EXPERIMENTAL STUDY ON LOBE BEARING

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ABSTRACT

Analysis of multi-lobe journal bearing is very difficult, particularly when thermal and elastic effects are considered. Most of the earlier work was based on the assumption of rigid bearings and isoviscous lubricant in which the elastic deformations and the effect of temperature on viscosity is neglected. Also the realistic conditions like cavitation as well as modification of the lubricant flow due to bearing shell deformation were neglected. As in multi-lobe bearings, each lobe has different film shape and generates different amount of heat, the problem becomes complex and iterative. Such a heat generation also affects the bearing performance parameters like load carrying capacity. Hence, for accurate prediction of performance of hydrodynamic journal bearings, one must consider thermal as well as elastic effects in hydrodynamic solution.

1. Introduction

A bearing is a machine element which support another moving machine element (known as journal). It permits a relative motion between the contact surfaces of the members, while carrying the load.[1]

Generally fluid film bearings are used in high speed rotating machinery because they have low frictional resistance, excellent operating characteristics, high viscous damping etc.

Hydrodynamic journal bearings operate under high speed and the shearing of the lubricant film causes generation of heat within the oil film, which in turn increase the temperature of the lubricant fluid film and the bearing surface. As the temperature distribution in the bearing varies, the viscosity of the lubricating oil also varies and lowers the value of minimum fluid-film thickness. Thus, the flow field of the lubricant becomes disturbed and the bearing performance is affected. Hence, the assumption of iso viscous fluid, for bearings is less meaningful in reality. Therefore, for accurate prediction of the bearing performance, thermo hydrodynamic need to be considered. In such cases, the pressure and energy Equations are coupled and the operating viscosity is updated in accordance with temperature variation.

Also, the hydrodynamic journal bearing experiences elastic deformations when subjected to heavy loads. The bearing deformations are generally of the order of the magnitude of fluid film thickness, which cause a modification in the fluid film profile and in turn change the performance of a bearing system. Hence, the isoviscous and rigid bush assumptions made previously may not be appropriate for an accurate prediction of the performance of the journal bearing system.

Hence, to predict the bearing performance more accurately and realistically, it is essential to consider the elastic deformations in the bearing shell along with the thermal effect (i.e. thermo-elasto-hydrodynamic, TEHD effect) in the journal bearing systems.

Multi-lobe journal bearings are becoming more popular now-a-days as a substitute to cylindrical journal bearings due to their good stability at higher speeds. Most of the earlier work on multi-lobe bearings were based on theoretical analysis and that is too up to a very small extent for a few types of bearings. The complete analysis by considering both elastic as well as thermal analysis along with cavitation is very rare. Also very little experimental work is done on multi-lobe bearings. Hence there is a strong need to develop a theoretical tool and validate the results experimentally.

2. Literature Review

R. S. Kerlekar (2015) works on 3 lobe bearing. In this work, for analytical method he uses Fluid Structure Interaction technique. With the help of FSI technique he analyze the total deformation and pressure fluid film of 3 lobe bearing. He uses Aluminum as a bearing material. From his experimental and analytical results he concluded that for higher load and RPM, pressure generated is more. [2]

Mahesh Aher (2013) compares 3 lobe journal bearing with plain journal bearing for different speed and load. He conducted the experiment on journal bearing test rig. He uses Brass as a bearing material for plain journal bearing and lobe journal bearing. As per his research he concluded in 3 lobe hydrodynamic journal bearing, the nature of maximum pressure is steady. The pressure distribution obtained from the result lobe journal bearing has maximum pressure value compare to plain journal bearing. [3]

K PhaniRaja Kumar (2018) works on multi lobe bearing to find out the effect of Isotropic surface roughness, Transverse surface roughness and longitudinal surface roughness. Time transient analysis was performed by using the fourth order Runge Kutta method. Finite difference method (FDM) is used for pressure distribution over the bearing surface. As compare to both surface roughness, longitudinal surface roughness is more effective. Maximum pressure is obtained at midway of the lobes. [4]

A.KH. EL-Said (2017) works on 3 lobe bearing with its plain bushing and 3 lobe bearing with textured bushing with uniform micro protrusions. The analysis is carried out by finite difference method. This two types of 3 lobe bearing is compare with plain journal bearing for the plain and textured bushing. This comparison is done on the basis of load carrying capacity, frictional loss and dynamic stability. It was concluded that 3 lobe bearing with textured bushing raising load carrying capacity and reducing their friction losses. [5]

Nabarun Biswas (2013) works on 3 lobe bearing at 80000 rpm used in gas turbine. In this bearing he uses lubricating oil SAE 50 and analyze the oil property with the help of FLUENT software. With the help of GAMBIT 3 lobe bearing model is design. As per his research distribution of static pressure exits on top layer and pressure is constant on inner side of oil zone. At the middle of shaft region, static temperature distribution is maximum. [6]

K.M. Pandey (2011) works on 3 lobe bearing at 20000 rpm used in gas turbine. In this bearing he uses lubricating oil SAE 50 and analyze the oil property with the help of FLUENT software. With the help of GAMBIT 3 lobe bearing model is design. As per his research distribution of static pressure exits on top layer and pressure is constant on inner side of oil zone. At the middle of shaft region, static temperature distribution is maximum. [7]

A. Singla (2016) works on Elliptical journal bearing, At the central plane of finite elliptical journal bearing, he experimentally evaluate both the oil film pressure and temperature. For this work he uses various speeds under different load ranging from 500N – 2000N using HYDROL 32, 68 and 150 oil grades. Experimentally he concluded that HYDROL 68 is suitable oil, which gives optimum rise of temperature and pressure under all operating condition.

3. Problem Statement

Experimental investigation of the elastic deformations in the 3 lobe bearing shell along with the thermal effect.

4. Objective

The objective of the present work is to design 3 lobe bearing and to analyze the elastic deformations in the bearing shell along with the thermal effect arising due to the motion of the shaft at the range from 1500 to 4500 rpm and load is 600N.

5. Experimental Analysis

5.1 Test Bearing Data

Bearing Material	Aluminum
Internal Dia.	25 mm
External Dia.	35 mm
3 Lobes	At 120^0 Spacing
Lubricant	SAE 50
Lubricant viscosity	0.004 kg/m-s
Lubricant density	839 Kg/m^3

5.2 Properties of bearing material

Properties	**Aluminum**
Elastic Modulus, GPa	68.9
Density, kg/m^3	2700
Poisson's ratio	0.33
Thermal Conductivity, W/m K	157
Coefficient of thermal expansion,$*10^{-5}$ per 1°C	2.3

6. Results and Conclusions

6.1. Effect of elastic deformation

The peak pressure value for rigid, EHD and TEHD cases are 0.5114 MPa, 0.46269 MPa and 0.4599 MPa respectively. Thus the peak pressure value drops by 9.5 % when only elastic deformations are considered and by 10.05% when thermoelastic deformations are considered. This was due to the fact that, the bearing deformation increased the clearance space, thus allowing more flow of fluid and thus decreasing the peak pressure value.

6.2. Effect of rotational speed and eccentricity ratio

It was observed that as the speed increases, the value of peak pressure increases. The peak pressure value at 1500 RPM rotational speed is 0.2MPa and it increases to 0.33 MPa at 2500 RPM, 0.4626 MPa at 3500 RPM and 0.5987 MPa at 4500 RPM. So there is 65 %, 131% and 200% increase in peak pressure value from the lowest speed value. The pressure distribution was wide for lower speeds but becomes narrower with increase in speed. The peak temperature value corresponding to 1500 RPM speed is 311.7 k and it increases to 336.8 K at 4500 RPM. Hence there is an increase in the peak temperature value by 9% with an increase in speed.

As the eccentricity ratio increase, the deformation of the bearing shell also increases. The pressure at £= 0.2 is 0.2857 MPa and it goes on increasing up to 1.519 MPa which is 400% of the peak pressure at the lowest value of eccentricity ratio. It was seen that, the eccentricity ratio dominated the pressure rise as compared to effect of increase in rotational speed. But the peak pressure shifted to lobe 3, where minimum film thickness lies and the multi-lobe effect was lost.

The peak temperature values are also higher as compared to the effect of increase in speed. The temperature is 324.6 K at £=0.2 and increases to 341.4 K when £= 0.9. Thus, there is an increase of 5% from the lowest value which is less as compared to 9% increase in the temperature value due to speed effect.

6.3. Effect of Thickness

It is observed that as the thickness increases, the peak pressure value increases or vice versa. This is evident as less is the bearing shell thickness more is the deformation and clearance, hence more drop in peak pressure value. When 10 mm thickness was considered as a base value, then the % drop in pressure value was 4.96, 10.43 and 16.14 for 7.5mm, 5mm and 2.5 mm respectively. The peak temperature value also increased as an increase in bearing shell thickness and was maximum in lobe 3 where minimum film thickness occurred. The peak temperature values increases with rise in bearing shell thickness value.

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