INTRODUCTION

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Chapter 1

INTRODUCTION

1.1 GENERAL

A bearing is a system of machine elements whose function is to support an applied load by reducing friction between relatively moving surfaces. Hydrodynamic journal bearing is a fluid film bearing in which the load applied on the shaft is supported by the pressure developed in the lubricating film due to hydrodynamic action, (Fig.1.1).

Modern large capacity machines such as turbogenerators usually run at high speed and support heavy load. In high speed applications where bearing stiffness and stability are major considerations, non circular bearings give better dynamic performance.

When bearing is subjected to heavy loads the bearing shell deforms. The deformation of the bearing shell modifies the film thickness and this in turn affects the performance characteristics of the bearing. Therefore elastohydrodynamic analysis is considered to recompute the performance characteristics of circular and non circular bearings.

Various additives are added to the lubricant oil to enhance certain characteristics of the lubricant. These materials commonly known as additives are used as rust inhibitors (amine phosphates), corrosion inhibitors (sulphurised olefins), fire resistors (halogenated hydrocarbons), viscosity index improvers (polymethacrylate), powders of graphite and molybdenum disulphide) etc. The additives along with the contaminants form a dilute suspension of solid particles in the oil and it is treated as micropolar fluid. These suspended solid particles in the lubricating oil produce thickening of the oil affecting various performance characteristics of the journal bearing. Also there is an increase of viscosity in the part of the lubricating film, which is in the vicinity of the journal and bearing surfaces due to adhesion and other surface phenomena. The lubricant becomes

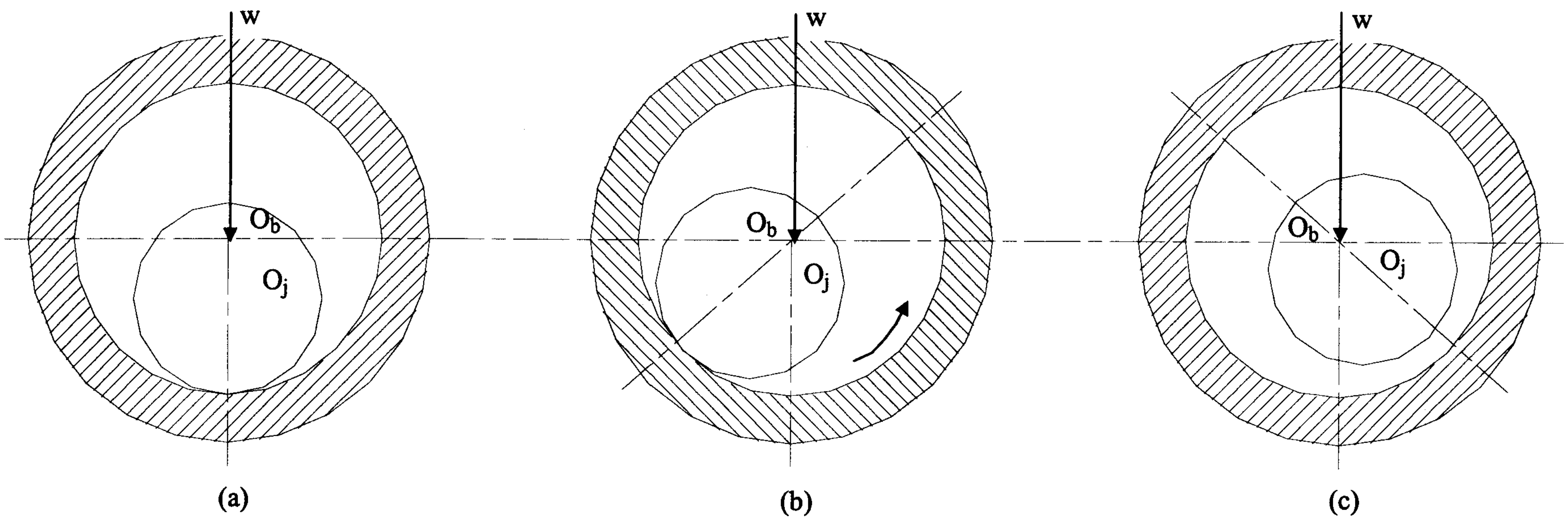

Fig. 1.1 Hydrodynamic Action of a Journal Bearing

non isoviscous. The non uniform distribution of the solid particles builds up concentration gradient which results in mass transfer of these particles across the film thickness and this affects bearing performance characteristics.

In the present work the static and dynamic performance characteristics of circular and non circular journal bearings are determined by considering the effect of deformation of the bearing liner and micropolar properties of the lubricant.

1.2 LITERATURE SURVEY

To identify the problem a survey of available literature on rigid hydrodynamic and elastohydrodynamic bearings is carried out. A literature survey is also carried out in the analysis of hydrodynamic journal bearings using micropolar lubricants.

1.2.1 Literature on Hydrodynamic Analysis

Conventionally, journal bearings are designed using the performance data computed on the assumptions that the bearing shell is rigid and viscosity of oil is constant. Based on these assumptions several investigations [1-8] are available on hydrodynamic analysis of circular and non circular bearings.

Hydrodynamic analysis of bearings are done by various methods. Singh et. al [9] determined various performance characteristics of a finite circular hydrodynamic bearing by solving Navier Stoke's equation for the flow field in the clearance space by using finite element method. Heller [10] solved fluid film equations for externally pressurized bearings for steady state performance and spring and damping coefficients using finite differences and a variable grid model. Ettler and Anderson [11] analyzed thrust bearings by using higher order finite elements for better accuracy in results. Chandrawat and Sinhasan [12] analyzed plain and two groove bearings by Gauss-Siedel iterative scheme and linear complementary problem approach. The results for these bearings wre compared when they operated in the laminar regime.

Goenka [13] presented a finite element formulation for the transient analysis of a journal bearing, which can be used for a partial or full arc bearing.

The results obtained for different cases of connecting rod bearings were presented. Chang [14] presented a new pressure starting boundary condition for calculating the oil film stiffness and damping coefficients of journal bearings using a perturbation method. Akers et. al [15] proved that in the case of journal bearings when out of balance load is added, the bearing becomes more stable if friction is included. Nikolajsen [16] investigated the stability of plain journal bearings and floating ring journal bearing against fractional frequency of whirl under laminar fluid film flow conditions. Shelly and Ettlest [17] applied finite element method for the whirl analysis of plain bearings. He presented several locus paths to show the separate and combined effects of rotor unbalance and unidirectional loading over a range of rotational speeds.

Rohde and Ezzat [18] analyzed hybrid journal bearing considering the compressibility of the lubricant in the bearing. Etsion and Pinkus [19] analyzed short journal bearings and showed that the bearing performance characteristics are functions of Sommerfeld number, length over diameter ratio and starting conditions at the film inlet. Taylor [20] obtained a numerical solution for the pressure distribution in a porous thrust bearing. He showed that the dynamic spring force and dynamic damping force vary with displacement for a given frequency.

Bearings may operate in the turbulent regime at high speeds. Analysis has been done by considering the variation in viscosity of the lubricating oil due to turbulence. Hydrodynamic bearings were analyzed by considering turbulence in oil flow by Wilcock [21]. He proved that in turbulent lubrication greater film thickness and larger power loss occur than from laminar theory. Venkateswarlu et.al [22] analyzed journal bearings considering the three dimensional motion in the lubricant layer operating from laminar to turbulent condition. Velocity and pressure fields were calculated from the governing differential equations using an iterative numerical method. Gardner and Ulshmid [23] determined the operating characteristics of two different types of journal bearings to indicate the effects of turbulence on these characteristics.

Literatures are available considering the non Newtonian nature of fluids in lubrication [24-26]. Dien and Elrod [27] analyzed a modified form of Reynold's Equation for fluids showing inelastic and non-Newtonian characteristics. The static and dynamic characteristics of a plane journal bearing were determined by Malik et. al [28] for non-Newtonian lubricants. He solved Reynold's Equation and the steady state pressure distribution was established by an iteration scheme.

Finite element method has been used for the hydrodynamic analysis of non circular bearings also. Goenka and Booker [29] extended the finite element formulation of cylindrical bearing to include non cylindrical bearings. In his analysis he determined the optimum bearing shape to maximize the minimum film thickness. Ashok Kumar et.al [30] determined the static and dynamic characteristics of an elliptical bearing by using a variational solution. Kumar Vaidyanathan et.al [31] analyzed non circular bearings considering the effects of turbulence and cavitation. Different static characteristics were determined for circular and two lobe bearings. Malik et.al[32] studied three lobe bearings assuming the viscosity to be constant. Malik et. al [33] analyzed the performance characteristics of tilted three lobe bearings. The effect of tilt angle on the dynamic performance characteristics was studied. Li et. al [34] analyzed the stability of elliptical, offset elliptical, three lobe and four lobe bearings using a numerical fast Fourier transform analysis.

1.2.2 Literature on EHD Analysis

Elastohydrodynamic analysis is considered as important because the effect of elastic deformations of the bearings shell on the performance characteristics of the journal bearing system is quite significant particularly for heavily loaded bearings. Flexibility of the bearing shell affects the performance of the bearing especially at higher values of deformation coefficient. During the last two decades elastohydrodynamic studies have received considerable attention with the recognition that large changes in the performance characteristics occur with flexibility of bearing liner under heavy load.

One of the first attempts on the study of EHD lubrication was made by Osterle and saibel [35 – 36]. For slider bearings, assuming the bearing pad to be a semi infinite elastic body, they determined the deformation and pressure field by neglecting the side leakage. They concluded that due to bearing deformation, pressure distribution and load capacity are different from those calculated for rigid bearings. For an infinitely long bearing, Higginson [37] studied the effects of elastic deformation of bearing shell on the journal bearing performance; these results were only qualitatively good because of his simplified approach to the calculation of elastic deformation. The theoretical investigation for elastic deformation of bearings was prompted by the experimental work of Carl [38] presented in 1964. He experimentally demonstrated the effect of bearing deformation on the pressure distribution in the clearance space of the journal bearing and showed that the distortion of bearing was important at pressures of the order of 13.8 Mpa. O' Donoghue et. al [39] studied the effect of elastic deformation of bearing on the performance of infinitely long journal bearing. Their studies also indicated that the elastic deformation of bearing introduces marked changes in bearing performance characteristics and their results confirmed the findings of Carl's experiments of maximum pressure and the extent of positive pressure film. An iterative procedure was used to find the pressure distribution satisfying both the hydrodynamic and elastic equations. A similar analysis for short bearings with flexible liners was also described by O'Donaghue et al [40]. Other studies [41-43] also demonstrated significant changes in pressure distribution in the clearance space of bearings when the deformation of the bearing shell was considered.

Benjamin and Castelli [44] solved Reynold's equation and three dimensional elasticity equations for analyzing journal bearings and they proved that the load capacity and attitude angle are reduced when deformations of bearing liner are considered. They showed that the elastohydrodynamic pressure distribution is significantly different from that obtained by neglecting the elastic deformation. Oh and Hubner [45] used finite element technique to solve elastohydrodynamic finite journal bearing problem. Reynold's equation for fluid film and three dimensional elasticity equations for the bearing housing were solved simultaneously using an iteration scheme. The analysis yielded pressure distribution and displacement distribution. From these distribution s stresses in the bearing and minimum film thickness in the lubricant were determined. Jain et. al [46] determined the various static performance characteristics of the bearing. Reynold's equation was solved to obtain the pressure distribution by using FEM. Three dimensional elasticity equations were used to determine the bearing shell deformations. The final pressure was obtained by using an iteration scheme.

Stafford et. al [47] used simplified equations for studying elastohydrodynamic lubrication. Elastohydrodynamic analysis of a connecting rod big end bearing was done by them assuming a two dimensional structural model. They also assumed half Sommerfeld conditions to simplify calculations. Frene et. al [48-49] analyzed bearings by using flow field equations and two dimensional elasticity equations. Labouff and Booker [50] studied the effect of mesh density and housing flexibility on bearing performances. They also proved that there is change in the maximum film thickness and maximum film pressure obtained in the bearing when flexibility is considered.

Conway et. al [51-53] studied the effect of pressure on viscosity of the lubricating oil in EHD analysis. The oil was first assumed to be isoviscous and the analysis was then extended to the case of pressure dependent viscosity and they showed that the performance characteristic depend on both flexibility of the bearing liner and the dependence of viscosity on pressure. Taylor and Callaghan [54] determined pressure distribution in elastic isoviscous and elastic viscous cases. The effect of increasing the effective elastic modulus of the cylinder material on minimum film thickness also is discussed. Jain et. al [55-56] analyzed flexible bearings considering the variation of viscosity in pressure and they proved that piezo viscous lubricants support more loads

Elastohydrodynamic analysis of two axial groove journal bearings operating in laminar and super laminar regimes was done by Sinhasan and Chadrawat [57]. Literature is available on the analysis of porous flexible bearings

[58-60]. . Performance characteristics of capillary and orifice compensated flexible thrust bearings were analyzed by Sinhasan et. al [61]. Fantino et. al [62] analyzed connecting rod bearing operating with piezo viscous lubricants. In the analysis, plane elasticity relations were used by them to calculate the bearing displacement. Partial bearing in different flow regimes were analyzed by Jain et. al [63]. Kohnos et al [64] analyzed the elastohydrodynamic lubrication characteristics of journal bearings with combined use of the boundary-element method and finite-element method. The boundary-element method was used to calculate the elastic deformation of the bearing housing and the finite-element method was applied to the solution of the Reynolds equation.

Literatures are available on EHD Analysis of non circular bearings also. Hamrock and Dowson [65] analyzed elliptical contacts for materials with low elastic modulus. Tae Jo Park and Kyung – Woom Kim [66] analyzed elastohydrodynamic lubrications of elliptical contacts and determined pressure distribution for various film thicknesses. Prabhakaran Nair et. al. [67] determined various static and dynamic characteristics of elliptical bearings by using finite element method. Three dimensional Navier Stoke's equation and continuity equation governing lubricant flow in the clearance space of the journal bearing and the three dimensional elasticity equations governing the displacement field in the bearing shell were solved to get the pressure distribution in the bearing. Various characteristics of the bearing were presented for a range of deformation coefficients, which take in to account the flexibility of the bearing liner. Goyal and Sinhasan [68] presented static and dynamic performance characteristics of two-lobe journal bearings with non-Newtonian lubricants. The Navier-Stokes and continuity equations were solved for Newtonian fluids using the finite element method in the cylindrical coordinates representing the flow field in the clearance space of a two-lobe journal bearing. The non-Newtonian effect was introduced by modifying the viscosity term for the model in each iteration. Deformation of the bearing shell was obtained by solving the three-dimensional elasticity equations. Chandrawat and Sinhasan [69] studied two-lobe journal bearing and determined the static performance characteristics of a flexible shell two-lobe bearing operating in laminar and turbulent (including transitional flow) regimes in the case of isoviscous and piezoviscous lubricants. Prabhakaran Nair et. al [70] extended the analysis used for elliptical bearings to include three lobe bearings. Chandrawat and Sinhasan [71] determined various static performance characteristics of a flexible shell three lobe bearing operating in the laminar and turbulent regime using isoviscous and piezoviscous lubricants.

Prabhakaran Nair et. al [72-81] analyzed circular and non circular bearings with flexible shell placed in a rigid housing considering also the change in temperature of Newtonian lubricants.

1.2.3 Literature on Micropolar Lubricants

The behaviour of micropolar fluids is explained by the theory of 'fluid microcontinua'. In this theory the intrinsic motion of molecular or granular constiluents of the medium is taken into account. The earliest formulation of a theory considering micromotions and deformations was done by Eringen [82]. He considered the continuum media as a set of structured micro volume elements whose kinematics can be independent of the motion of macrovolume. This theory has been further simplified to the theory of micropolar fluids [83] by ignoring the deformation of micro elements. Assuming the solid contaminants and additives in the lubricant oil to be rigid spherical particles, dilute suspensions of the same in the oil can be treated as micropolar fluids.

Allen and Kline [84] analyzed two dimensional problem of lubrication with micropolar fluids to get an approximate solution to the problem of a slider bearing. Shukla [85] studied the effects of additives on load carrying capacity of a hydrostatic bearing. He showed that the load carrying capacity increases with increase in the concentration of additives. Khader and Vachon [86] analysed laminar flow of micropolar fluids between two circular disks. The analysis indicated that significantly larger resultant pressures and shear stress occur in the lubricant due to the presence of the microstructures. They showed that load carrying capacity increases when compared with the result of a similar analysis employing nonmicropolar fluids. A one dimensional slider bearing with micropolar fluid was analyzed by Shukla and Isa [87] by considering a generalized form of Reynold's equation including micropolarity. The derivation of a generalized Reynold's equation and an analysis for steady state characteristics of one dimensional journal bearing under the condition of micropolar lubrication was presented by Zaheeruddin and Isa [88]. Prakash and Sinha [89] studied the behaviour of micropolar fluids when it passes through narrow passages. The studies on the squeeze film characteristics of micropolar fluid lubricated journal bearing for both full and half bearings were also done by Prakash and Sinha. [90]. A practical example of using micropolar model is in the design of journal bearings in the area of nuclear power where heat transfer agent sodium is used as lubricant [91]. Albert et. al [92] studied hydrodynamic lubrication of a slider bearing with oil containing additives. It was shown that the load capacity and the frictional force of the slider bearing increase with increase in concentration of additives. The static characteristics of a journal bearing with micropolar lubricants were determined by Prakash and Prawal Sinha [93] considering the problem as that of a steady laminar flow of an incompressible micropolar fluid. The steady state performance of infinitely long journal bearing based on the theory of micropolar fluids revealed that the prominent feature of a micropolar fluid is an increased effect of viscosity which in conformity with experimental results. Alber et. al [94] developed an analytical hydrodynamic model to predict the behaviour of a two phase lubricant. Analysis showed that the presence of suspended solid particles in a Newtonian lubricant enhanced the load carrying capacity. The short bearing performance with micropolar fluids has been analyzed by Nicolae Tipei [95]. The performance of finite width journal bearings lubricated with micropolar fluids is analysed by Tsai-Wang HuangCheng-I Weng and Chao-Kuang Chen [96] using the finite difference method to solve the generalized three-dimensional Reynolds equation. The characteristics of finite journal bearings with micropolar and Newtonian fluids are obtained and presented graphically. The analysis reveals that the prominent feature of increasing load capacity and decreasing friction coefficient for micropolar fluids is more pronounced at higher eccentricity ratio. Khonsari and Brewe [97] studied the performance parameters for a journal bearing of finite length lubricated with micropolar fluids. They proved that a significantly higher load-carrying capacity than with Newtonian fluids result, depending on the size of the material characteristic length and the coupling number. It is also shown that, although the frictional force associated with a micropolar fluid is in general higher than that of a Newtonian fluid, the friction coefficient of micropolar fluids tends to be lower than that of Newtonian fluids. A modified momentum equation and continuity equation were used to analyze the bearing. Prabhakaran Nair et. al [98] analyzed hydrodynamic circular journal bearing with micropolar fluids. Prabhakaran Nair et. al [99] also determined the static and dynamic characteristics of circular rigid bearings operating with micropolar fluids. Albert E. Yousif and Thamir M. Ibrahim [100] determined the characteristics of conventional infinitely long thrust bearings by solving numerically the modified Reynolds equation for the steady laminar flow of an incompressible micropolar fluid that has an increased effective viscosity, especially in thin films. They proved that the bearing performance, in general, is improved by micropolar lubricants. The static and dynamic characteristics of a hydrostatic circular thrust bearing were studied by Jaw-Ren Lin [101]. Modified Reynold's equation and flow continuity equation were used to solve the problem and results were obtained by using a perturbation technique. The convergence to the stationary solution for large viscosity flows was analyzed by Lukaszewicz [102]. He determined long time behaviour of micropolar fluids in two dimensional flows. Raghunandana et.al [103] studied the effect of non Newtonian behaviour of lubricants, resulting from addition of polymers, on the performance of hydrodynamic journal bearings. Das et. al [104] solved a modified Reynold's equation based on the theory of micropolar lubricant by using finite difference method to obtain the film pressure for a hydrodynamic bearing with misalignment. With the help of this pressure, steady state characteristics of the bearing were determined. Literature survey shows that the performance characteristics of hydrodynamic circular bearings considering the effect of deformation of bearing liner and mass transfer of micropolar lubricant on the performance have not been investigated. It is also seen that no literature is available for static and dynamic analysis of elastohydrodynamic noncircular bearings operating under micropolar lubricant. So it is felt that there is a need to compute the performance characteristics of circular and noncircular bearings considering the effect of deformation of bearing liner and mass transfer of micropolar lubricants.

1.3 OBJECTIVES OF THE PRESENT WORK

In the present work the static and dynamic characteristics of the rigid or deformable, circular or non-circular (two lobe or three lobe) bearings operating under micropolar lubricants have been studied A two dimensional Reynold's equation was derived from Navier Stoke's equation and Fick's mass transfer equation and this modified Reynold's equation and three dimensional elasticity equations were solved to compute the pressure distribution of the flow field and deformation of the bearing respectively. The finite element method, which offers several advantages, when applied to lubrication problems is used to analyze bearings.

In the case of micropolar fluids the variation of viscosity is governed by the following relation [85].

$$\mu = \mu_0 (1+\lambda c_r) \tag{1.1}$$

where λ is a parameter which specify the shape, size, deformation, distribution and material properties of the additive particles.

Assuming spherical non deformable additive particles according to Einstein's relation [94].

$$\mu = \mu_0 (1+2.5c_r). \tag{1.2}$$

The static characteristics in terms of load capacity, attitude angle, end leakage and frictional force and dynamic characteristics in terms of stiffness coefficients, damping coefficients, threshold speed and damped frequency of whirl have been determined for rigid and deformable, circular and non circular bearings operating with Newtonian and micropolar lubricants. Results are obtained for a wide range of deformation coefficients and volume concentration of additives.The effect of mass transfer of additives on the performance characteristics of bearings are also analysed

The computed results show that the presence of additives in the lubricating oil affects the performance characteristics. The change in volume concentration of additives and mass transfer rate produce significant effects in the performance characteristics of the bearing especially when the bearing operates at higher eccentricity ratios. The deformation of the bearing liner also affects the performance characteristics of the bearing.

1.4 THESIS ORGANIZATION

The work carried out is organized into five chapters. The first chapter Introduction high lights the literature survey and states the objectives and scope of investigation. The second chapter deals with the theoretical analysis of bearings. Governing differential equations and finite element formulations for fluid flow field and displacement field with boundary conditions are discussed in this chapter.

The static and dynamic performance characteristics of the bearing are described in Chapter 3.

The fourth chapter contains details of solution schemes. Flow charts for various segments of solution procedure are also given in this chapter.

The results and discussions are given in Chapter 5. Scope for future studies of the work also is given in this chapter.

The non circular bearing geometries and details of FEM formulations are given in Appendices A_1 and A_2.