CHAPTER 1

HYDRODYNAMIC LUBRICATION

1.1 Introduction

A fluid film bearing may be defined as a bearing in which the opposing or mating surfaces are completely separated by a layer of fluid lubricant. A widely used bearing type is the journal bearings with application in many cases such as different types of engines, compressors, turbines, electric motors and electric generators.

Nearly, all heavy industrial turbo machines use fluid film bearings of some type. The two primary advantages of fluid film bearings over rolling element bearings are their ability to absorb energy to damp vibrations and their longevity due to the absence of rolling contact stresses.

The purpose of lubrication is to separate two surfaces sliding past each other by a film of some material which can be sheared without causing any damage to the surfaces. To ensure that no contact occurs between the opposing surfaces, the dimensions of the bearing surfaces are chosen, such that a lubricant film of sufficient thickness is available under all operating conditions.

In surveying the entire field of lubrication with its many vital physical and chemical manifestations, it seems that the one phenomenon that occurs most frequently and with great importance, is the creation of a load-carrying hydrodynamic fluid film. This kind of lubrication, in which the load carrying pressure is self-induced, is known as hydrodynamic. Hydrodynamic action is generally described for only two rather simple types of elements: the cylindrical journal bearing, the tilting-shoe and tapered-land bearing. But, because the creation of this fluid film is so universal, it operates effectively with many geometric forms. The simple case of one solid body with a plane surface sliding steadily over another plane surface, such that they are slightly inclined to each other, was analyzed first by Reynolds in 1886. This theory was later modified to take into account the side leakage of the lubricant and also to be applied to the journal bearings.

The analysis of journal bearings is probably the most important part of the classical hydrodynamic theory of lubrication. There are many calculation methods which aid the design of steadily loaded pressure-fed 360° hydrodynamic journal bearings. These are generally based on the approximate analytical or numerical solution of the so called Reynolds lubrication equation and the results are mostly displayed in dimensionless form as a variety of design charts.

In all journal bearings, the shaft is in eccentric position within the bearing clearance, which depends on the value of shaft load. At the equilibrium position, the pressure field generated in the tapered film between the shaft and the bush balances the applied load. If this load is relatively small in magnitude, the corresponding eccentricity will be small and the shaft and bush run almost concentric. At high loads, the reacting pressure field is produced by a highly tapered lubricant film associated with the high eccentricity.

For all bearing operations, there should be a sufficient lubricant supply to ensure that in the tapered or converging film section, the lubricant extends completely across the full width of the bearing. This is achieved by using an axial groove at the lubricant inlet or a circumferential groove at the midsection of the bearing. The former type is used for predominantly unidirectional loads, while the latter is generally employed when the load varies in direction by more than 180°.

Once geometry has been decided upon for a particular application, the global performance parameters which are of most concern to the designer are load carrying capacity, power loss and the lubricant flow rate. As mentioned above, these quantities are generally obtained from the solution of the Reynolds equation which is adequate for a vast number of problems occurring in medium ranges of the load and speed. However, "off design" loads and speeds may occur which result in unsatisfactory operating conditions, i.e, different from those for which the bearing has been designed. For example, at higher speeds and loads, excessive heat may be generated which can bring failure because of excessive temperature in the bearing. Thermal effects are usually included in a bearing design by assuming that the heat generated by viscous action is convected away by the continuous flow of lubricant and by conduction through the solid elements of the bearing.

Another problem which is of interest in lubrication technology is the stability of flow between rotating eccentric cylinders. It is clear that the lubricant flow is laminar for small values of Reynolds number. Taylor vortices are known to occur when the Taylor number, T, defined as:

$$T = \frac{\rho \bar{V} c}{\mu}\sqrt{\frac{c}{r_S}} \tag{1-1}$$

reaches the value of 41.2, after which the flow breaks down to turbulence [1]. Transition between vortex and turbulent flow is well known to be more gradual and can not thus be characterized by one single number. Many experiments show a wide range of $100 \leq T \leq 150$ for fully turbulence [2]. In this expression, ρ and μ are the fluid density and viscosity, r_s and $\bar{V}$ are the radius and circumferential velocity of the inner cylinder respectively, and c is the radial clearance between cylinders.

Taylor's study was followed by a series of studies by other investigators, using different theories to analyze the stability of flow between rotating eccentric cylinders. It was found that the critical Taylor number, T_c, varies with the eccentricity ratio, ε. With regard to the relationship between T_c and ε, there are some discrepancies between the results of these studies. Some results indicate that T_c

initially decreases with ε, and others show that T_c monotonically increases with increasing ε.

Moreover, a difficulty arises from the behavior of the fluid film in the diverging flow areas of the bearing. It is well known that under hydrodynamic lubrication conditions, a negative pressure is developed in the diverging flow region, causing the lubricant to cavitate, which calls for a careful treatment of the cavitated region.

Thus, if the performance of the bearings under extreme conditions is to be assessed, a more rigorous analysis must be employed.

1.2 Literature Survey

The analysis of journal bearings is probably the most important part of the classical hydrodynamic theory of lubrication. It is the most difficult as the integration of journal bearing equations is more complex than for the classical tilting-pad thrust bearing.

In this section, a few papers concerning the history of study of the lubrication theory and flow between rotating eccentric cylinders are reviewed. This is followed by reviewing papers on the thermohydrodynamic (THD) analysis of journal bearings.

As it was mentioned before, for obtaining the lubricant pressure distribution in journal bearings, the approximate Reynolds equation was solved numerically or analytically by many investigators. This equation is obtained from the Navier-Stokes equation with considering the following assumptions:

1- Neglecting the effect of flow curvature.

2- Considering uniform pressure across the lubricant film.

3- Omitting the inertia terms in momentum equation.

It is clear that the Reynolds equation may be accurate under the condition of low Reynolds number and small clearance ratio. Thereby, applying this equation for hydrodynamic analysis of journal bearing running under high speed in which the lubricant flow becomes turbulent is questionable.

As explained later, in many studies by investigators, the Reynolds equation in various forms was solved in thermohydrodynamic analysis of journal bearings for both laminar and turbulent regimes. Besides, there are a few number of studies in which computational fluid dynamic (CFD) techniques are employed, but they are restricted to the assumption of laminar flow, such that to the best of author's knowledge, the THD characteristics of turbulent lubricant flow in journal bearings have not been analyzed by CFD techniques. The following studies are those in which the Reynolds equation was employed in THD analysis of journal bearings.

The theoretical treatment of turbulent flow in bearing fluid film was first attempted by Wilcock [3] and then by Constantinescu [4], who used mixing length theory of Prandtl. Constantinescu assumed that the mean fluid inertia stresses are negligible compared to fluctuating inertia forces, known as turbulent stresses. However, at low Reynolds number, this argument may not stand, because mean fluid inertia forces may be of the same order as viscous forces. Following Prandtl's mixing length theory; Constantinescu decoupled the different equations, which could lead to velocity expression for journal bearings. He presented the analytical expression for pressure in turbulent condition. Then, the basic turbulent lubrication theory has been developed by several other researchers [5-8].

In 1965, Ng and Pan [9] studied the adaptability of the "law of wall" for turbulent shear flows to analyze turbulent lubrication. This new approach took into account many well-established facts concerning turbulent shear flows. Isotropy of turbulent momentum transport (eddy viscosity) was assumed in treating non planar mean flows. A linearized version of the governing differential equation was established. Sample results agreed well with experimental data.

In 1967, Pan and Vohr [10] studied possible fluid dynamic processes in thin fluid of low kinematic viscosity fluids and parameters governing the flow regimes were identified. Their work was much directed to fluid seals as it is to fluid film bearings. They concluded that inertia effects, in the presence of turbulence, are the most important problems which remain to be solved.

Thermohydrodynamic lubrication in laminar and turbulent regimes was considered by Safar and Szeri [11] in 1974. They endeavored to develop a method for the calculation of bearing performance in turbulent regime that took into account the dependence of lubricant viscosity on temperature. In that treatment, the shaft was isothermal and the bearing conducted heat only in the radial direction. The resulting effective viscosity was used to calculate the performance of finite journal bearings. Sample calculations showed that the mathematical model is sensitive to the values of ε (eccentricity ratio), Re and T_s (dimensionless shaft temperature). These calculations also indicated significant thermal effects on bearing performance in both laminar and turbulent regimes.

A thermohydrodynamic analysis of journal bearings operating under turbulent conditions was introduced by Bowen and Medwell [12] in 1978. The theoretical analysis of a journal bearing operating under turbulent conditions requires the simultaneous solutions of the continuity, momentum and energy equations. However, because of the extreme complexity of this type of flow phenomenon, several simplifying assumptions were considered in that study and the following approximate Reynolds equation which describes the pressure field generated in the lubricant film was obtained.

$$\frac{\partial}{\partial x}\left(\frac{G_x h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{G_z h^3}{\mu}\frac{\partial p}{\partial z}\right)=\frac{v}{2}\frac{\partial h}{\partial x} \qquad (1\text{-}2)$$

in which G_x and G_z are equal to 12 for laminar flow and are given by $G_x = [12 + 0.013\,\mathrm{Re}^{0.9}]^{-1}$ and $G_z = [12 + 0.0043\,\mathrm{Re}^{0.96}]^{-1}$ for turbulent flow. Also, h is the local film thickness, V is the surface velocity of the shaft, and Re is the local Reynolds number.

Neglecting the cross-film and stream wise conduction effects (the adiabatic condition), the following energy equation was derived.

$$\rho c[(\frac{vh}{2}-\frac{G_x h^3}{\mu}\frac{\partial p}{\partial x})\frac{\partial T_m}{\partial x}-\frac{G_z h^3}{\mu}\frac{\partial p}{\partial z}\frac{\partial T_m}{\partial z}]=-\tau_c v+\frac{h^3}{\mu}[G_x(\frac{\partial p}{\partial x})^2 z+G_z(\frac{\partial p}{\partial z})^2] \quad (1\text{-}3)$$

where $T_m(x, z)$ is the local mean temperature, c is the specific heat of the lubricant and τ_c is the Couette shear stress equal to one for laminar flow and given as $1+0.0012\ Re^{0.94}$ for turbulent flow. The variation of the lubricant kinematic viscosity was assumed to be represented by the Walter equation as:

$$\log[\log(\upsilon+0.6)]=10.36-41.5\log T \quad (1\text{-}4)$$

By solving the energy equation and the turbulent Reynolds equation simultaneously, estimates for the temperature and pressure distributions in a hydrodynamic journal bearing were obtained. It was concluded that the inclusion of the variation of lubricant properties due to heat dissipation in a bearing causes a considerable reduction in its load-carrying capacity at high speed ranges. Also, the bush torque and therefore, the bearing power loss do not appear to be reduced to the same extent.

Thermohydrodynamic effects in journal bearings operating under steady-state loading were investigated by Khonsari and Beaman [13] in 1986. In that analysis, an analytical model for the finite journal bearing was formulated. The model included correction factors for the cavitation effects and the mixing of the recirculating oil and supply oil at the inlet. Assuming homogeneous mixing of the recirculating oil, Q_{rec}, and the supply oil, Q_s, film temperature immediately after the mixing point was represented by the following equation:

$$T_{mix} = [Q_{rec}T_{rec} + Q_S T_S]/[Q_S + Q_{rec}] \tag{1-5}$$

A finite difference computer program was developed to solve numerically the Reynolds and energy equations for lubricant flow, Laplace heat conduction equations for bush and shaft and a viscosity-temperature relation, simultaneously. Computational findings showed the oil-bush interface temperature drops slightly in the vicinity of the inlet followed by a rapid rise in the circumferential direction and drop in the cavitated region as expected. Also, the temperature gradient in the cross film direction was found to be much greater than those of circumferential direction.

In 1986, Medwell and Gethin [14] investigated a misaligned turbulent journal bearing analytically. The analysis dealt with the evaluation of pressure and temperature fields which were generated in thin fluid films of varying thickness. The particular problem of a misaligned journal bearing was studied by solving simultaneously the Reynolds and energy equations, which also included the effects of viscous dissipation and the variation of fluid viscosity with temperature. The method was used to predict pressure and temperature fields as well as global performance parameters for a typical journal bearing operation.

In 1994, Nagaraju et al. [15] investigated a thermohydrodynamic solution of a finite two-lobe journal bearing. They obtained temperature and pressure distributions by the simultaneous solution of the generalized Reynolds equation with the energy equation and the heat conduction equation. In the analysis, the static characteristics (in terms of load capacity, attitude angle, end leakage and friction parameter) and the dynamic characteristics (in terms of critical mass, threshold speed and damped frequency of whirl) were determined. Various values of the thermal parameter at each eccentricity ratio, ranging from 0 to 0.45, were used. Also, the ellipticity ratio and the aspect ratio were 0.5 and 1, respectively.

It was reported that at any eccentricity ratio, the load-carrying capacity decreases with increase in the values of the thermal parameter, and a large reduction is obtained at higher values of eccentricity ratio. Also, within the given operating range of load, the end leakage and friction parameter increase with increase in the thermal parameter at any load. It should be noted that at lower loads the critical mass and threshold speed remains almost the same when the speed of the bearing is increased and at any load, the damped frequency of whirl decreases with increase in thermoviscous coefficient.

Thermohydrodynamic simulation in the form of design charts for journal bearings was presented by Khonsari et al. [16] in 1996. In that study, a THD analysis of journal bearings referred to a realistic solution of the generalized Reynolds equation in which the viscosity field was predicted based on the computation of temperature, obtained from the conservation of energy. In those calculations, cavitation effect was also considered and the thermal characteristics of lubricant such as viscosity and thermal conductivity varied in the cavitation zone using an affective length model. Simultaneous solution of generalized Reynolds equation, energy equation and heat conduction equation in the bearing was obtained. The numerical solution scheme was iterative and required several trial-and-error attempts to satisfy the boundary conditions, in particular, in predicting the shaft temperature which was considered as an isothermal element with zero net heat flux. The results were presented in the form of design charts for journal bearings which enable one to evaluate the maximum temperature and an effective bearing temperature.

The effects of variable density and variable specific heat on maximum pressure, maximum temperature, bearing load, frictional loss and side leakage in high-speed journal bearing operation were examined by Chun [17] in 2004. In that study, the influences of variable density and variable specific heat on a high-speed journal bearing were compared with those using constant density and constant specific heat. In order to investigate whether the conventional approach using constant density and constant specific heat was suitable for analysis of the turbulent lubrication problem of a high-speed journal bearing, some physical parameters of high-speed bearings were investigated under the following three conditions.

Condition 1: constant density and constant specific heat of oil.

Condition 2: variable density and constant specific heat of oil.

Condition 3: variable density and variable specific heat of oil,

and the calculated results under these conditions were compared with one another. Through the results of this analysis, the following conclusions were drawn:

1- Under high shaft speed operation, the consideration of variable density and variable specific heat on the calculation of the bearing load and frictional power loss can not be ignored.

2- At higher shaft speed, if the real density and specific heat are not considered, the calculated temperature may appear excessive compared with real temperature.

Transition criteria for flow between concentric rotating cylinders were studied experimentally by Frene and Godet [2] in 1974. High speed Couette flow was studied for two small clearance ratios of 0.0055 and 0.0031. Water flow pattern visualizations and torque measurements were performed. Some of the results have been presented as a series of photographs showing the flow pattern at different speeds. From the descriptions of photographs, the following results can be summarized as are given in Table (1-1).

Table (1-1): Flow pattern between two concentric rotating cylinders at different speeds

Flow pattern	Taylor Number	Reynolds Number	V(rpm)	c/R
Occurrence of the first vortices	40.0	545	442	0.0055
Appearance of vortex distortion	54.9	740	600	0.0055
Superposed turbulence in vortex effect	183	2468	2000	0.0055
Last vortices in essentially turbulent flow	277	3730	3022	0.0055

where R and V are the radius and peripheral speed of the inner cylinder and c is the radial gap.

An experimental and theoretical investigation of thermohydrodynamic performance of finite length journal bearings has been done by Ferron et al. [18] in 1983. That analysis took into account the heat transfer from the film to the shaft and bush. Also, cavitation was considered in that study and the distributions of pressure and temperature along the wall of the bearing were obtained. The values of eccentricity for different speeds and different values of loads were calculated, too.

Temperature distribution in a full circular bearing was measured by Mitsui et al. [19] in 1986. The effects of journal speed, lubricant viscosity, and clearance ratio on the bearing maximum temperature and its location were discussed. The bearing maximum temperature was found to increase considerably with the increase of speed or lubricant viscosity and also with the decrease of clearance ratio. Also, it was found that the angular position of bearing maximum temperature varies considerably with the increase of speed towards the direction of journal rotation from the upper stream side of minimum film thickness location to the lower stream side of it.

Temperature distribution and the heat transfer in journal bearings were experimentally investigated by Fitzgerald and Neal [20] in 1992. Tests were performed on a 76 mm diameter bearing with two values of length to diameter ratio and two values of the diametral clearance. Presented data included bush temperature distribution, journal surface temperature and oil flow rate for different values of loads and speeds. The tests have demonstrated that large temperature variation can exist around the bush, particularly for combinations of high load and high speed. The axial temperature variation is negligible except in the vicinity of the oil hole feeding the downstream groove where the incoming oil causes local cooling of the bush. Bush circumferential temperature variation increases with both load and speed. However, while both the bush maximum temperature and journal temperature increase with speed, the effect of load is to produce temperature minima at eccentricity ratio around 0.4 to 0.6. In consequence, the operating temperature (bush maximum temperature) of a long bearing can be higher than that of a short bearing under the same light load, although at higher load the longer bearing runs cooler than the shorter bearing.

As you can see, in all of the studies including the thermohydrodynamic investigations of high speed journal bearings, Reynolds equation with some approximation has been used. On the other hand, CFD techniques are applied only to journal bearings with laminar lubricant flow. In the following, some of studies using