International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 3 Issue 7, July - 2014

Analysis of Hydrodynamic Journal Bearing Using CFD and FSI Technique

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Abstract—Hydrodynamic journal bearings are analyzed by using Computational fluid dynamics (CFD) and fluid structure interaction (FSI) approach in order to find Pressure profile and temperature distribution in the bearing structure, satisfying the boundary conditions. The Journal bearing is designed in ANSYS software, the journal is modeled as a ''moving wall'' With an absolute rotational speed of 3000rpm and bearing is modeled as a "stationary wall". Design parameters like pressure distribution and temperature distribution are considered for the analysis. It is assumed that the flow of lubricant is laminar and steady. Also cavitations effects in the bearing are neglected by setting all negative pressures to ambient pressures. Design data like journal diameter, clearance, L/D ratio, minimum film thickness, journal speed and oil viscosity are taking by machine design data book for making analytical calculation. The CFD results were compared in order to validate the model with the analytical results and good agreements were found.

Keywords—. Fluid Structure Interaction, Static Pressure distribution, temperature distribution.

Nomenclatures—

D = journal diameter
D_b = bearing diameter
C = clearance
h_{min} = minimum film thickness
h_{max} = maximum film thickness
N = journal speed
L = length of journal
η = oil viscosity
v_m= peripheral velocity of journal
ϵ = Eccentricity ratio
W = safe working load
ϕ = Attitude angle
ΔT = temperature rise
ψ = relative clearance

1. INTRODUCTION

Hydrodynamic type journal bearings are considered to be a vital component of all rotating machinery whose function is to support an applied load by reducing friction between the relatively moving surfaces. A journal bearing consists of a circular shaft, called the journal, is made to rotate in a fixed sleeve is called the bearing. The bearing and the journal operates with a small radial clearance of the order of $1/1000^{th}$ of the journal radius. The clearance space between the journal and the bearing is assumed to be full of the lubricant. The radial load squeezes out the oil from the journal and bearing face and metal-to-metal contact is established. When the journal begins to rotate inside the bearing, it will climb the bearing surface and as journal speed is further increased; it will force the fluid into the wedge-shaped region. Since more and more fluid is forced into the wedge-shaped clearance space, which begins to exert pressure with increasing journal speed. At a particular speed, the pressure becomes enough to support the load and the closest approach between journal and bearing where the oil film thickness is the minimum. A condition of perfect lubrication will exit when minimum oil film thickness is greater than the quantity dependent on the nature of the irregularities of the contacting surfaces. The value of minimum oil film thickness, the angle between the line of center with the vertical is called the attitude angle and the location of the maximum film pressure is important considerations in journal bearing lubrication [10]. Load carrying capacity of journal bearing is dependent on pressure in layer of lubricant during rotation of shaft. Hence, it is necessary to analyze the fluid film of lubricant using the capabilities of commercial CFD code incorporating the technique of Fluid Structure Interaction (FSI). The pressure field for a full journal bearing operating under laminar flow regime with L/D = 1.5 ratio is obtained by CFD, satisfying the boundary conditions. The results show reasonable agreement in general.

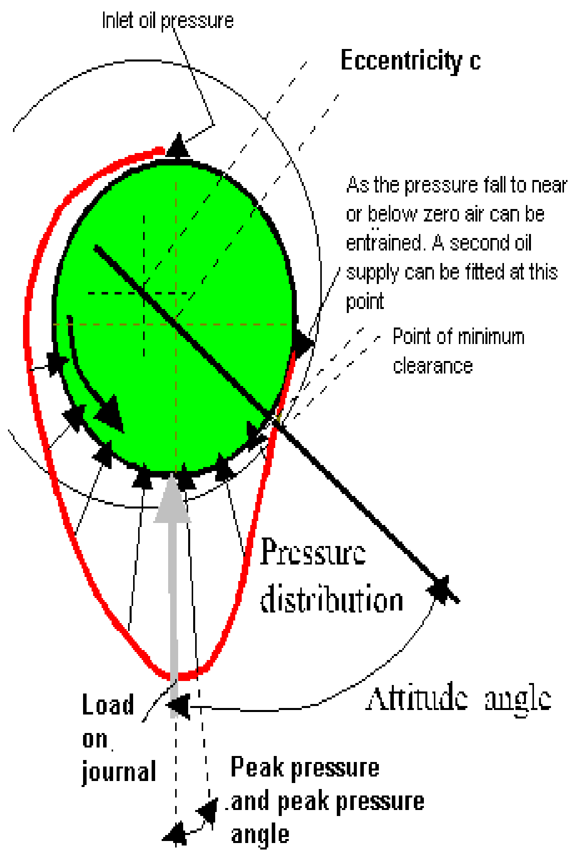

Figure 1. Hydrodynamic journal bearing

2. LITERATURE REVIEW

The performance characteristics of journal bearing are investigated by means of three-dimensional computational fluid dynamics analysis. The three dimensional Navier Stokes compressible equations were integrated to simulate the flow. Turbulence effects were considered in the computation of unsteady transient analysis of journal bearing, taking into account gravity. The Journal bearing is designed in Gambit software. The journal is modeled as a ''moving wall'' with rotational speed of 3000rpm. The flow is simulated by using ANSYS Fluent software. Design parameters like relative eccentricity, dimensionless load carrying capacity, dimensionless wall shear stress, Reynolds number, Sommerfeld number, friction coefficient, strain rate, pressure distribution, temperature distribution and lubricant flow properties like turbulent viscosity, and velocity magnitude are considered for the analysis. It is assumed that the flow of lubricant is laminar as well as isothermal. Unsteady transient analysis is carried out for the journal bearing with various L/D ratios of 0.25, 0.5, 1, 1.5, and 2 and the corresponding results: relative eccentricity vs. Sommerfeld number, Dimensionless load carrying capacity vs. relative eccentricity, and dimensionless friction coefficient vs. relative eccentricity are presented in this journal [7]. COMSOL Multiphysics 4.3a software is used for 3D model of hydrodynamic plain journal bearing and pressure distribution in plain journal bearing is obtained by steady state analysis. Generalized Reynolds equation is used for analyzing hydrodynamic journal bearing by COMSOL as well as by analytical method by applying Sommerfeld boundary conditions. This Reynolds equation is applied for two theories of hydrodynamic journal bearing called infinitely short journal bearing and infinitely long journal bearing. Results for pressure distribution obtained by COMSOL simulation are compared with analytical results shows that the solutions are approximately similar to the analytical solutions[9].The performance characteristics and the core formation of a hydrodynamic journal bearing lubricated with a Bingham fluid are derived by means of three-dimensional computational fluid dynamics analysis. The Navier–Stokes equations are solved using the FLUENT. Three-dimensional computational fluid dynamics model are found to be in very good agreement with experimental and analytical data from previous investigations on Bingham fluids. The validated Computational Fluid Dynamics (CFD) model is used to extract a series of diagrams in the form of the Raimondi and Boyd graphs and can be use in the smart bearing design [8]. The thermoelastohydrodynamic study for analysis of elliptical journal bearing (Two-lobe) operating with Newtonian lubricant has been presented and thermoelastic deformations of the solid parts are taken into account. To solve the Reynold's equation generalized form, equation of energy and the displacement field, respectively, using two numerical techniques Computational Fluid Dynamic (CFD) and Fluid Structure Interaction (FSI). The CFD is used to determine the pressure, temperature and velocity fields in the lubricant film and the FSI simulation is used to obtain the stress intensity and displacement field. The effect of the operating conditions on the fields' pressure, temperature, displacement and stress intensity is also analyzed [2]. Hydrodynamic journal bearings are analyzed by using CFD and FSI approach in order to find deformation of the bearing. Journal bearing models are developed for different speeds and eccentricity ratios to study the interaction between the fluid and elastic behavior of the bearing. Cavitation effects in the bearing are neglected by setting all negative pressures to ambient pressures. The CFD results were compared in order to validate the model with the experimental work and a good agreement was found. It is observed that CFD-FSI method provides a useful platform to study the elastohydrodynamic behavior of the bearing. It is observed that the bearing deformations are significant and should be considered in order to predict accurate performance of the hydrodynamic journal bearings [5]. A comparative study of pressure distribution and load capacity of a cylindrical bore journal bearing is presented by using finite element method and analytical method. In this calculation the isothermal analysis and Newtonian fluid film behavior were considered. The analytical results and finite element results were compared in order to validate the work and these results were also compared with the available published results. Finally it is realized that the finite element results showed better agreement than analytical results [4]. The structure of lubricant film is modified by using double layer of lubricant in to clearance space of bearing surfaces in place of single layer of lubricant. The composite-film bearing combines the advantages of high-viscosity with the low-viscosity lubricant. The low-viscosity lubricant will be to reduce viscous dissipation, while the high-viscosity lubricants maintain the desirable thickness to separate out the bearing surfaces. The basic Reynolds equation is used for composite films under the restrictive assumptions by applying boundary conditions. On comparing the performance of four bearings, which are lubricated, respectively, by a homogeneous film of ISO50 oil, a composite film of ISO130 oil+water, a composite film of ISO500 oil+water, and a composite film of ISO1000 oil+water with identical dimensions and the operating parameters of bearing. Composite-film bearings have considerably lower frictional losses in comparison to traditional bearings [1].

3. ANALYTICAL CALCULATION

Analytical calculation is made by using design data book; we collect the design data for journal bearing (given in Table 1) then we used different formulas [6] for calculating safe maximum pressure, safe operating load and temperature rise.

3.1 Bearing pressure

General electric company's formula:-

$P_a = 6.2*10^5 \sqrt[3]{V}_m$

We know

$V_m = \pi$ D N/60

V_m = 21.99 m/s

Then

$P_a = 1.74*10^6$ N/m^2 or 1.74 MPa

[Average pressure]

Victor Tatarinoff's equation:-

$$P = 13.5 \frac{\eta\, n'}{\psi^2}\left\{\frac{L}{L+D}\right\}$$

P = 4.76MPa [safe maximum pressure]

H.F. Moore's equation for critical pressure:-

$P_c = 7.23 * 10^5 \sqrt{V_m}$

P_c = 3.4MPa

3.2 Safe oil film thickness

$h_{min} = 2.37*10^{-5} V_m^{\ 0.4} A^{0.2}$

h_{min} = 0.05067mm or 50.67μm

3.3 Eccentricity

Now we know that

$h_{min} = C_r (1 - \epsilon)$

$\epsilon = \frac{e}{c_r}$ so **eccentricity e** = $2*10^{-4}$ m or 200μm

3.4 Attitude angle

Attitude angle

$$\phi = \mathrm{Tan}^{-1}\left\{\frac{\pi\sqrt{1-\epsilon^2}}{4\epsilon}\right\}$$

$\phi = 30.5^0$

$$\boldsymbol{\theta_{max}}\ \textbf{at}\ \boldsymbol{P_{max}} = \mathrm{Cos}^{-1}\left\{\frac{1-\sqrt{1+24\epsilon^2}}{4\epsilon}\right\} = 162^0$$

3.5 Safe operating load

Victor Tatarinoff's equation for safe operating load W =

$$\frac{\eta\, n'\, D^3 \left(\frac{L}{D}\right)^2}{0.295\ \mathrm{hmin}\ \psi\left(1+\frac{L}{D}\right)}$$

Hence W = 351.60 KN

3.6 Temperature rise

The temperature rise of the lubricant film is due to heat generated which is to be carried away by the lubricant, can be found

$$\Delta T = \frac{8.3*10^{-6}\mu P}{Q/D^2\, n'L}$$

Where Q is [Volume of film that is π /4($D_b^{\ 2}$ - D^2)x length] per second

So Q = $2.31*10^{-5}$ m^3/s

P = 10826.87N/m^2

250N load of shaft for bearing portion as we have not decided any load initially i.e.

[π /4*D^2*L*density (7850kg/m^3)] = 25kg approx.

Or 250N

$$\Delta T = \frac{8.3*10^{-6}*0.15*10826.87}{2.31*10^{-5}/(140*10^{-3})^2 *3000/60*210*10^{-3}}$$

$$\Delta T = \frac{0.0135}{1.122*10^{-4}} = 120.08967 \text{ in degree Kelvin}$$

Table 1: journal bearing dimensions and oil Properties. [6]

Journal diameter (D)	140mm
Bearing diameter (D_b)	140.5mm
Radial clearance (C)	500μm
Length to diameter ratio (L/D)	1.5
Operating speed (N)	3000 rpm
Lubricant Density	7850 kg/m^3
Lubricant Viscosity (μ)	0.15 N-s/m^2
Minimum film thickness (h_{min})	50 μm

4. CFD MODEL – ANALYSIS

The model is constituted as one cylinder with a diameter D of 140 mm and another one with a diameter of 140.5 mm, with eccentricity of $2*10^{-4}$ m or 200μm. The model is designed with the help of AUTOCAD and then import on ANSYS for Meshing and analysis. The analysis by CFD-FSI approach is used in order to calculating pressure profile and temperature distribution.

For meshing, the fluid ring is divided into two connected volumes. Then all thickness edges are meshed with 360 intervals. A tetrahedral structure mesh is used. So the total number of elements is 7290579. The load is calculated to be 250N weight of shaft for bearing portion as we have not decided any load initially.

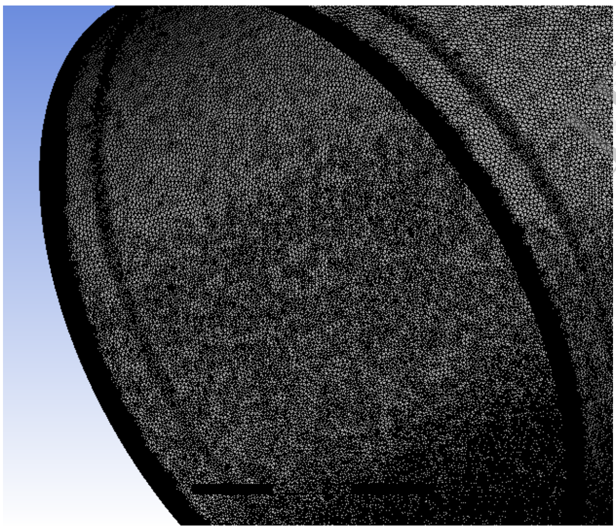

Fig 2. Meshed Model

The journal is modeled as a ''moving wall'' With an absolute rotational speed of 3000rpm and bearing is modeled as a "stationary wall" with no slip condition. Pressure p=0 at L=0 and p=0 at L=210

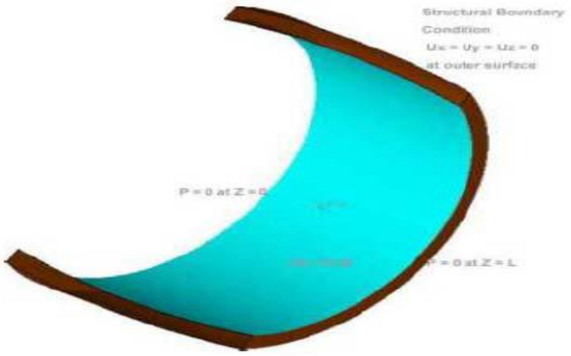

Figure 3: Boundary conditions. **[2]**

4. RESULTS AND DISCUSSION

The steady state analysis of journal bearings has been carried out for the case of infinitely long journal bearing (L/D= 1.5) at eccentricity ratios of 0.8. Pressure distribution have been determined by using ANSYS software and compared with analytical results. The results obtained have shown for eccentricity ratio of 0.8 in table 2. After simulation pressure distribution on journal surface has been found out as contour representation. The maximum pressure is reached in a region closer to the minimum film thickness and negative pressure results due to appropriate boundary conditions. The pressure contours and pressure distribution are shown in fig.4 and fig.6 respectively. Pressure profile also generated on polar plot in fig.8. The temperature contours and temperature distribution are shown in fig.5 and fig.7 respectively. Results for pressure distribution and temperature rise obtained by CFD are compared with analytical solution and found that they are approximately matching.

Table 2: Comparison of Analytical results and CFD results

Parameters	Analytical Results	CFD Results
Maximum Pressure	4.76MPa	4.383MPa
Temperature Rise	120.08967 in degree Kelvin	412 in degree Kelvin (Max.)

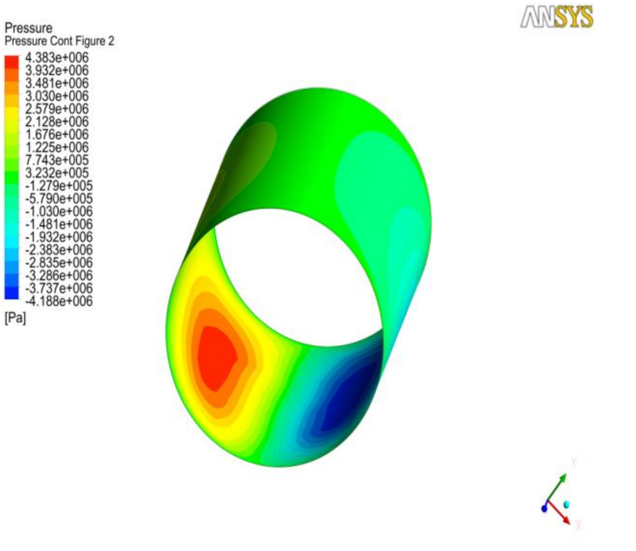

Figure 4. Pressure distribution for L/D ratio 1.5 at eccentricity ratio, ε = 0.8.

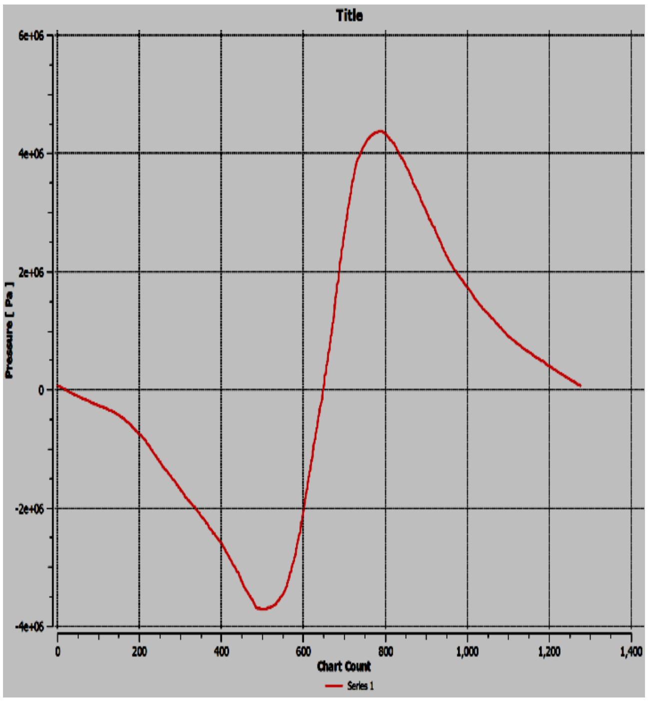

Figure 6. Pressure distribution for L/D ratio 1.5 at eccentricity ratio, ε = 0.8.

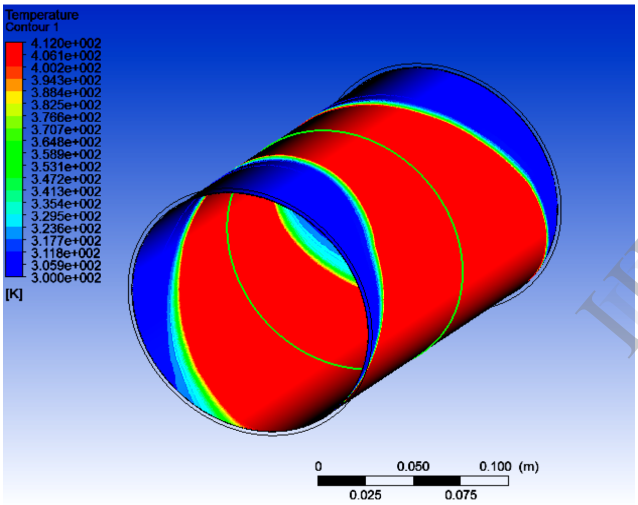

figure 5. Temperature distribution for L/D ratio at eccentricity ratio, ε = 0.8.

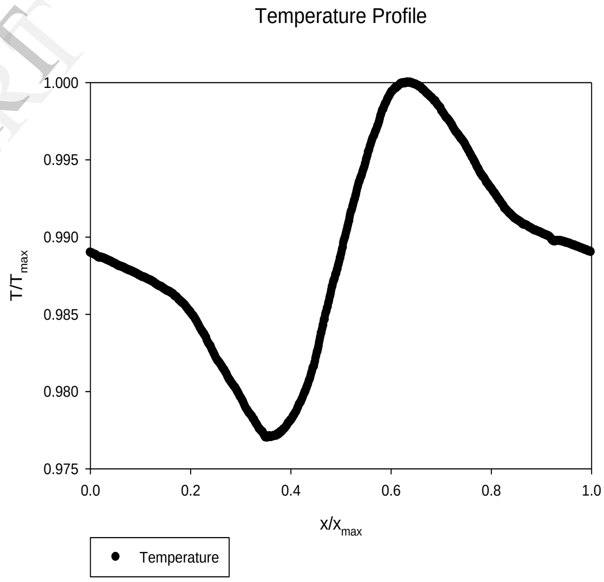

Figure 7. Temperature distribution for L/D ratio 1.5 at eccentricity ratio, ε = 0.5.

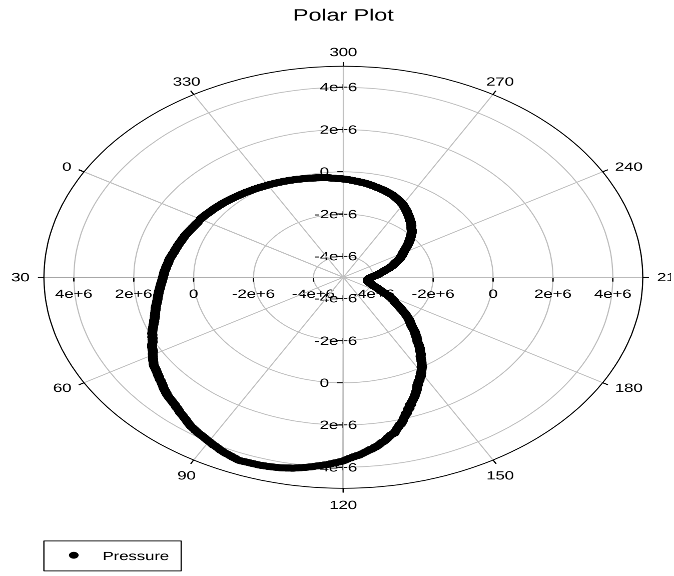

Figure 8. Polar plot: Pressure **vs**. Angle

5. CONCLUSION

The pressure distribution of the hydrodynamic journal bearing lubricated with oil under steady state consideration has been analyzed. Based on the results and discussion presented in the preceding part, following conclusions can be made for journal bearing studied.

Design data like journal diameter, clearance, L/D ratio, minimum film thickness, journal speed and oil viscosity are taking by machine design data book for making analytical calculation for hydrodynamic journal bearing.

Using different formulas analytical model is developed for infinitely long journal bearing to find steady state characteristics of journal bearing. Furthermore, this analytical model is implemented on ANSYS software particularly on CFD and FSI for more advanced analysis. Using CFD pressure distribution for journal bearing is simulated and compared with analytical solution. It is found that CFD gives approximately identical solution for infinitely long (L/D=1.5) journal bearing, hence CFD solution get validated with analytical solution.

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