A Review of Fluid Film Bearing

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Abstract: In the present study, theoretical and experimental studies related to fluid film bearing design are reviewed. There is a requirement for the refinement of the existing analogies and design procedures for better understanding and accuracy in production. Several features like design procedures, incorporation of accurate boundary condition in journal bearing, thermal effects, surface roughness, use of non Newtonian lubricants and fluid inertia, use of couple stress fluids are important contributors in the design of fluid film bearings. Much more experimental observation is required to validate the existing solution and to indicate their range of applicability for fluid film bearing.

Keywords: Fluid film bearing; Thermal effect; Non-Newtonian lubricant; Newtonian fluid: Couple stress; Surface roughness effect; Fluid inertia.

1. Introduction

Fluid bearings are bearings which solely support the bearing's loads on a thin layer of liquid or gas. They can be broadly classified as fluid dynamic bearings or hydrostatic bearings. Hydrostatic bearings are externally pressurized fluid bearings, where the fluid is usually oil, water or air, and the pressurization is done by a pump. Hydrodynamic bearings rely on the high speed of the journal self-pressurizing the fluid in a wedge between the faces. Fluid bearings are frequently used in high load, high speed or high precision applications where ordinary ball bearings have short life or high noise and vibration. They are also used increasingly to reduce cost. For example, hard disk drive motor fluid bearings are both quieter and cheaper than the ball bearings they replace.

The recent trends towards high power output and high speed require a better understanding of design of fluid film bearing. Most of the work done in this field is related to bearing analysis rather than bearing design. The objective of this paper is to provide available information in the field of fluid film bearings and the subsequent scope of improvement to meet the future demand. Lubrication results introduction of low shear strength material between the interacting surfaces, which is one of the best processes to reduce friction between two surfaces in relative motion. Lubrication includes hydrodynamic, hydrostatic means of utilizing liquid and gas as lubricant. Although the concept of hydrodynamic lubrication was developed earlier, the application of this theory in design took considerable period of time. The objective of the study is to outline the scope for further refinement of fluid film bearing design using modern computational facilities and by introducing modern sophisticated measuring instruments.

2. Neutonian and non-Neutonian fluid

The fluid which obeys the rule, that shear stress is proportional to rate of shear strain is called Neutonian fluid and the other fluid is call non-Neutonian fluid.

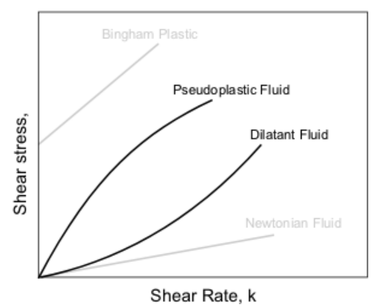


Fig 1: Neutonian and non-Neutonian fluid

3. Basic Bearing Theory

The primary equation for the fluid film bearing is, Reynold's equation. The equation appears as:

$$\partial/\partial x[dh^3(\partial d/\partial x)]+ \partial/\partial z[dh^3(\partial d/\partial z)]=6\eta U\partial/\partial x(dh)+12\eta\partial/\partial t(dh) \quad (1)$$

Equation (1) is valid for incompressible and compressible flow under dynamic condition. For steady state solution time dependant term is set to zero. The density h will remain constant for bearing using incompressible lubricant. Closed form solution for infinite long and short bearing is also available. These solutions are quite useful but later onPinkus4 and Raimondi and Boyd solved numerically the finite bearing using digital computer. A large amount of data was published for journal bearing having various L/D and eccentricity ratio.

4. Boundary Condition

The solution of journal bearing under steady state condition requires realistic boundary condition. As per Summerfield and Gambel boundary conditions were extensively used but they are not the realistic. Swift on the basis of stability argument and Stiber from the condition of flow continuity at film cavity interface provide identical solution at $\partial p/\partial x = 0$ and $d = C$avity, the cavitations boundary. The Swift-Stiber condition does not explain the sub cavity pressure that occurs just upstream from the cavity boundary. Floberg introduced the concept of gaseous cavitations assuming that all the lubricants were carried away by air fingers or saturation incurring in the cavitations region and the alternative boundary condition has been obtained. A refined boundary condition was proposed by Kicinshi1 that is more reliable under dynamic condition. In this cavitations model, the rupture and reformation boundary condition are determined by the flow in the cavitations zone boundaries. The flow is balanced by the flow in the cavitations zone and an additional flow due to boundary movement with time. Cole and Hughes made a quantitative study of the behavior of the oil film in dynamically loaded journal bearing. Leeuwestein calculated the free boundary and cavitations region using finite element method (FEM). Dowson *et al* had developed an algorithm for determination of film rupture and reformation boundaries.

Kumar and Booker had developed a finite element cavitations algorithm, where they validate the theoretical result and found that mass is not conserved always throughout the cavitating fluid film and mass conservation may not always be a determinant of journal motion.

5. Working principle of Fluid film Bearing

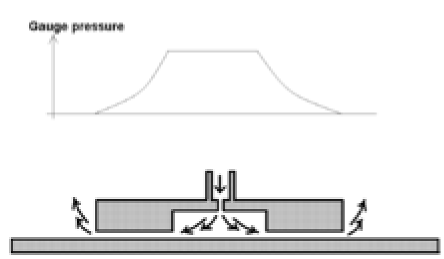


Fig 2: Operation of fluid film bearing

A hydrostatic bearing has two surfaces which has a fluid forced, via a restrictive orifice, in between the surfaces so that it keeps them apart. If the gap between the surfaces reduces then the outflow via the edges of the bearing is reduced and the pressure goes up, forcing the surfaces apart again very strongly, giving excellent control of the gap and giving low friction. Fluid bearings use a thin layer of liquid or gas fluid between the bearing faces, typically sealed around or under the rotating shaft.

There are two principal ways of getting the fluid into the bearing:

- In fluid static, hydrostatic and many gas or air bearings, the fluid is pumped in through an orifice or through a porous material.
- In fluid-dynamic bearings, the bearing rotation sucks the fluid on to the inner surface of the bearing, forming a lubricating wedge under or around the shaft.

Hydrostatic bearings rely on an external pump. The power required by that pump contributes to system energy loss just as bearing friction otherwise would. Better seals can reduce leak rates and pumping power, but may increase friction.

Hydrodynamic bearings rely on bearing motion to suck fluid into the bearing and may have high friction and short life at speeds lower than design or during starts and stops. An external pump or secondary bearing may be used for startup and shutdown to prevent damage to the hydrodynamic bearing. A secondary bearing may have high friction and short operating life, but good overall service life if bearing starts and stops are infrequent.

6. Characteristics and principles of operation

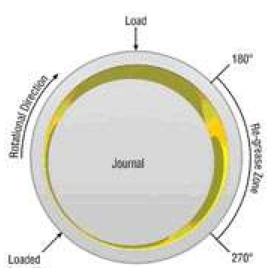
Fluid bearings can be relatively cheap compared to other bearings with a similar load rating. The bearing can be as simple as two smooth surfaces with seals to keep in the working fluid. In contrast, a conventional rolling-element bearing may require many high-precision rollers with complicated shapes. Hydrostatic and many gas bearings do have the complication and expense of external pumps.

Most fluid bearings require little or no maintenance, and have almost unlimited life. Conventional rolling-element bearings usually have shorter life and require regular maintenance. Pumped hydrostatic and aerostatic (gas) bearing designs retain low friction down to zero speed and need not suffer start/stop wear, provided the pump does not fail.

Fluid bearings generally have very low friction—far better than mechanical bearings. One source of friction in a fluid bearing is the viscosity of the fluid. Hydrostatic gas bearings are among the lowest friction bearings. However, lower fluid viscosity also typically means fluid leaks faster from the bearing surfaces, thus requiring increased power for pumps or seals.

When a roller or ball is heavily loaded, fluid bearings have clearances that change less under load (are "stiffer") than mechanical bearings. It might seem that bearing stiffness, as with maximum design load, would be a simple function of average fluid pressure and the bearing surface area. In practice, when bearing surfaces are pressed together, the fluid outflow is constricted. This significantly increases the pressure of the fluid between the bearing faces. As fluid bearing faces can be comparatively larger than rolling surfaces, even small fluid pressure differences cause large restoring forces, maintaining the gap. However, in lightly loaded bearings, such as disk drives, the typical ball bearing stiff nesses are ~10^7 MN/m. Comparable fluid bearings have stiffness of ~10^6 MN/m Because of this, some fluid bearings, particularly hydrostatic bearings, are deliberately designed to pre-load the bearing to increase the stiffness. Fluid bearings often inherently add significant damping. This helps attenuate resonances at the gyroscopic frequencies of journal bearings (sometimes called conical or rocking modes).It is very difficult to make a mechanical bearing which is atomically smooth and round; and mechanical bearings deform in high-speed operation due to centripetal force. In contrast, fluid bearings self-correct for minor imperfections. Fluid bearings are typically quieter and smoother (more consistent friction) than rolling-element bearings. For example, hard disks manufactured with fluid bearings have noise ratings for bearings/motors on the order of 20-24 dB, which is a little more than the background noise of a quiet room. Drives based on rolling-element bearings are typically at least 4 dB noisier. Tilting pad bearings are used as radial bearings for supporting and locating shafts in compressors.

7. Different types of Fluid Film Bearings



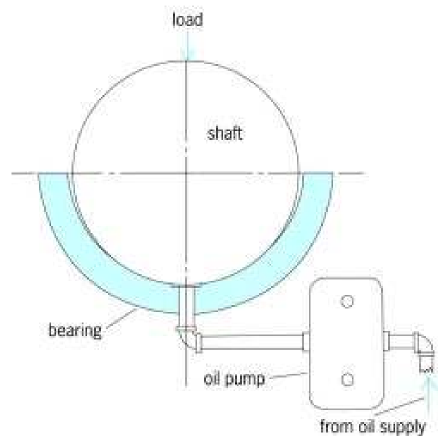


Fig 3: Fluid film bearing

A) Foil bearings

Foil bearings are a type of fluid dynamic air bearing that was introduced in high speed turbine applications in the 1960s by Garrett AiResearch. They use a gas as the working fluid, usually air and require no external pressurization system.

B) Journal bearings

Pressure-oiled journal bearings appear to be plain bearings but are arguably fluid bearings. For example, journal bearings in gasoline (petrol) and diesel engines pump oil at low pressure into a large-gap area of the bearing. As the bearing rotates, oil is carried into the working part of the bearing, where it is compressed, with oil viscosity preventing the oil's escape. As a result, the bearing hydroplanes on a layer of oil, rather than on metal-on-metal contact as it may appear.This is an example of a fluid bearing which does not use a secondary bearing for start/stop. In this application, a large part of the bearing wear occurs during start-up and shutdown, though in engine use, substantial wear is also caused by hard combustion contaminants that bridge the oil film.

C) Air bearings

Unlike contact-roller bearings, air bearings utilize a thin film of pressurized air to provide an exceedingly low friction load-bearing interface between surfaces. The two surfaces don't touch. Being non-contact, air bearings avoid the traditional bearing-related problems of friction, wear, particulates, and lubricant handling, and offer distinct advantages in precision positioning, such as lacking backlash and stiction, as well as in high-speed applications. The fluid film of the bearing is air that flows through the bearing itself to the bearing surface. The design of the air bearing is such that, although the air constantly escapes from the bearing gap, the pressure between the faces of the bearing is enough to support the working loads.

An example of a fluid bearing is ice skating. Ice skates form a hydrodynamic fluid bearing where the skate and ice are separated by a layer of water caused by entropy.

8. Design of Fluid Film Bearing

A vast amount of literature covering theory of fluid film bearing is available taking into account the Swift-Strieber boundary condition, however practicing engineers in the industry are not sufficiently familiar with this information to arrive at efficient design of many components. The increasing use of the available modern computer and graphics package can provide an efficient method for designing of fluid film bearing, which usually requires a total numerical analysis of pressure distribution and corresponding design quantity such as load capacity, load position, oil flow rate, power loss, coefficient of friction etc. Another option is to determine the design quantities from graphs or design diagram but this involves error if extrapolation is needed. So, one can avoid this by using design function describing the design quantity. Stabl and Jacobson had developed design function for hydrodynamic bearing. Jones, *et al* had discussed the development of a mathematical model for bearing performance prediction. Reason and Narang had developed a rapid design and performance technique for steady state journal bearing. Similarly Hirani, *et al* had developed a rapid globally convergent method for dynamically loaded journal bearing where an approximate analytical pressure expression is proposed and minimum and maximum film thickness is analyzed. The expertise and knowledge which are available in the field of fluid film bearing could be programmed into an expert system that would be available to the designer as and when required.

9. Thermal Effect

An important area in the fluid film lubrication is thermodynamics because the variable viscosity can affect the performance considerably. Previously the thermal effect in the fluid film bearing design were considered by representing the entire lubricant viscosity field with a single parameter called the effective viscosity, compatible with temperature rise in the bearing. Sternlicht and Maginihiss presented a detailed numerical procedure for simultaneous solution of Reynolds and energy equation. Following this work Raimondi gave performance curves for a finite slider bearing in which viscosity was allowed to vary as a function of pressure and temperature. Mccallion, *et al* provided the first complete thermo hydrodynamics solution. Safer provide a thermo hydrodynamic solution for finite journal bearing under dynamic loading condition. In this work a numerical solution for the given journal locus in the thermo hydrodynamic flow was suggested. Pinkus and Bupara had developed an adiabatic solution for finite journal bearing where the variation of viscosity with temperature was considered but the variation with pressurewas neglected. Pinkus and Wilcock studied the thermal effect in fluid film bearing and they had recommended some further research work. Mistry, *et al* had studied the thermal profile and cavitations in a circular journal bearing. Khonsari, *et al* had developed a design procedure for the thermodynamics design based on the work of Khonsari. Banwait and Chandrawat had developed a three dimensional energy and heat conduction equation for fluid film and bush temperature, and a one dimensional equation for the journal temperature. Gomiciga and Keogh has studied the orbit induced journal temperature variation in hydrodynamic bearing where computational fluid dynamics technique is used to analyze the dynamic flow and heat transport in lubricant film. Keogh and Khonsari has studied the influence of inlet condition on the thermodynamic state of a fully circumferential grooved journal bearing. Pierre, *et al* had suggested a numerical model to predict the bearing characteristics under steady loading. The model includes the film rupture and reformation phenomenon by conserving mass flow rate.

10. Non-Newtonian Lubricant

Most of the fluids in use as lubricant have a archeology that can not be described as Newtonian behavior. Such lubricant is called non-Newtonian lubricant. Additives are added to oil to improve performance but these additives change the basic properties of tribo-mechanical system such as load capacity, oil flow, film thickness, friction etc. The two non Newtonian behaviors are characterized by two non linear effects: shear thinning, where viscosity decrease with increasing shear rate and visco elasticity where the fluid has memory for its deformation history. The non-Newtonian analysis is both robust and computationally efficient, requires only 50% more CPU time than the Newtonian analysis. But in the field of hydrodynamic lubrication no significant progress has been made to find out the performance characteristics of bearing using non-Newtonian lubricants. Steady state characteristics of these bearings are available in various works. Some of these papers use viscosity as logarithmic function of shear stress and others analyzed non-Newtonian effect of shear thinning on the pressure distribution by assuming a certain shear-strain relation. Dien and Elrod had developed a regular perturbation expansion for velocity and pressure field for non-Newtonian type situation. An attempt has been made to investigate the stability characteristics of journal bearing using non-Newtonian inelastic pseudo plastic and shear thinning fluids. Buckholz and Lin had studied the effect of bearing misalignment on load and cavitations for non-Newtonian lubricant. Kacou, et al had done a thermo hydrodynamic analysis of journal bearing using non-Newtonian lubricant. In this paper the authors had found out that the temperature effect is reduced due to non linear characteristics of the fluid. Sheeja and Prabhu had done the thermo hydrodynamic analysis of journal bearing using non-Newtonian fluid. This analysis includes the thermal and non-Newtonian effect on steady state characteristics like Sommerfield number, fluid film thickness, flow rate, friction etc. Wang, *et al* had studied the non-Newtonian lubricant behavior on the performance of a dynamically loaded elliptical journal bearing. They found that there was an increase in maximum film pressure and flow rate and decrease in minimum film thickness and power due to non-Newtonian effect.

11. Surface Roughness Effect

The classical theory of hydrodynamic lubrication does not consider the surface roughness of the element having relative motion. This theory is applicable in thick film lubrication, when the load is very high and film thickness is very small, there is a probability of asperity –asperity contact. Rough surface has been modeled as a stochastic process by Chistensen41 in hydrodynamic bearing. Both one dimensional (longitudinal and transverse) and two dimensional (Isotropic) models of roughness were considered for roughness slope of about 10-12. It has been reported that surface roughness has significant effect on steady state characteristics of hydrodynamic bearing when roughness height is of the same order of magnitude as film thickness. Majumdar and Hamrock have studied the effect of roughness on finite journal oil bearing. The effect of surface roughness parameter, surface pattern, eccentricity ratio and length to diameter ratio on hydrodynamic load and side leakage was investigated. Turaga, *et al* had found that the transverse roughness increases the stability and isotropic roughness decreases the stability of a rigid rotor system supported by journal bearing with rough surface. Raja and Sinha found that incase of short bearing transverse roughness results in an increase in side leakage and the effect of roughness was more pronounced when load was reversed. Mokhtar, *et al* found that wavy journal surface under certain condition display higher load capacity, lower friction. Guhas46 had analyzed the steady state characteristics of misaligned hydrodynamic journal bearing with isotropic roughness effect. Weng and Chen combined the effect of surface roughness and flow archeology on the linear stability of a rotor supported on short length journal bearing analysis. The effect of flow rheology and roughness is significant in high eccentricity region. The appropriate control of surface roughness parameter can improve the static and dynamic performance of short journal bearing. In dealing with non-linear transient method cavitations model proposed by Kicinski was appropriately modified to take into account the surface roughness effect. This study also includes the thermal effect. Using stochastic finite element method stability margin of these bearings is found. Future work on this aspect is still required to predict the performance characteristics accurately.

12. Fluid Inertia Effect

The basic assumption in the theory of hydrodynamic lubrication includes negligible inertia forces in comparison to the viscous forces. However, the fluid inertia can not be neglected when the viscous and the inertia forces are of the same order magnitude. There are basically three methods used to estimate the inertia effect of fluid.

1. Linearization of the Navier-Stokes equation by first order perturbation in modified Reynolds number.The equation obtained by this approach is valid for only low value of modified Reynolds number.

2. A modified form of Reynolds equation derived from the full set of Navier-Stokes equation adopting an iteration method.

3. It has been found that the velocity profile is not affected much and it remains parabolic even if inertia terms are included. Based on this a set of equation are derived by Constantinescu. Many investigations were made to study the effect of fluid inertia, however there is a still little information about the dynamic characteristics of the journal bearing. The choice of boundary condition plays a significant role when inertia effect is included. Mori and Mori and Hashimoto observed that the result could vary by using different boundary condition while inertia effects are included. Kakoty and Majumdar had studied the effect of fluid inertia on stability of flexibly supported oil journal bearing; here the linear perturbation technique is used. The movement of the cavitations boundary with time makes it necessary to consider the time history of the fluid film shape in

dynamic analysis. Kakoty and Majumdar had studied the effect of fluid inertia on stability of oil journal bearing.

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