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EXPERIMENTAL ANALYSIS OF PRESSURE DISTRIBUTION OF HYDRODYNAMIC JOURNAL BEARING: A PARAMETRIC STUDY

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ABSTRACT
The main objective of the present study is to analyze the pressure distribution in hydrodynamic journal bearing for various loading conditions and various operating parameters. The space between the shaft and the bearing is called lubrication gap and is filled with lubricant. Journal bearing test rig is used to test the 140 mm diameter and 70 mm long bearing. Test bearing is located between two antifriction bearings. The bearing is loaded mechanically. The bearing is tested under various parameters like type of lubricant, loading conditions, speeds etc… In the last, experimental results are compared with the theoretical results and results are found satisfactorily.

Keywords: Hydrodynamic bearing, Pressure distribution, Loading.

1. INTRODUCTION
The task of the bearing is to transform directional motion into rotational motion. Historically seen, the bearing is a very old machine element that was already used in antiquity. Leonardo da Vinci identified and formulated the first laws of friction. Today bearings are very widespread, particularly due to their use in piston engines. Nonetheless, because of its complex operation especially in the reciprocating engine, the bearing continues to resist exact analytical examination. Gy. Szota B. Kovács and F. J. Szabó [1] have done their work on Optimized Gap Shapes for Sliding Bearings On the basis of previously developed and published theoretical results of the authors an optimization process was elaborated for the calculation of the optimal lubricant film shape between lubricated sliding surface pairs. Dipl.-Ing. Rolf Lasaar, Prof. Dr.-Ing. Monika Ivantysynova [2] has done the work on Gap Geometry Variations in Displacement Machines and their Effect on the Energy Dissipation. The energy dissipation of displacement machines is mainly influenced by the design of individual lubricating gaps between parts, having relative motion to each other. Fanghui Shi and Qian Wang [3] worked on A Method of Influence Functions for Thermal Analyses of Tribological Elements. Calculations for temperature and thermal deformation in finite tribological systems usually require finite-element (FEM) procedures. Ern Baka [4] has worked on Calculation of the Hydrodynamic Load Carrying Capacity of Porous Journal Bearings. The paper is about the calculation of hydrodynamic load carrying capacity of porous journal bearings. Rolf Lasaar [5] worked on The Influence of the Micro and Macro Gap Geometry on the Energy Dissipation in the Lubricating Gaps of Displacement Machines. The lubricating gap represents one of the central design elements in displacement machines. Zhengchun Peng [6] has done his Work on Thermo hydrodynamic Analysis of Compressible Gas Flow in Compliant Foil Bearings. The work deals with the development of mathematical models and numerical schemes for simulating the hydrodynamic pressure and temperature rise of compliant foil bearings lubricated by a thin gas film in between its compliant bearing surface and the rotating shaft.

2. PRINCIPLE OF JOURNAL BEARING
When a journal bearing which has an adequate supply of lubricant is carrying a load it normally runs with the geometric centers of the shaft and housing displaced so that a region of convergent flow is established. In this region large hydrodynamic pressures are set up within the oil film and these pressures when summated over the total bearing surface are found to completely support the load. If bearing conditions change; for instance the load may vary, the displacement or “attitude” of the centers changes so that the new pressure distribution is sufficient to support the new load. Fig. 1 illustrates the basic principles.

Various non-dimensional load parameters are used to assess the performance of a bearing. The parameters are formed from terms such as the oil viscosity and the load and speed of the bearing. For a given bearing it is found that there is a critical value of load parameter at which the convergent film is unable to support the total bearing load and the surfaces touch. Under these conditions the friction torque suddenly starts to rise and boundary lubrication occurs in the region of minimum film thickness. Figs. 2 illustrate these phenomena. Based on his theoretical investigation of cylindrical journal bearings, Professor Osborn Reynolds showed that oil, because of its adhesion to the journal and its resistance to flow (viscosity), is dragged by the rotation of the journal so as to form a wedge-shaped film between the journal and journal bearing (Fig. 3).

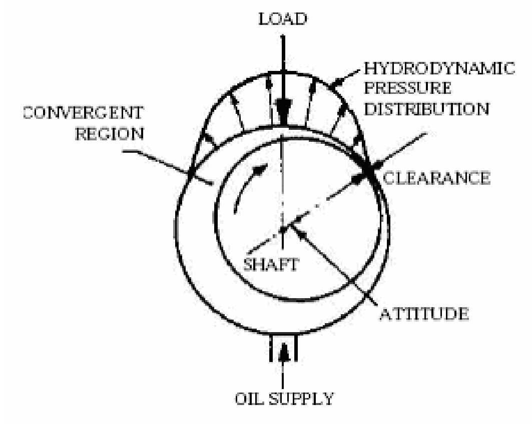


Fig 1. Convergent Region (Pressure Distribution)

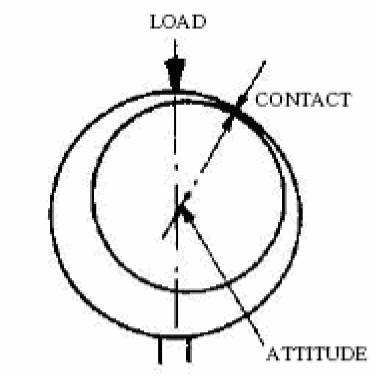


Fig 2. Metal to metal contact

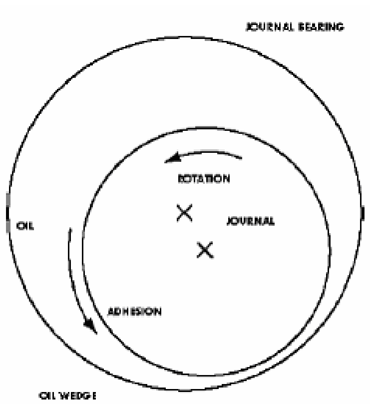


Fig 3. Hydrodynamic Principle

This action sets up the pressure in the oil film which thereby supports the load (Fig.4).This wedge-shaped film was shown by Reynolds to be the absolutely essential feature of effective journal lubrication. Reynolds also showed that “if an extensive flat surface is rubbed over a slightly inclined surface, oil being present, there would be a pressure distribution with a maximum somewhere beyond the center in the direction of motion.”

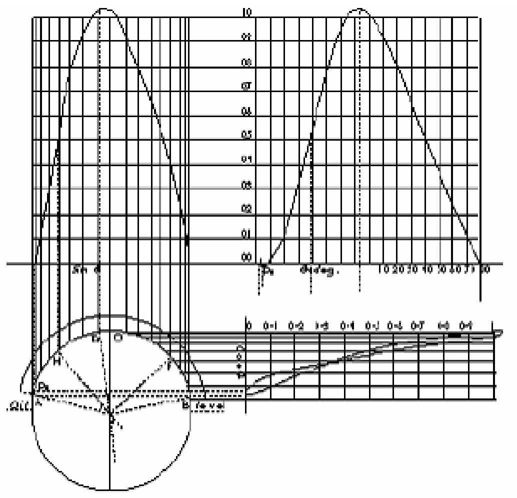


Fig 4. Oil Film Pressure Distribution

3. THEORETICAL ASPECT

When so much heat is generated by hydrodynamic action that the normal lubricant flow is insufficient to carry it away, an additional supply of lubricant must be furnished under pressure. To force a maximum flow through the bearing and thus obtain the greatest cooling effect, a common practice is to use a circumferential groove is to at the center of the bearing, with an oil supply hole located opposite the load-bearing zone. Such a bearing is shown in fig (5).The effect of the groove is to create two half bearings, each having a smaller *l/d* ratio than the original. The groove divides the pressure distribution curve into two lobes and reduced the minimum film thickness, but it has wide acceptance among lubrication engineers and carries load without overheating. To set up a method of solution for oil flow, we shall assume a groove ample enough so that the pressure drop in the groove itself is small. Initially we will neglect eccentricity and then apply a correction factor for this condition. The oil flow, then, is the amount which flows out of the two halves of the bearing in the direction of the concentration shaft. If we neglect the rotation of the shaft, we obtain the force situation shown in fig (6).

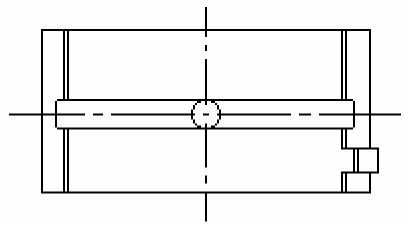


Fig 5. Central located full annular groove

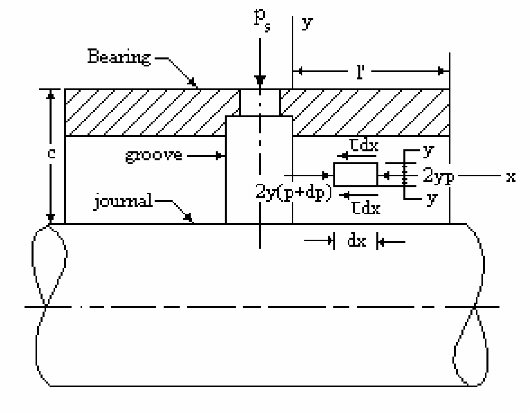


Fig 6. Flow of lubricant

The pressure distribution of the hydrodynamic bearing can be calculated by the following equation

$$P = (20\, Z'\, V\, D\, \epsilon / C^2)$$

$$\{(2+ \epsilon \cos\theta) \sin\theta \;/\; (2 + \epsilon^2)\,(1+ \epsilon \cos\theta)^2\}\theta \qquad (1)$$

4. EXPERIMENTAL ANALYSIS

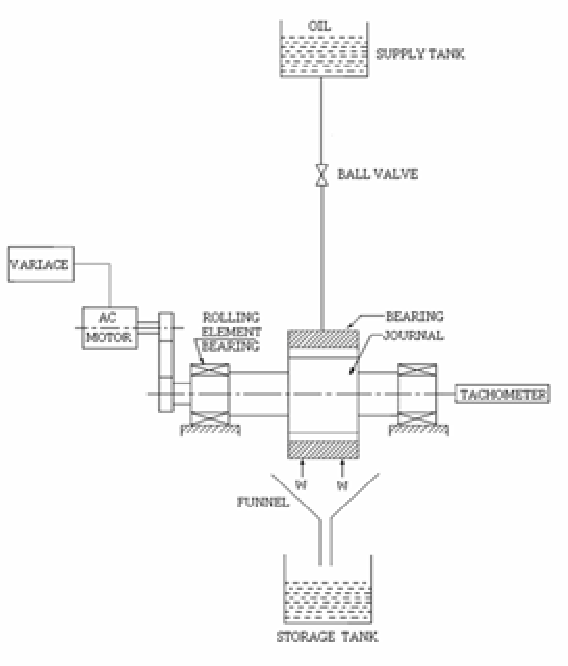
A journal bearing test rig, shown in fig (7), is used to test the 150 mm diameter and 75 mm long bearing. The test bearing is located between two anti friction support bearing. The bearing is loaded mechanically. The 500 micrometer radial clearance was provided in order to get turbulence with low viscous transformer oil. A variable speed DC motor is used to obtain the different journal speed up. The transparent bearing shell of methylmethacralyte is used. A red color petroleum dye is mixed in lubricant for better visualization. The temperature mapping of bearing inner surface is done by means of copper insert fitted along the circumference of the bearing. The RTD (resistance temperature detector) probe with a digital temperature display unit is used to record the temperature. The supply groove of 5mm width and nearly equal to length of bearing is located in vertical and supply to this groove is through a 2m long pipe of 5mm diameter. The flow rate of oil is measured manually at regular interval.

4.1 Operating Parameter

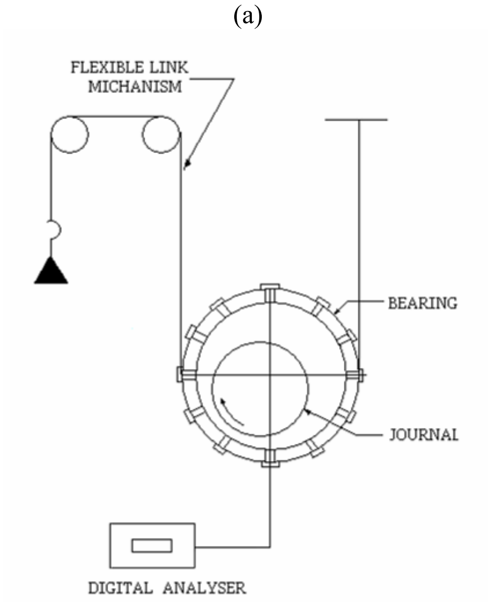
Material for journal: carbon steel with chrome plating.
Material for bearing: Methylmethacralyte
Types of lubricant: Transformer oil.
Journal radius: 70mm.
Bearing outer radius: 75mm
Bearing length: 75mm
Radial clearance: 0.5mm
Rotational speed range: 100-2000 rpm
Length of supply pipe: 2m
Diameter of supply pipe: 5mm
Supply pressure range: 0-0.3 N/mm^2 group (if any).

5. RESULTS

Results of pressure distribution of hydrodynamic journal bearing for various speeds and loading conditions are shown in the figure 8 to figure 10.



(a)



(b)

Fig 7. Experimental Test Rig

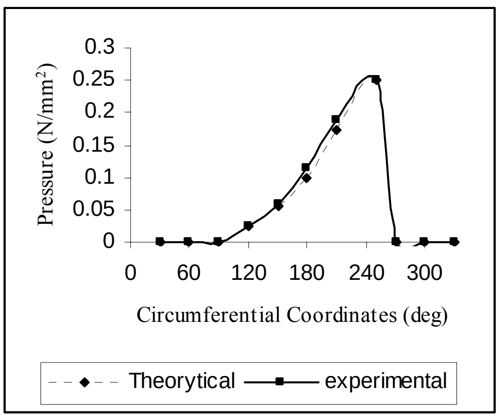


Fig 8. Graph of pressure versus Circumferential coordinates (W = 280 N, N = 800 rpm)

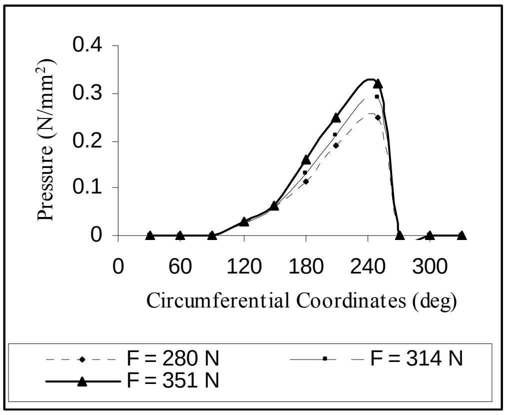


Fig 9. Graph of pressure versus Circumferential coordinates (Experimental results)

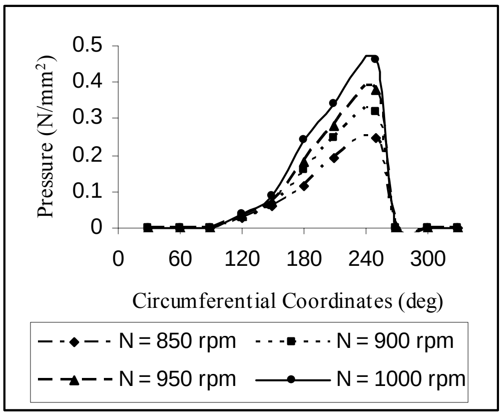


Fig 10. Graph of pressure versus Circumferential coordinates (Experimental Results)

6. CONCLUSIONS

From the pressure versus circumferential co-ordinates graphs one can say that maximum pressure obtained where oil film thickness is minimum & zero pressure in cavitations zone. Also as the speed and load on the bearing increase, the pressure also increases.

The pressure variation graphs shows that, pressure at the periphery of bearing follows the sinusoidal form, theoretically as well as practically.

7. REFERENCES

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8. NOMENCLATURE

Symbol	Meaning	Unit
Z’	Absolute viscosity of the oil	N.Sec/mm^2
V	Surface speed of journal	m/min
D	Diameter of journal	m
Є	eccentricity	mm
C	Diametral clearance	mm
θ	Circumferential coordinates	deg
h_0	Minimum film thickness	mm
μ	Coefficient of friction	μ