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PRESSURE DISTRIBUTION AND FRICTION COEFFICIENT OF HYDRODYNAMIC JOURNAL BEARING

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ABSTRACT

The research approach of this paper is.A journal bearing supports the load in radial direction and which is working on hydrodynamic lubrication.Lubricated journal bearings present a common challenge for construction equipment manufacturers in the world. The common design methodology is based on actual data and has worked very well historically because the industries and governments have accepted that bearings in construction equipment need frequent lubrication and exchange of worn parts. These requirements call for better methods of designing lubricated journal bearings.The aim of the carried work is to develop data for better design methods for lubricated journal bearing design used in heavy duty construction equipment machines, in order to prolong life and lubrication intervals.

Keywords:* *Friction coefficient, Hydrodynamic Lubrication, Journal bearings, Pressuregeneration, Sliding Surfaces

I INTRODUCTION

During the past fewyears hydrodynamic journal bearings have got great attention from analytical and practical engineers. Due to wide range of engineering application such as pumps, compressors, turbines, nuclear reactors, precision machine tools, the growth of journal bearing technology increases in considerable amount. Fluid film lubrication is hydrodynamic phenomenon characterized by a lubricant flowing in narrow gap between closely spaced surfaces; it avoids solid to solid contact and thus reduces friction and wear in significant amount which results in increases component life. “Hydrodynamic lubrication is defined as system of lubrication in which load supporting fluid film is created by shape and relative motion of the sliding surfaces”. The main aim of the study was to determine the oil film pressure in hydrodynamic journal bearings carrying realistic loads. Therefore, the study also included the determination of the operating range, friction lossand oil film temperature and oil film thickness.

The theory of hydrodynamic lubrication is based on a differential equation derived by Osborne Reynold. This equation is based on a fallowing assumption,

1. The lubricant obeys Newton’s law of viscosity.

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2. The lubricant is incompressible.

3. Oil film inertia forces should be negligible.

4. Viscosity of lubricant should be constant.

5. The shaft and bearing are rigid and there is continuous supply of lubricant.

Reynolds Equation is,

$$\frac{\partial}{\partial x}\left[h^3\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial z}\left[h^3\frac{\partial p}{\partial z}\right]=6\mu U\left(\frac{\partial h}{\partial x}\right)$$

Where:

h – local oil film thickness,

η – dynamic viscosity of oil,

p – local oil film pressure,

U – linear velocity of journal,

x - circumferential direction.

z - Longitudinal direction.

Theoretically, exact solution of above equation can be obtained if the bearing is assumed to be either infinitely long or very short. These two solutions are called as Sommerfield's solution.

2.1 Sommerfeld's Solution

The equation is solved with the assumption that there is no lubricant flow in the axial direction (infinitely long bearing assumption).

$$S=\left(\frac{r}{c}\right)^2\frac{\mu\omega}{p}=\left(\frac{r}{c}\right)^2\frac{\mu\omega LD}{W}$$

Where:

S is the Sommerfeld Number or bearing characteristic number

r is the shaft radius

c is the radial clearance

μ is the absolute viscosity of the lubricant

N is the speed of the rotating shaft in rev/s

P is the load per unit of projected bearing area

2.2 Balance Equation

$$\frac{\partial p}{\partial x}=u\frac{\partial^2 u}{\partial y^2} \qquad (1.1)$$

$$\frac{\partial p}{\partial z}=\mu\frac{\partial^2 w}{\partial y^2} \qquad (1.2)$$

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2.3 Flow Velocity Equation

$$\mathbf{u} = -\frac{1}{2\mu}\frac{\partial p}{\partial x}y(h-y) + \left[\left(1-\frac{y}{h}\right)u_1 + \frac{y}{h}u_2\right] \quad (1.3)$$

$$\mathbf{u} = -\frac{1}{2\mu}\frac{\partial p}{\partial z}y(h-y) + \left[\left(1-\frac{y}{h}\right)w_1 + \frac{y}{h}w_2\right] \quad (1.4)$$

In equation 1.3 for the flow velocity u in x direction, the latter half of the right hand side (in bracket) shows fluid velocity due to movement of solid surfaces relative to each other in x direction. It changes linearly as shown in Figure 1. This is called as shear or Couette flow.

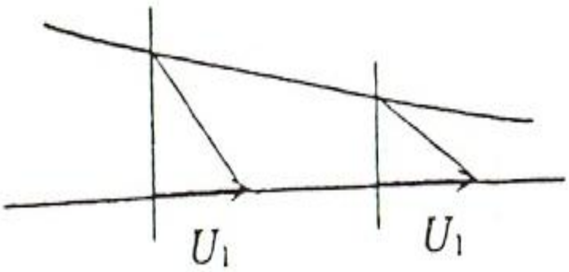

Figure 1 Shear or Couette Flow.

The former half of right hand side shows the flow velocity due to pressure gradient. It is proportional to pressure and changes parabolicallyacross the film thickness as shown in Figure 2.This is called as Pressure or Poiseuille flow.

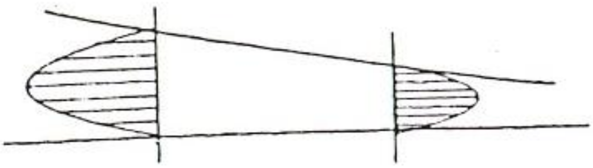

Figure 2 Pressure or Poiseuille Flow

The flow velocity is in general case is the sum of above two. Figure 2 shows such an example in which flow in the reverse direction to shearing direction occurs due to pressure gradient at the left end and shear flow is accelerated by negative pressure gradient at the right end. At the point of maximum pressure $\partial p/\partial x=0$, and therefore the flow at that point consists of shear flow only.

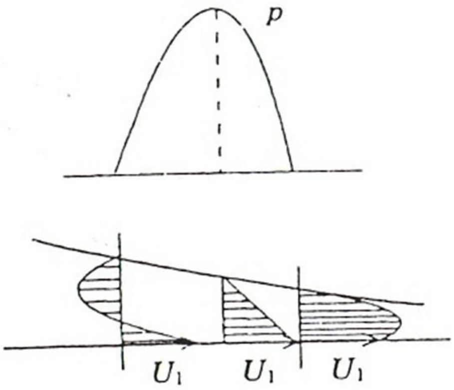

Figure 3 Pressure Distributions along Sliding Direction

III LITERATURE REVIEW

1.Miss. KirteeL.Chidle - Oil pressure and temperature in bearing depends on various factor such as bearing geometry, properies of fluid , rotational speed and force developed.

2.AmitSingla- Pressure and temperature increases markedly with increase in journal speed. Due to the geometry of the bearing under study two different lobes has been observed in both pressure and temperature distribution.

3. Steve Pickering -Successful solutions include enhanced materials and lubrication additives that form layers on the contacting surfaces with superior frictional and wear resistance than the parent material. Improvement in wear resistance is an ongoing process and questions remain open regarding the viability of environmentally friendly lubrication fluids.

4. PriyankaTiwari-The stress types, magnitudes and distributions in journal bearings are affected by the geometrical and physical properties of the bearing system. Brinkman model (BM) significant improves the lubrication performance of long, flexible, porous journal bearings in comparison with Darcy model.

5.ChetanMehara--In this paper the preliminary study of pressure profilesaround a journalbearing under hydrodynamic lubrication were described and compared withtheoreticalprofiles obtained from Raimondi and Boyd charts.

IV MECHANISM OF PRESSURE GENERATION

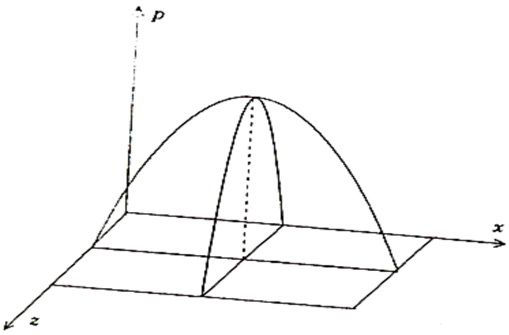

Figure 4 -Pressure Generation

Reynolds pressure equation is given by,

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)-\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\mu\left[u_1\frac{\partial h}{\partial x}+h\frac{\partial u_1}{\partial x}+2v_2\right]$$

First left hand side indicates approximately the average curvature of the pressure distribution surface as shown in Figure 4. If the left hand side is negative it means that the pressure distribution is upward convex, or pressure generated is positive. Second right hand side represents the causes of pressure generation and the three terms correspond to the following three mechanisms of pressure generation respectively.

a. The first term represents the ‘Wedge effect’:- Pressure generation is due to fluid being driven from thick end to thin end of wedge shaped fluid film by the surface movement.

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b. The second term represents 'Stretch effect':- Pressure generation is due to the variation of the surface velocity from place to place.

c. The third term represents 'Squeeze effect':- Pressure generation is due to the variation of surface gap (film thickness).

V THE OPERATION OF HYDRODYNAMIC JOURNAL BEARINGS

Lubrication reduces friction between two surfaces (such as sliding surfaces of a bearing and a shaft) in relative motion. It is typically categorized as boundary, mixed and hydrodynamic lubrication, for example by Heywood (1988), Becker (2004) and Gleghorn and Bonassar (2008). When a journal bearing operates under boundary lubrication, the sliding surfaces of the bearing and shaft are practically in direct contact and friction is at its highest level. Lower friction levels are achieved through the use of mixed lubrication, where the sliding surfaces are partially separated by the lubricant, and of hydrodynamic lubrication, where the sliding surfaces are completely separated by the lubricant. To illustrate how friction varies under different lubrication conditions, Stribeck curves (or diagrams) have been used widely in different engineering sciences. In Stribeck curves, the friction coefficient is presented as a function of a dimensionless parameter calculated from the dynamic viscosity; angular speed and pressure (see Figure 5). The above-mentioned parameter is typically called the duty parameter or Hersey number. The minimum of the friction coefficient is reached at the critical value of the duty parameter, at the dividing line between the mixed and hydrodynamic lubrication zones. Heywood (1988) presented a Stribeck curve for a journal bearing. Methods for the calculation of Stribeck curves were studied by de Kraker*et al*. (2007). They calculated the friction coefficient as a function of the journal frequency at different values of the projected bearing pressure. A hydrodynamic journal bearing is designed to operate normally under hydrodynamic lubrication, where hydrodynamic pressure (see Figure 6) in the lubricant keeps the sliding surfaces of the bearing and shaft apart from each other. By sliding motion, hydrodynamic pressure is created.

Figure indicate that if we are operating to the right of line and something happens say, an increase in lubricant temperature, this cause reduction in viscosity and hence a smaller value of μN/P which lead to reduction in friction but same time heat is generated in shearing the lubricant, and consequently the lubricant temperature drops.

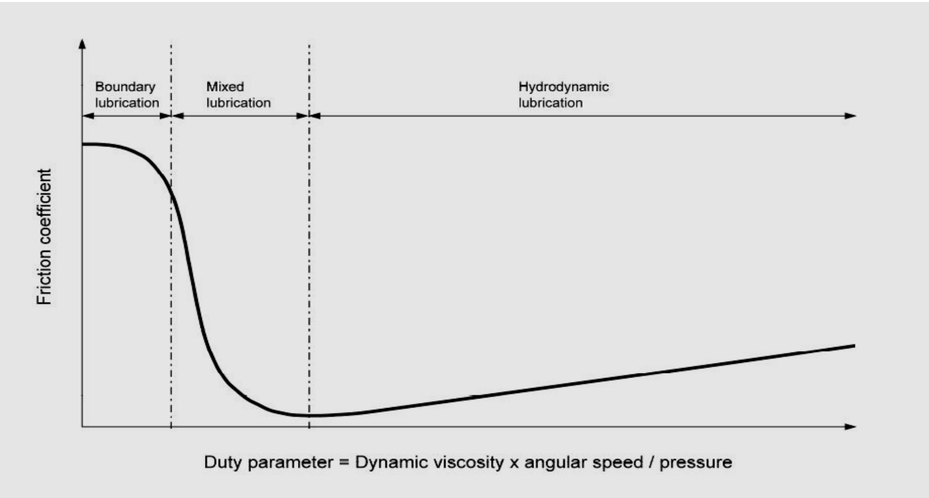

Figure- 5 Stribeck's Curve

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Curveshowing the friction coefficient as a function of the dutyparameter under boundary, mixed and hydrodynamic lubrication**.**

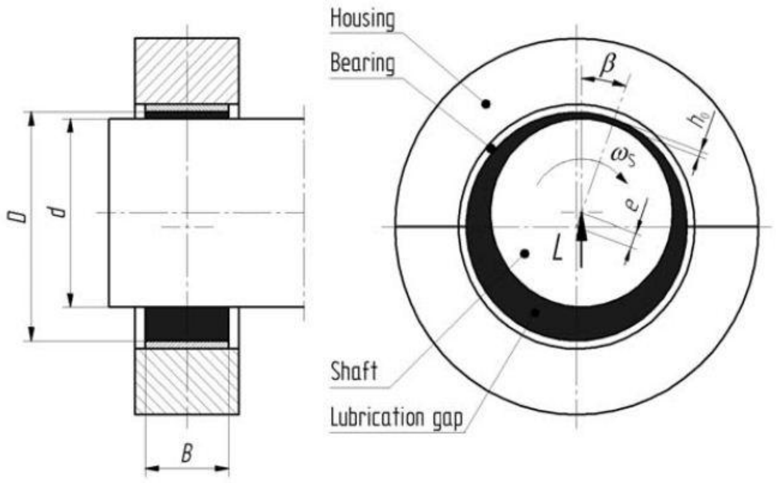

Figure 6- Hydrodynamic Journal Bearing

Hydrodynamic journal bearing with the diameter *D* and the width *B* carrying the Bearing load *L*.Distribution of the hydrodynamic pressure *p* in the oil film on the sliding surface of a hydrodynamic journal bearing with the diameter *D* and the width *B* carrying the bearing load *L*.The shaft with the diameter *d* runs at the angular velocity S.

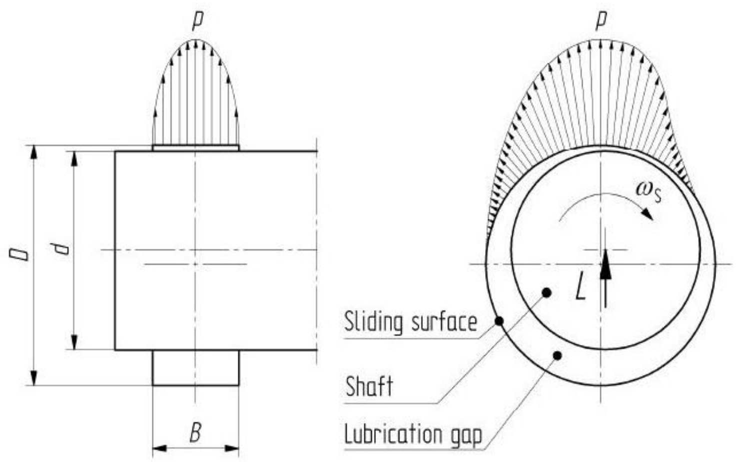

Figure 7- Pressure Distribution

Hydrodynamic journal bearings are simple but critical components and numerous parameters influence their operation. Knowledge for research into bearings can be obtained in test rig experiments and by mathematical means. There are advantages and disadvantages to each of the above mentioned ways to collect information about bearings.

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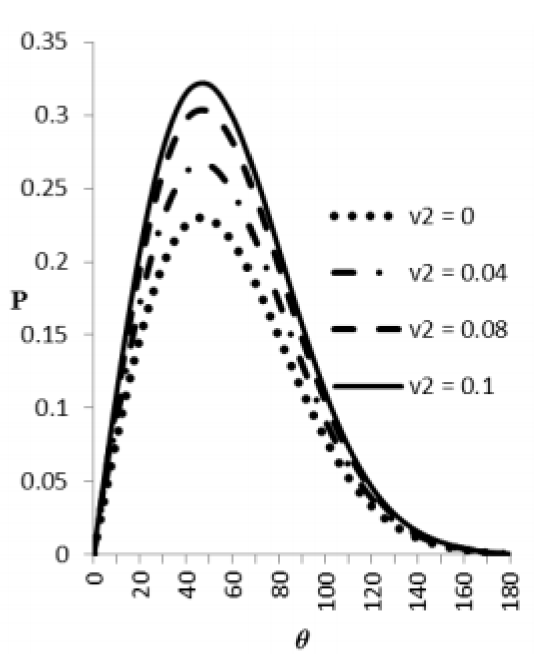

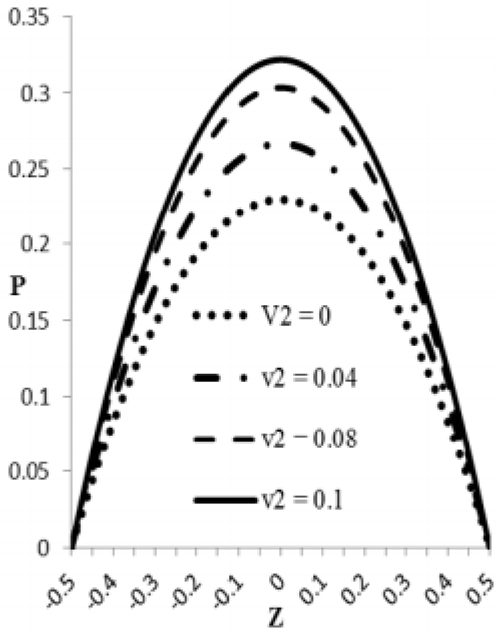

Figure- 8 pressure distribution curve Figure- 9 P vs Z under various volume fraction

A pair of volume fraction of additives blended with the base lubricants have been used with different viscosity ratio. Dimensionless pressure distribution over the lubricant film has been identified. Fig. 8 and fig. 9 show the relationship between dimensionless pressure (P) versus circumferential coordinate (θ) and dimensionless coordinate in the z-direction (Z) respectively.

VI CONCLUSION

Hydrodynamic journal bearings are commonly used throughout numerous industries as a way to support a rotating shaft. The frictional contact causes wear that can lead to unwanted performance degradation or failure which ultimately results in costly maintenance and repair. In Hydrodynamic bearings metal to metal contact occurs at beginning, which results in higher starting friction. However, under running conditions when full hydrodynamic film has developed, the power losses due to friction are less. The frictional coefficient increases with the degree of misalignment at lower values of the eccentricity ratio.Hydrodynamic journal bearings are simple but critical components and numerous parameters influence their operation. Pressure generation is due to fluid being driven from thick end to thin end of wedge shaped fluid film by the surface movement, the variation of the surface velocity from place to place and the variation of surface gap (film thickness).

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