Also an experimental stability analysis of such types of bearings showed that the analytical stability analysis provides the general trends in the experimental data. The study of non-cylindrical pressure-dam bearings such as finite elliptical, half elliptical, offset-halves, and three-lobe pressure-dam bearings have proved that the performance of the bearings is improved.

Fredrick T. Schuller[1] intended principally as a guide in the selection and design of antiwhirl bearings that must operate at high speeds and low loads in low-viscosity fluids such as water or liquid metals. However, the various fixed-geometry configurations can be employed as well in applications where other lubricants, such as oil, are used and fractional-frequency whirl is a problem. The important parameters that affect stability were discussed for each bearing type, and design curves to facilitate the design of optimum-geometry bearings are included.

J. Lund[2], A comparison of the stability of the different bearing configurations tested was obtained. This volume treats three special bearing types selected for study because of their favorable stability characteristics and, hence, their potential for use in high speed rotating machinery applications. The three bearing types are, the Three Lobe Journal Bearing, the Floating Sleeve bearing with an Incompressible Lubricant, the Floating Sleeve Bearing with a Compressible Lubricant. The volume gives extensive design data in form of charts and tables from which the bearing dimensions can be obtained for a given application. Data are given for bearing flow, friction power loss and the speed at which hydrodynamic instability sets in. In addition, two computer programs accompany the volume, and instructions and listings of the programs are included. The programs may be used to obtain data for cases not covered by the presented design data.

G. Bhushan[3], Multilobe bearings are found to be more stable than circular bearings. Rakesh sehgl[4], an experimental setup/rig was developed to investigate the behavior of non-circular bearings. G. Bhushan[5], deal with a theoretical investigation of stability of four lobe bearing. Stanislaw strzeleckI[6] considered the rotors of turbogenerators operate in 2-lobe journal bearings. These bearings can be designed with the same or different profile of upper and bottom lobe, e.g. the upper lobe has cylindrical and bottom one the offset profile. Chaitanya K Desai and Dilip C Patel[7] worked on the method of to analyze the pressure distribution in hydrodynamic journal bearing for various loading conditions and various operating parameters.

3 .Hydrodynamic Journal Bearing

Hydrodynamic lubrication means that the load-carrying surfaces of the bearing are separated by a relatively thick film of lubricant, so as to prevent that the stability thus obtained can be explained by the laws of fluid mechanics. Hydrodynamic lubrication does not depend upon the introduction of the lubricant under pressure, though that may occur; but it does require the existence of an adequate supply at all times. The film pressure is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a velocity sufficiently high to create the pressure necessary to separate the surfaces against the load on the bearing. Hydrodynamic lubrication is also called full-film, or fluid lubrication.

Let us now examine the formation of a lubricant film in a journal bearing. Fig.1 shows a journal which is just beginning to rotate in a clockwise direction. Under starting conditions, the bearing will be dry, or at least partly dry, and hence the journal will climb or roll up the right side of the bearing as shown in Fig.1. Under the conditions of a dry bearing, equilibrium will be obtained when the friction force is balanced by the tangential component of the bearing load.

Now suppose a lubricant is introduced into the top of the bearing as shown in Fig.1. The action of the rotating journal is to pump the lubricant around the bearing in a clockwise direction. The lubricant is pumped into a wedge-shaped space and forces the journal 1 over to the other side. A minimum film thickness ho occurs, not at the bottom of the journal, but displaced clockwise from the bottom as in Fig1. This is explained by the fact that a film pressures in the converging half of the film reaches a maximum somewhere to the left of the bearing center.

Figure 1. shows how to decide whether the journal, under hydrodynamic lubrication, is eccentrically located on the right or on the left side of the bearing. Visualize the journal beginning to rotate. Find the side of the bearing upon which the journal tends to roll. Then, if the lubrication is hydrodynamic, mentally place the journal on the opposite side.

The nomenclature of a journal bearing is shown in Fig.1. The dimension c is the radial clearance and is the difference in the radii of the bushing and journal. In Fig 1 the center of the journal is at 0 and the center of the bearing at *0'*. The distance between these centers is the eccentricity and is denoted by e. The minimum film thickness is designated by ho, and it occurs at the line of centers. The film thickness at any other point is designated by *h*. We also define an eccentricity ratio ϵ as

$$\varepsilon = e / c$$

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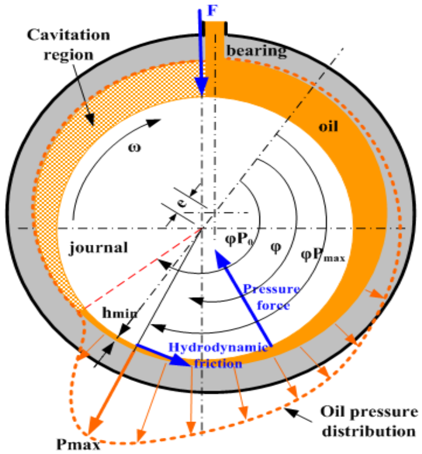

Fig 1.Plain Journal bearing

The bearing shown in the Fig.1 is known as a partial bearing. If the radius of the bushing is the same as the radius of the journal, it is known as a. fitted bearing. If the bushing encloses the journal, as indicated by the dashed lines, it becomes a. full bearing. The angle β describes the angular length of partial bearing. For example, a 120^0 partial bearing has the angle β equal to 120^0

3.1 Theoretical Equation to find pressure distribution

1) Bearing Pressure $P = W/2R_jL$
2) Summerfeld Number $S = (R_j/C)^2\ (\mu N_j/P)$
3) Eccentricity ratio ε obtained by Raimondi &Boyd chart.
4) Pressure Distribution

$P = 6{*}W_j\mu{*}(r/C)^2\varepsilon\sin\theta\ (2+\varepsilon\cos\theta) / [(2+\varepsilon^2)(1+\varepsilon\cos\theta)^2]$

3.2 Neccessity Of Lobe

In the design of journal bearings there are more requirements toward higher speeds of operation and specific loads which allow for the design of bearings characterized by higher efficiency and reliability. Among different design means which are applied for receiving the optimum operation of bearing, there is the possibility to vary the bore profile of bearing.

By use of lobe can improve properties including increase in maximum load capacity, stiffness and damping and decrease in power losses. Also it give provision of good stability at higher speed application.

3.3 Advantages of Multilobe bearing-

- It gives good stability at higher speed.
- Dampened, low-oscillation, noise and wear-free shaft operation. If the oil supply is operating properly, virtually unlimited bearing service life
- Several supportive lubrication films distributed around the shaft circumference guarantee that the shaft is generally centered, thus significantly improving concentricity
- Permits high continuous loading
- Shock loads of several times the level of the continuous load are acceptable
- Low friction losses
- Good lubricant flushing and cooling effects

3.4 Geometery of Three Lobe Bearing

The geometry of the 3 lobe bearing is shown schematically in fig 2[8]. The bearing is composed of 3 circular arcs whose centers of curvature are, removed from the center of the bearing by the distance r. Thus, even when the journal is centered in the bearing, the pads are loaded. In this way, the stability threshold of the bearing is raised and even a vertical rotor, or a rotor operating in a "zero-field," can run stably which is not possible with a conventional full circular bearing. However, the improved stability threshold is paid for by an increase in friction power loss and a smaller operating minimum film thickness which makes the bearing more sensitive to impurities in the lubricant.

The journal diameter is D, its radius is R and the bearing length is L. The design parameters are the length-to-diameter ratio, the Sommerfeld number S and the film thickness.

Sommerfeld Number: $S = \frac{\mu NDL}{W}(R/C)^2$

The eccentricity ratio ε and the attitude angle θ defines the steady-state position of the journal center relative to the bearing center and the static load line (the x-axis). From this the minimum film thickness in the bearing can be determined. Under almost all conditions, the minimum film thickness occurs at the bottom lobe in which case:

Minimum thickness $h = \left[1 - \sqrt{\varepsilon^2 + \delta^2 + 2\delta\varepsilon\cos\theta}\right]$

The minimum film thickness gives a relative measure of how heavily, the bearing is loaded. The acceptable lower value for the minimum film thickness depends on the condition of the lubricant.

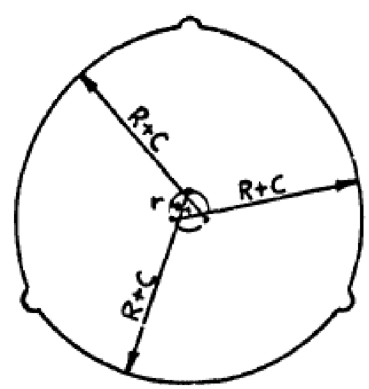

Fig 2. Lobe Bearing

4.1 Experimental Setup

This section includes schematic diagram of the Journal Bearing Test rig setup on which the experimentation has been conducted.

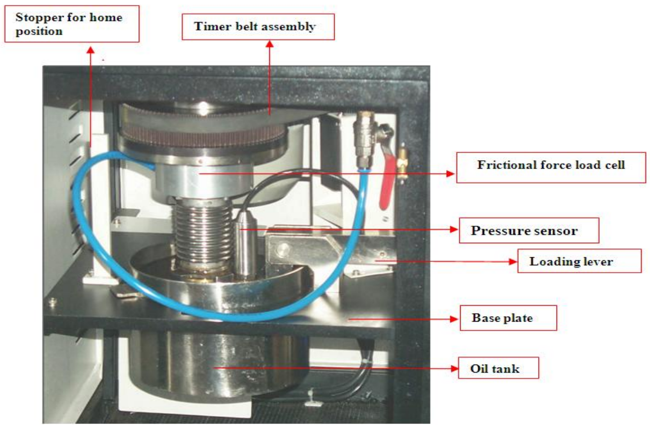

Fig 3.Journal Bearing Test Rig.

This is a sturdy versatile machine, which facilitates study of pressure at corresponding angular position of the pressure sensor with the load line. The JBTR equipment consists of a vertically mounted shaft and driven by a variable speed motor. A metallic bellow connects brass bearing at bottom and top is fixed to frictional torque load cell. Bearing made of brass material encloses the shaft at the lower end and is immersed in an oil sump. An rpm sensor disc is mounted on the driven pulley to measure the revolution of the shaft per minute. A stepper motor moves the bearing in the direction of the rotation of the shaft unto 180^o in steps of 9^o. A pressure sensor is fixed on the bearing, which measures the film pressure distributed in the oil film. Radial load is applied by dead weights through a lever mechanism. 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. Normal load, rotational speed can be varied to suit the test conditions. Frictional torque value can also be displayed on the controller. Data obtained are transmitted to PC through data acquisition cable.

The JBTR Equipment consists of Journal assembly, Bearing assembly, Loading assembly, Lubrication system, Sensors, Controller, Data Acquisition software and cables.

4.2 Experimental Procedure

Journal is housed in spindle supported on bearing. Spindle housing is fixed firmly to base plate. Journal is rotated by a AC motor through pulley arrangements, on top of the spindle an rpm sensor disc is mounted to measure speed.

Diameter and length of the brass bearing is made equal to get ratio l/d = 1. A narrow orifice inside bearing lets oil into pressure sensor mounted area. Bearing is mounted on bottom of a metallic bellow and on top of bellow a frictional torque load cell is mounted. Load cell unit is mounted on to indexing pulley. Indexing pulley is driven by stepper motor. Angular range for stepper motor is set upto 180^o, in 9^o steps.

A lever arrangement is provided with a lever ratio of 1:5. To one end of the lever a loading pan is attached on which the weights can be placed in the range of 150N to 750N. To the other end is mounted a ball bearing through which the load is applied to the brass bearing. Lever is pivoted to get 1:5 loading ratio. An oil filling arrangement is provided on the top of base plate to gravity feed the brass bearing by means of a flexible tube before start of test to remove any trapped air. An oil sump is provided to ensure continuous lubrication for the journal bearing. The oil sump is placed on bottom base plat, which is fixed on to the supporting structure.

A calibrated pressure sensor to measure unto 1000psi is fixed to the bearing to measure pressure at various points around the journal (Fig no.3). Output is 100mv for 1000 psi. Test parameter such as disc speed can be set with the front panel settings on the controller. The load is applied by dead weights through loading lever assembly. The pressure and Torque readings are displayed along with indexing angle during test. Pressure and Angle readings are processed and serially transmitted through data acquisition cable.

4.3 Operating Parameter:

Material of Journal	: Chromium Nickel Steel
Material of Bearing	: Brass material
Diameter of journal (D_j)	: 39.96 mm
Length of Bearing (L)	: 40.00 mm
Clearance (C)	: 0.185 mm
C/R_j	: 0.005
Speed range	: 1000-2000 rpm
Load (W)	: 300 N
Lubricant	: SAE20W40
μ	: 0.0981pa_sec

4. RESULTS AND DISCUSSION

Comparison of Plain Journal Bearing and Lobe Journal Bearing for different load and speed :

Table gives the value of nature of pressure distribution of plain journal bearing and lobe journal bearing.

Load (N)	Maximum Pressure(kpa) for 1000rpm		Maximum Pressure (kpa) for 1500rpm		Maximum Pressure (kpa) for 2000rpm	
	Plain bearing	Lobe bearing	Plain bearing	Lobe bearing	Plain bearing	Lobe bearing
300	424.42	824.17	482.58	826.59	482.58	868.64
450	785.91	985.84	861.75	937.58	903.11	1006.50
600	861.75	986.84	951.37	1096.1	965.16	1049.7
750	910.0	1054.70	1034.1	1137.5	1151.2	1192.60

- It shows the maximum value of plain journal bearing is 1151.20 for higher speed(i.e 2000rpm) and at higher load(750N) and minimum value of pressure is 427.42 at lower speed and load(i.e 1000rpm and 300N).
- For, lobe the maximum pressure value at higher speed and load is 1192.60 and minimum pressure value is 834.17 which is greater than plain journal bearing at lower speed and load.

- The lobe bearing shows higher maximum pressure at speed of 2000rpm (maximum speed) and load of 750N (maximum load).
- Fig.4 and Fig.5 shows the pressure distribution of plain journal bearing and lobe journal bearing for different load at constant speed (1500rpm). By the observing graph, it can reveal that for both (i.e plain and lobe journal bearing), as speed and load increases the value of pressure also increases.

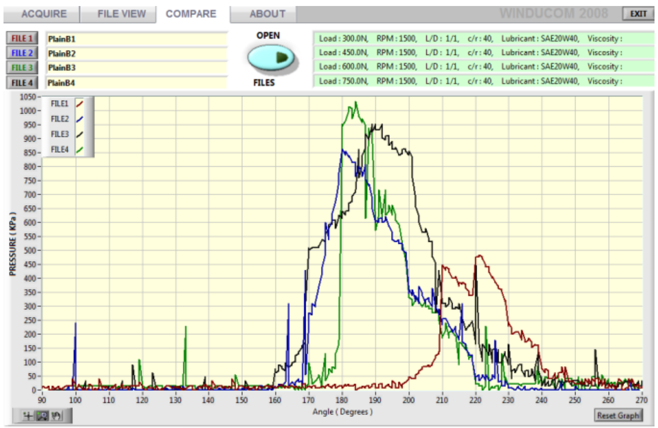

Fig 4.Graph of pressure versus angle showing the comparison of plain bearing for different load and at constant speed.

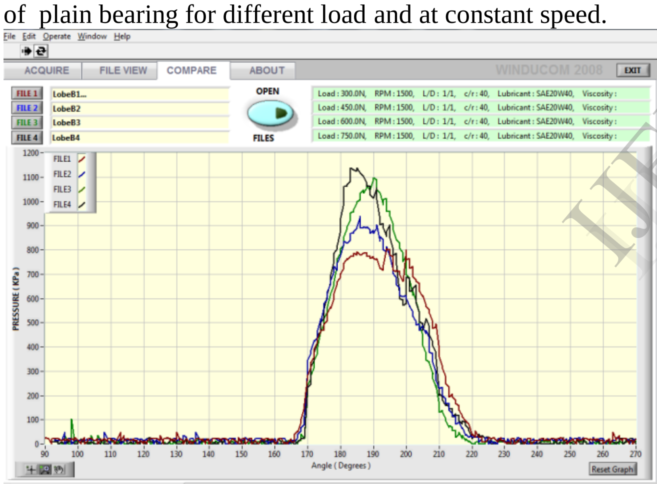

Fig 5.Graph of pressure versus angle showing the comparison of lobe journal bearing for different load and at constant speed.

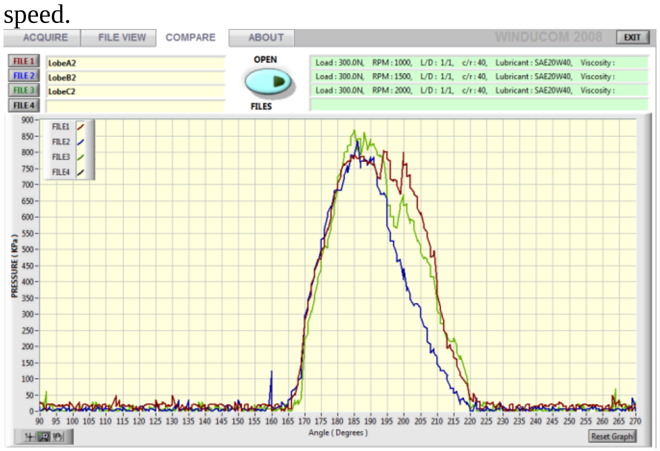

Fig 6

Fig 6.Graph of pressure versus angle showing the comparison of plain bearing for different load and at constant speed.

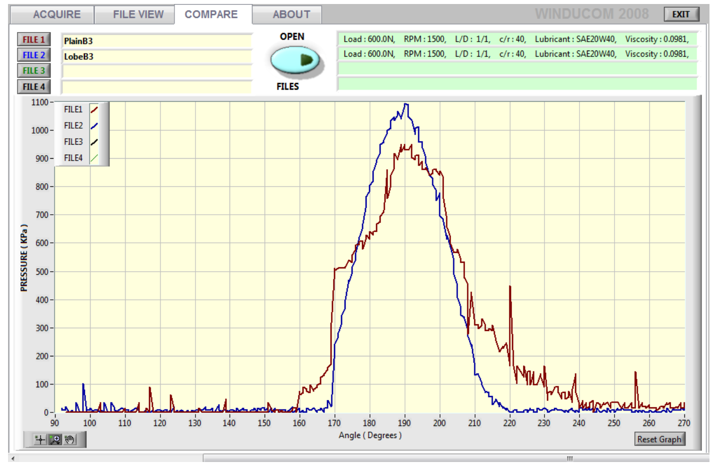

Fig 7.Graph of pressure versus angle showing the comparison of plain bearing and lobe journal bearing for speed of 1500rpm.

5. CONCLUSIONS

The pressure distribution obtained from the experimental work of three lobe bearing has the maximum pressure value over the plain hydrodynamic journal bearing. The value of maximum pressure is increased by 30% for lower load and 7 to 10% for higher load. It can be concluded that the load carrying capacity of lobe manufactured bearing is higher than the plain bearing.

The average maximum pressure duration for the plain hydrodynamic journal is in between 160^{o}-230^{o} and for lobe journal bearing the value is in between 165^{o} to 225^{o}. Hence, there is sudden rise of pressure for short duration. It can be reveal that stability of lobe bearing is better than that of plain journal bearing. This satisfy the theory that lobe hydrodynamic bearing is more stable than plain bearing at higher speed as well as lower speed.

The nature of maximum pressure is steady in lobe hydrodynamic journal bearing. The angle of maximum pressure gets shifted (reduces) as the load increases in both the cases(i.e plain and lobe journal bearing).

References

1. Fredrick T. Schuller, "Experiments on the Stability of water lubricated three lobe hydrodynamic bearing ",NASA TND, volume -6315, Page no.-1-35.
2. J. Lund, "The Stability analysis of three lobe bearing" Technical Report AFAPL-65-45, part VII, feb 1971,page 1-250.
3. G.Bhushan, "Effect of load Orientation on the stability of three lobe bearing supporting rigid and flexible rotors" Wear, volume-81, April 2011, pages 195-199.

4. Rakesh Sehgal, "Experimental Measurement of oil film temperature of Elliptical Journal Bearing profile using different grede oil", Tribology online, April 2010, pages 291-299
5. Fredrick T. Schuller, "Design of various fixed-geometry water lubricated hydrodynamic journal bearing for maximum stability", NASA TND, volume -333, Volume 1, Page no.-1-65.
6. G. Bhushan, S. S. Rattan, and N. P. Mehta," Effect of L/D Ratio on the Performance of a Four-Lobe Pressure Dam Bearing" World Academy of Science, Engineering and Technology 32 2007108-103
7. Chaitanya K Desai & Dilip C. Patel, "Experimental Analysis of Pressure Distribution of Hydrodynamic Journal Bearing: A Parametric Study", (ICME 2005), Dhaka, Bangladesh.
8. Fredrick T. Schuller, "Experiments on the Stability of water lubricated three lobe hydrodynamic bearing ",NASA TND, volume -6315, Page no.-1-35.
9. Stanisław STRZELECKI," Effect of lobe profile on the load capacity of 2-lobe journal Bearing". SCIENCE IN CHINA (Series A), Vol. 44, Aug2001.
10. Giovanni Adilettaa,n, ErasmoMancusi b, SalvatoreStrano a, "Nonlinear behavior analysis of a rotor on two-lobe wave journal bearings" Tribology International 44 (2011) page-42–54.
11. Dr.S.S Banwait, "A comparative performance Analysis of non- circular two-lobe journal bearing" IE(I), march 31.2006, pages 202-210.
12. G. H. Jang,J. W. Yoon,'Stability Analysis of a Hydrodynamic Journal Bearing With Rotating Herringbone' ASM, vol.-125, 2003, pages 291-391, e-mail: ghjang@hanyang.ac.kr
13. Raghunandana. K "Inverse Design Methodology for the Stability Design of Elliptical Bearings Operating with Non-Newtonian Lubricants", WCECS 2007, October 24-26, 2007.
14. N.P.Mehta,S.S.Rattan, Rajiv Verma*,Stability Analysis of Three Lobe Hydrodynamic Journal Bearing on couple stress fluid effects", International Journal of Engineering Science and Technology, Vol. 2(10), 2010,page 5736-5745.
15. S. STRZELECKI*, S.M. GHONEAM, "The effect of clearance variation on the maximum temperature of the oil film of cylindrical 3-lobe journal bearing", Scientific Problems of Machine Operation and Maintanance ,1 (161) 2010
16. N. P. Mehta, S. S. Rattan and Rajiv Verma, "Stability Analysis of two Lobe hydrodynamic journal bearing with couple stress lubricant" ARPN journal, Vol-15, January 2010,69-75.
17. James F. Walker, Florin Dimofte, Harold E. Addy, "Experimental Pressure Measurements and Fractional Frequency Whirl Threshold for Fractional Frequency Whirl Threshold for wave and plain journal bearing" ",NASA TND, volume -2985. Stanisław STRZELECKI Institute of Machine Design, Stefanowskiego 1/15, 90-924 Łódź, Poland Received July 5, 2001
18. J.Ramesh, B.C.Mujumdar, N.S.Rao,"thermodynamic Analysis of Submerged Oil Journal Bearing considering surface roughness effects", Transactions of the ASME, Volume 110, Jan.1997, pp.100-106.
19. A Paydas, and E H Smith, "A flow-continuity approach to the analysis of hydrodynamic journal bearings" Proc Instn Mech Engrs Vol 206, page 57-70.
20. Ravindra R. Navthar & Dr.N.V. Halegowda, "Stability Analysis of Hydrodynamic Journal Bearing using Stiffness Coefficients ", International Journal of Engineering Science & Technology (2010), Volume 2, No.(2), Page no.-87-93