Numerical Unsteady Analysis of Thin Film Lubricated Journal Bearing

K. M. Panday, P. L. Choudhury, and N. P. Kumar

***Abstract*—The performance characteristics of a thin film lubricated journal bearing are investigated by means of three-dimensional computational fluid dynamics analysis. The 3D Navier Stokes compressible equations were integrated to simulate the flow. Turbulence effects were included 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 an absolute rotational speed of 3000rpm. The flow is simulated using Ansys Fluent software. Design parameters like relative eccentricity, dimensionless load carrying capacity, dimensionless wall shear stress, friction coefficient, Reynolds number, Sommerfeld number, 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 and isothermal. Unsteady transient analysis is carried out for the journal bearing with different 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 the analysis.**

***Index Terms*—Eccentricity, journal bearing, load carrying capacity, turbulent dissipation rate, unsteady, wall shear stress.**

I. Introduction

A bearing is a system of machine elements whose function is to support an applied load by reducing friction between the relatively moving surfaces. As modern science and technology is developing, one discovered that the lubricant is affected by the gap of bearing. The interaction degree between the lubricant and solid surface influences the lubrication property of the bearing clearly. A thin film lubricated journal bearing is to develop positive pressure by virtue of relative motion of two surfaces separated by a fluid film. Thin film bearings are those in which although lubricant is present, the working surfaces partially contact each other at least part of the time. These bearings are also called Boundary Lubricated Bearings. Recently, following the progress in computer technology, many researchers began to use commercial computational fluid dynamics (CFD) programs in their investigations. The main advantage of CFD code is that it uses the full Navier–Stokes equations and provides a solution to the flow problem, whereas finite difference codes are based on the Reynolds equation. The results obtained by the two approaches are therefore likely to differ. Moreover, the CFD packages are applicable in very complex geometries.

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II. Literature Review

K. P. Gertzos, P. G. Nikolakopoulos and C. A. Papadopoulos [1] worked on CFD analysis of journal bearing hydrodynamic lubrication by Bingham lubricant. They study the performance characteristics and the core formation in a hydrodynamic journal bearing lubricated with a Bingham fluid. They solved Navier–Stokes equations using the FLUENT package and compared the results of the developed 3-D CFD model with theoretical and experimental results of previous investigations, for both Newtonian and Bingham lubricants, and found to be in very good agreement. The volume that a core occupies is greater for a larger value of L/D and T_0 for a specific relative eccentricity. A core is formed at a specific relative eccentricity and adheres to the bearing surface at the inlet side. The core "moves" to the outlet side for greater values of relative eccentricity. At a high value of relative eccentricity, a core is formed and adheres to a small region of the journal. As the value of eccentricity increases, the solid on the bearing separates into two or three parts and a floating core between these parts is observed. The load carrying capacity, the film pressure, and the frictional force of a Bingham solid are larger than those of a Newtonian fluid and they increase as the yield stress T_0 increases. For low of eccentricity ratios, the effect of yield stress T_0 on the journal behavior is small. In ER and MR fluids, the yield stress increases by varying the electric and magnetic field consequently. Therefore, semi-active control of the journal is possible by varying these quantities. The above CFD-validated model should be used for such an investigation.

F. Stefani and A. Rebora [2] worked on steadily loaded journal bearings: Quasi-3D mass–energy-conserving analysis they developed a finite-element approach to thermo elasto hydrodynamic lubrication analysis by extending a previous mass- and energy-conserving algorithm to include wall-convection boundary conditions, groove-mixing theory, and thermo-mechanical deformations. To this end, the cross-film-averaged energy equation is coupled with the heat conduction equations relevant to the bearing sleeve and the journal by fitting the temperature profile across the film thickness with a fourth-order polynomial. A finite-element condensation technique is used to reduce the unknowns in heat conduction equations in the bush and in the journal to the temperatures of the sleeve surface and journal axis, respectively. The proposed method yields temperature predictions in good agreement with the experimental results, whenever consistent boundary conditions are imposed. The heat conduction between the lubricant in the supply ducts and the metal may be important for a realistic evaluation of sleeve temperatures near the feed grooves, and should be included in the bush thermal model, i.e. by adopting proper boundary conditions. A suitable thermal model of the journal may be chosen on the basis of the general arrangement of the rig. In some cases, results may depend on the effective conductive area of the grooves, which must also be reasonably chosen. Taking into account the thermal deformations of the kinematic pair enables us to improve the reliability of the model and to avoid having to make hypotheses on the actual clearance in working conditions.

K. Gururajan and J. Prakash [3] worked on Roughness effects in a narrow porous journal bearing with arbitrary porous wall thickness. They analyze the effect of surface roughness in a narrow porous journal bearing. An exact solution valid for an arbitrary porous wall thickness is presented. A comparison with earlier approximate solution, based on an assumption pertaining to thin-walled porous bearings, is made to determine the range of operating parameters for which the approximate solution is acceptable from an engineering viewpoint. The exact analysis shows that the approximate solutions are generally acceptable for low to moderate values of permeability and porous wall thickness. It is shown that for a smooth case, the results based on an approximate analysis can safely be used for thin-walled bearings of low permeability; the relative error being < 1.5% for wall thickness ratio H_0/R 0.2 and permeability parameter Ψ 0.01. The error increases to nearly 8% for values of H_0/R up to 0.8. However, for large values of Ψ (>0.1) and large, the approximate solution tends to underestimate the results, and the error increases up to 57% when Ψ=1 and H_0/R=0.8; for such values of Ψ and H_0/R, exact analysis must be used. For the rough case, the same guidelines are applicable as the error is not significantly affected by variations in C. For the slip case the error in the approximate solutions are marginally lower than that for the no-slip case.

Yu-cheng PENG, Xi-yang CHEN, Ke-wei ZHANG and Guo-xiang HOU [4] worked on Numerical research on water guide bearing of hydro-generator unit using finite volume method With the consideration of the geometry of tilting pad journal bearing, a new form of the Reynolds equation was derived in this article. The film thickness, the squeeze motion of the journal and the rotation motion of the pad were explicitly contained in the equation. Based on this equation, together with the equilibrium equation of pad pivot, the water guide bearing used in the Gezhouba 10 F hydro-generator unit was numerically researched. The new Reynolds equation for the lubricating film was solved using Finite Volume (FV) discretization; Successive Over-Relaxation (SOR) iteration method and C++ code are included. According to the numerical solution, and the stability of the film and the influences of the film thickness, the journal squeeze effect and the pad rotation effect on film force were discusses The results indicate that the squeeze effect cannot be neglected, although the rotation effect is negligible for both low-speed and high-speed bearings, so the computing time could be greatly reduced.

Radfordand D. Fitzgeorge [5] The effects of journal lobing on the performance of a hydrodynamic plain journal bearing Accurately ground journals nominally of 44.25 mm diameter and 63.5 mm wide were produced. One was cylindrical and the others had three, four, five, six, nine or twelve equi-spaced lobes, 0.025 mm high, on their peripheries. Each journal was fitted into a cylindrical lead-bronze bearing liner with a radial working clearance of 0.05 mm and operated at speeds of up to 1000 rev min−1 and at bearing loads of up to 10 kN whilst the oil pressure was maintained at or just above atmospheric pressure. The test results indicated that when a comparison is made between the lobed journals and the cylindrical journal. There are no significant differences in the load-carrying capacity of the bearing and In the higher portion of the loading range the boundary lubrication conditions are improved when using three- or four-lobed journals whilst the presence of lobes on the other journals produces only a slightly adverse effect, The effect of lobing, which often occurs accidentally during the manufacture of plain cylindrical journals, is not detrimental to the performance of the bearing when operated within the range of speeds and loads used.

Basri and D.T. Gethin [6] worked on A comparative study of the thermal behavior of profile bore bearings This paper describes principally a theoretical analysis of the thermal behavior of offset half, orthogonally displaced, three-lobe and four-lobe bearing geometries. The thermal analysis illustrates the implication of type selection with regard to the parameters of load-carrying ability, power loss, lubricant requirements and operating temperatures. The comparisons show that for all profiles considered, they have inferior load-carrying ability when compared with the cylindrical geometry along with significantly larger lubricant supply requirements. Thermal effects in profile bore bearings are less extreme than those encountered in the cylindrical profile. T.S.R. Murthy, Y. Balaramaiah and V.C. Venkatesh [7] worked on An Analysis of a Special Hydrodynamic Bearing for Machine Tool Spindles. The paper deals with the analysis of a special hydrodynamic bearing incorporated in a precision cylindrical grinding machine developed at CMTI. The special bearing is an improvement of the earlier conical multilobe bearing developed at CMTI. The improvement effected is particularly in automatic preloading through hydrodynamic thrust bearing where the preload increases with increase in speed and thereby improving the performance of the bearing over a wide range of speeds. The initial axial clearances in the thrust bearing, taper on the thrust surface, the lobe depth in the conical bearing are the main controlling parameters'.

Al-Bender and K. Smets [8] worked on Development of Externally Pressurized Foil bearings. This paper describes the development of a new type of externally pressurized partial journal foil bearing, giving an overview of design theory, fabrication and experimental validation. The bearing consists of a thin metal foil, equipped with feeding hole(s), which enfolds the shaft, and forms an effective, low cost alternative for (rigid) cylindrical bearings, whose application is hampered by technical as well as economical limitations. The basic design problematic appears to be to make the right trade-off between foil stiffness, load capacity, and total stiffness of the system. The latter is the resultant of the foil and air-gap stiffness, in series. Several prototype bearings have been built and tested for their load capacity, air consumption and stiffness characteristics. The first two are far superior to conventional air bearings; the last is comparable to rigid bearings of the same dimensions. Measurements on a prototype bearing, (50 mm, diameter, 40 mm wide, 6 bar(g) supply pressure), shows a load capacity of more than 800 N, a radial stiffness of 8.5 N/micron, and air consumption of around 2 Normal-liter/min.

Anjani Kumar, S. S. Mishra [9] worked on Steady state analysis of noncircular worn journal bearings in nonlaminar lubrication regimes The steady state behavior of non-circular worn journal bearings is analyzed for various wear depth parameters (δ_0), following Constantinescu's turbulent lubrication theory. Computed results are compared with published results. It is observed that geometric change caused by wear has a significant effect on the steady state characteristics of bearings.

U. Singh, L. Roy, and M. Sahu [10] worked on Steady-state thermo-hydrodynamic analysis of cylindrical fluid film journal bearing with an axial groove A steady-state thermo hydrodynamic analysis of an axial groove journal bearings in which oil is supplied at constant pressure is performed theoretically. Thermo hydrodynamic analysis requires simultaneous solution of Reynolds equation, energy equation and heat conduction equations in the bush and the shaft. The temperature gradient across and along the fluid film is important. Results show that the oil–bush interface temperature drops slightly in the vicinity of the inlet followed by a rapid rise in the circumferential direction and a decrease in the cavitations region. The temperature gradients in the cross-film direction were found to be much greater than those of the circumferential direction, supporting the validity of the parabolic formulation of the energy equation. Heat recalculates from the hottest point to the groove area in the fluid due to the convection, in the bush due to conduction and in the shaft due to shaft rotation. The role of the supply groove geometry on the performance of the bearing cannot be ignored. The groove angle of 36° and groove length of half of the total length of bearing promoted a reduction in the maximum temperature and increase in the maximum hydrodynamic pressure. Increasing oil supply pressure (lowering bearing number) causes a decrease in bearing operating temperatures, which is more significant for low loads. The influence of shaft speed has also been investigated. Increasing the shaft speed (increasing bearing number) resulted in increased load carrying capacity, bush temperature, flow rate and friction variable. A bearing having smaller groove angles gives a higher load capacity; this is due to higher pressures in the larger land region. As the flow rate shows an increase in magnitude with eccentricity and speed, it appears that this bearing allows the removal of heat more efficiently than the pain journal bearing. It can be concluded that THD analysis presents more realistic operating characteristics for a single axial grooved journal bearing.

S. Basri, D.T. Gethin [11] worked on axially profiled circular bearings and their potential application in high speed lubrication. The isothermal and thermal characteristics of an axially profiled circular bore bearing are determined using the finite element technique. These are compared with the predicted performance characteristic of a cylindrical bearing operating over the same range of conditions. The results obtained lead to the conclusion that the present analysis may be used to obtain general design data for an axially profiled bore bearing operating at high sliding speed. Axial profiling reduces the bulk operating film temperature by a small amount and it decreases the load carrying ability of the film and frictional losses also.

L. Roy [12] worked on Steady state thermo-hydrodynamic analysis and its comparison at five different feeding locations of an axial grooved oil journal bearing is obtained theoretically. Reynolds equation is solved simultaneously along with the energy equation and heat conduction equation in bush and shaft. From parametric study it is found that 12° feeding groove position is better in comparison to other feeding locations. Feeding from the bottom is very less preferable since the load capacity is lesser and temperature development is more. It is very difficult to obtain the solution due to numeric instability when the bearing operates at higher eccentricity ratio.

J. D. Knight ; L. E. Barrett [13]worked on An Approximate Solution Technique for Multilobe Journal Bearings Including Thermal Effects, with Comparison to Experiment . An approximate solution method for multilobe journal bearings that includes thermal effect is presented. The method is based on the assumption of an axial pressure distribution in the form of a polynomial, which allows the solution of Reynolds' equation by a one-dimensional finite element routine with little loss in accuracy from two-dimensional methods. A first order form of the energy equation for an adiabatic film is used to predict temperatures within the bearing, and the viscosity is determined as an exponential function of temperature. Comparison of solutions obtained by the variable viscosity method to effective viscosity solutions after Lund and Thomsen illustrates discrepancies in operating eccentricity and stiffness coefficients between the two approaches. Good correlation was obtained between the variable viscosity solutions and experimental measurements reported by Tonnesen and Hansen of eccentricity, pressures, and temperatures in a two-axial groove bearing.

III. Analysis

The coordinate system and the geometry of a journal bearing are shown in Fig. 1. The journal rotates with an angular velocity ω and is in an equilibrium position under the external vertical load was well as the pressure of the lubricant film. The journal axis Oj is at distance e from the bearing axis Ob. The film thickness h(y) varies from its maximum value h0 at bearing angle y ¼ 0 to its minimum value h1 at y ¼ p.

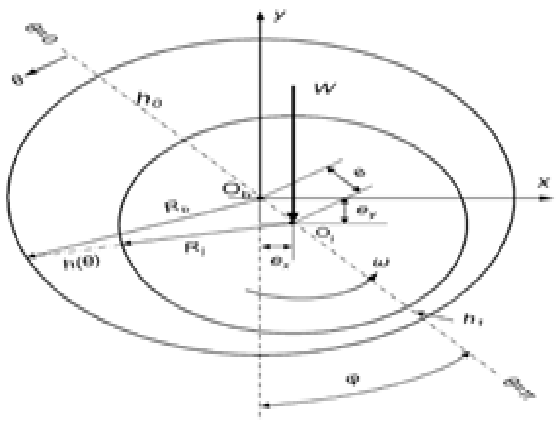


Fig. 1. Schematic diagram of Journal Bearing

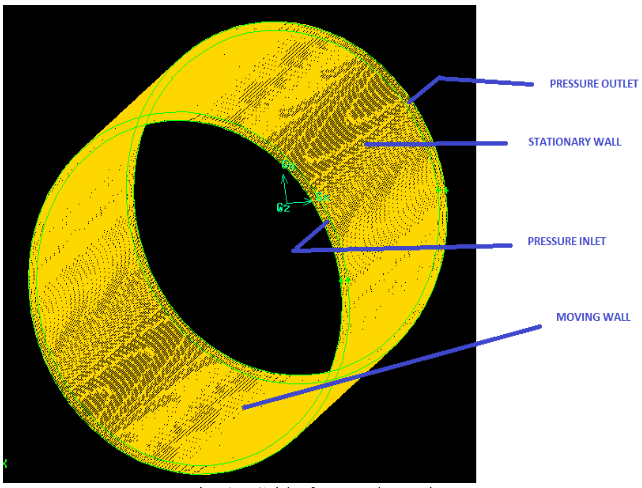


Fig. 2. Grid of Journal Bearing

A. *Equations to be Solved*

Mass conservation equation

The equation for conservation of mass, or the continuity equation, can be written as

$$\frac{d\rho}{dt} + \nabla.(\rho\vec{\vartheta}) = 0 \tag{1}$$

Eq. (1) is the general form of the mass conservation equation and is valid for incompressible as well as compressible flows.

Momentum conservation equations

Conservation of momentum in an inertial (non-accelerating) reference frame is described by

$$\frac{\partial}{\partial t}(\rho\vec{\vartheta}) + \nabla.(\rho.\vec{\vartheta}.\vec{\vartheta}) = -\nabla p + \nabla.(\bar{\bar{\tau}}) + \rho\vec{g} + \vec{F} \tag{2}$$

where $\rho\vec{g}$ and $\vec{F}$ are the gravitational body force and external body forces, respectively. The stress tensor $\bar{\bar{\tau}}$ is given by

$$\bar{\bar{\tau}} = \mu_a\left[\left(\nabla\vec{\vartheta} + (\nabla\vec{\vartheta})^T\right) - \frac{2}{3}.\nabla.\vec{\vartheta}\bar{I}\right] \tag{3}$$

where the second term on the right hand side is the effect of volume dilation.

All investigations that are referenced in this work solve the fundamental Reynolds Eq. (4), or a modified form of (4) that takes into account the equations describing the non-Newtonian lubricant,

$$\nabla.\left(\frac{h^3}{12\mu}.\nabla p\right) = \nabla.(h\bar{U}) - V \tag{4}$$

where U is the velocity of journal surface parallel to the film and V the squeeze-film velocity. When the film pressure drops down to the atmospheric pressure due to heavy external load or high operating rotational speed, rupture of the film or cavitations occurs. Taking into account this situation, the following boundary condition is used which applies the non-sub-atmospheric pressure constraint:

$$p - p_a \geq 0 \; at \; \pi \leq \theta \leq 2\pi, z = 0, z = L \tag{5}$$

The Reynolds boundary condition that makes the pressure curve to drop in parallel with y-axis just after 1801 could be used instead of Half Sommerfeld BC. The Reynolds boundary condition gives in some cases more accurate results than the half Sommerfeld BC. Although it is relatively accurate, the Reynolds BC is still an approximation to the transition from full fluid flow to flow with cavitations. The FLUENT can handle cavitation flows, but cannot use Reynolds BC. This happens because the Navier Stokes equations are solved instead of Reynolds equation. It is very difficult to modify the Navier Stokes equations so as to include the Reynolds BC. More work has to be done in a following investigation where the cavitation should be taken into consideration. When cavitation is included, the vaporization pressure, the surface tension coefficient and the non-condensable gas mass fraction are also needed.

Viscosity for non-Newtonian fluids

For incompressible Newtonian fluids, the shear stress is proportional to the rate-of-deformation tensor D:

$$\bar{\bar{\tau}} = \mu\bar{\bar{D}} \tag{6}$$

where $\bar{\bar{D}}$ is defined by,

$$\bar{\bar{D}} = \left(\frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j}\right) \tag{7}$$

and μ is the viscosity, which is independent of D. For some non- Newtonian fluids, the shear stress can similarly be written in terms of a non-Newtonian viscosityμ_a:

$$\bar{\bar{\tau}} = \mu_a(\bar{\bar{D}}).\bar{\bar{D}} \tag{8}$$

In general, μ_a is a function of all three invariants of the rate-of deformation tensor $\bar{\bar{D}}$. However, in the non-Newtonian models available in FLUENT, μ_a is considered to be only a function of the shear rate $\dot{\gamma}$ that is related to the second invariant of $\bar{\bar{D}}$ and is defined as,

$$\dot{\gamma} = \sqrt{\frac{1}{2}.\bar{\bar{D}}:\bar{\bar{D}}} \tag{9}$$

where the operand: denotes the double dot (or inner–outer) product.

IV. CFD Model-Analysis

A 3-D simulation model is developed using the CFD package FLUENT 6.3. The pre-processor Gambit 2.3 is used for the grid generation. The individuality of the problem examined here (the clearance size is very small compared to journal diameter and length) enforces the use of only hexahedral cells because the use of tetrahedral cells leads to an enormous number of cells. Eight divisions were used across the journal-bearing film, 360 divisions were used in the circumferential direction, and 25 divisions were used in the axial direction. The number of cells is 72000. One hundred and twenty divisions were used in the circumferential direction initially, giving unacceptable results for the angle of the maximum pressure.

The dimensions of the journal bearing used in this simulation are: diameter 0.05m and clearance 235 mm. The C/D value is 4.7×10^3, denoting a clearance in the upper limit of acceptable. The C/D value is 1×10^3, denoting a clearance in the lower limit of acceptable values. The results of the two above cases are almost identical, denoting that the results are independent of the value of clearance. The viscosity is 0.2 Pa s while the density is 960 kgm^3. As the Re number is 0.35 (for rotational speed 3000 rpm), the viscous model is set to laminar.

The Navier–Stokes equations are solved in steady state, taking into account gravity forces. The operating pressure is set to 101325 Pa. To simplify the geometry, one side of the clearance is used as a lubricant inlet and the other as an outlet. The boundary conditions are: ‘‘pressure outlet’’ with gauge pressure at zero Pascal and ‘‘pressure inlet’’ with an appropriate value leading to the right-side lubricant flow rate. The bearing shell is modeled as a ‘‘stationary wall’’. The journal is modeled as a ‘‘moving wall’’ with an absolute rotational speed of 300 rpm. The rotational axis origin is set to the value of eccentricity. When the problem is solved for a constant eccentricity, this value is known. When the problem is solved for a constant external force, the final position of the origin is computed automatically by the ‘‘dynamic mesh’’ technique. The ‘‘dynamic mesh’’ model in FLUENT is used to model flows where the shape of the fluid domain is changing due to motion on the domain boundaries.

V. Results and Discussion

Transient response of a dynamically loaded thin film lubricated journal bearing at unsteady condition is carried out. The results obtained for a bearing with the following parameters are presented here: journal diameter D = 50 mm; Radial clearance= 0.025mm; journal speed n = = 3000 rpm; viscosity of lubricant μ = 1.06 Pa-sec with constant load acting on the journal is 25MPa.

The transient variation of oil film thickness and oil pressure are studied.

A. Pressure Analysis

Static pressure and Total pressure variation on the walls of the bearing is presented in the Fig. 3-7 and the path lines of pressure variations with respective to time is presented in Fig. 8. The maximum Static and Total pressure are given in the Table I.

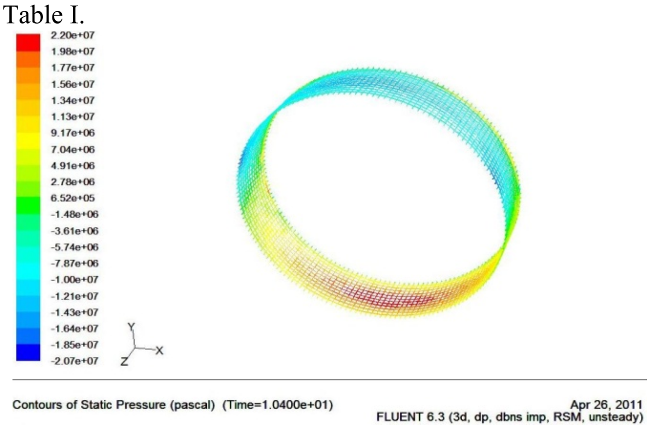


Fig. 3. Countors of Static pressure for L\D=0.25

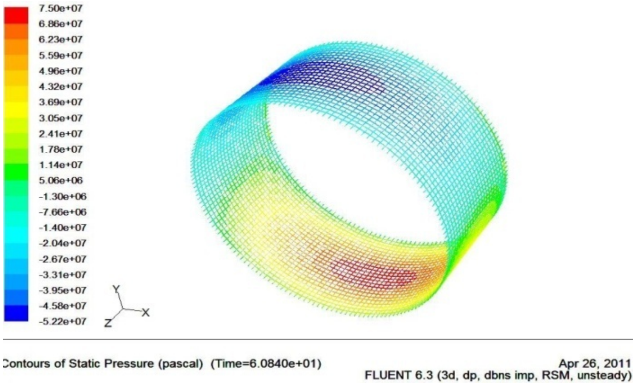


Fig. 4. Countors of Static pressure for L\D=0.5

TABLE I: Maximum Static and Total Pressures

L/D ratio	Max. Static Pressure (Pa)	Max. Total Pressure (Pa)
0.25	2.195×10^7	4.252×10^7
0.5	7.50×10^7	7.54×10^7
1	2.057×10^8	2.057×10^8
1.5	2.840×10^8	2.844×10^8
2	3.506×10^8	3.506×10^8

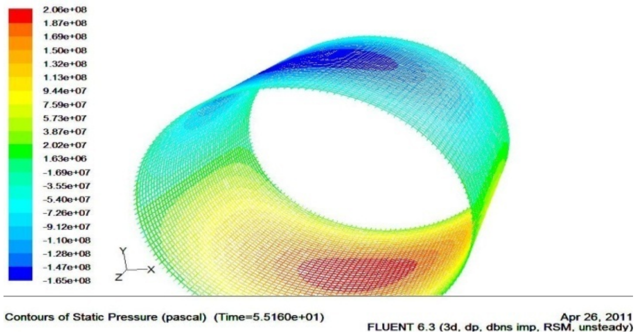


Fig. 5. Countors of Static pressure for L\D=1

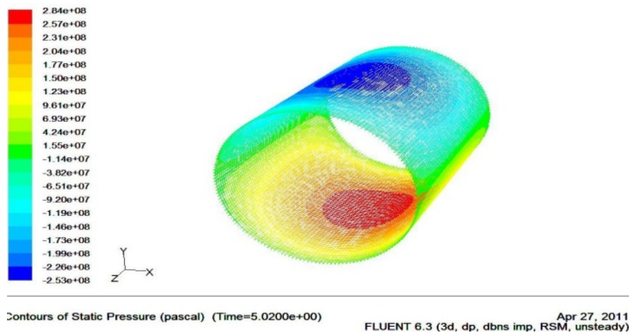


Fig. 6. Countors of Static pressure for L\D=1.5

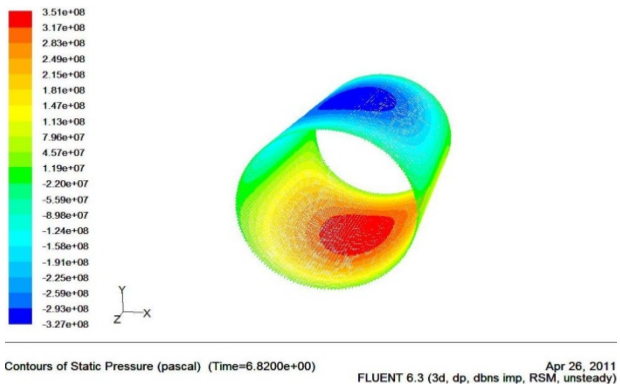


Fig. 7. Countors of Static pressure for L\D=2

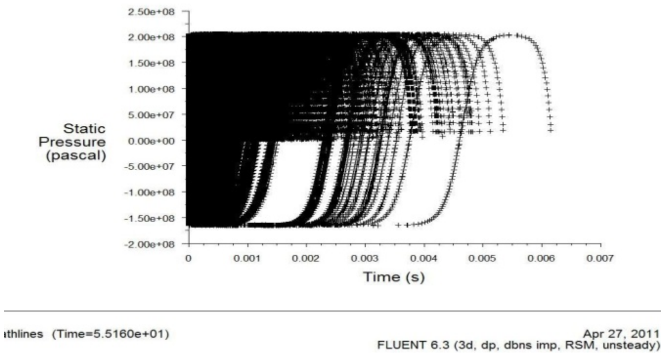


Fig. 8. Path lines of Static pressure

B. Wall Shear Stress

Shear stress developed on journal and bearing walls are presented in the table below. The Fig. 9 shows the shear stress distribution across the walls of the bearing with L/D ratio of 1. And the path lines of wall shear stress variation are presented in the Fig. 10. From the path lines we can observe that after 0.04 seconds the shear stress is steady. The maximum and minimum wall shear stress for all the L/D ratios are presented in Table II.

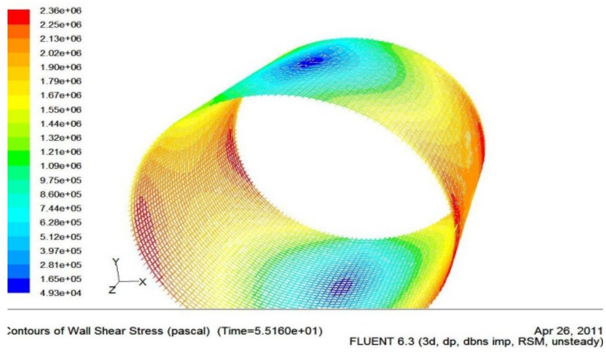


Fig. 9. Countors of Wall Shear Stress for L/D=1

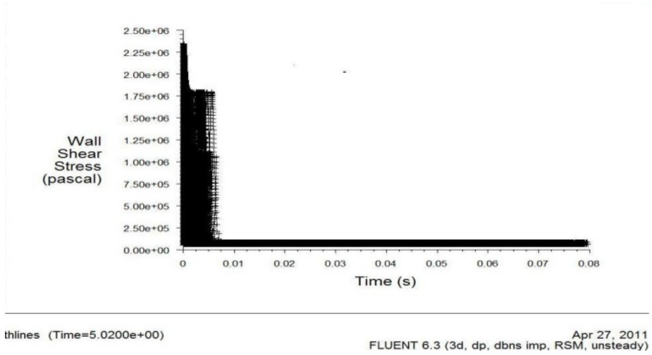


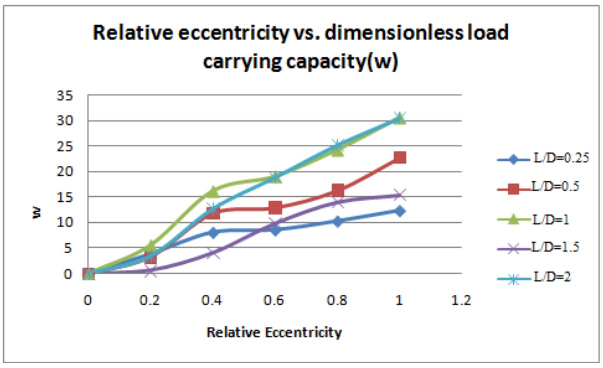
Fig. 10. Path lines of wall shear stress

TABLE II: SHEAR STRESS VARIATIONS FOR DIFFERENT L/D RATIOS

L/D ratio	Min. Wall shear stress (Pa)	Max. Wall shear stress (Pa)
.025	51843.33	2489254
.05	54218.66	2423201
1	49343.11	2364375
1.5	47613.75	2348063
2	40331.52	2335389

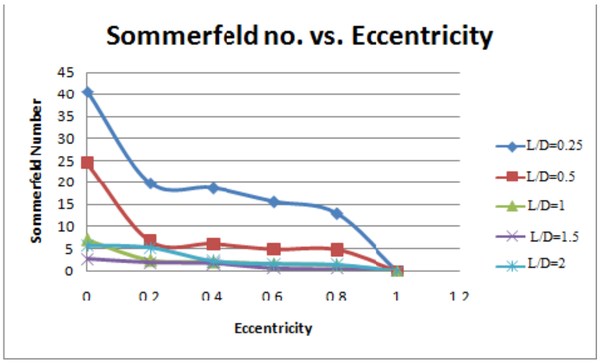
C. Load Carrying Capacity

The dimensionless load carrying capacity of the bearing has been found out for different L/D ratios. These results are presented for different eccentricity values in the graph.



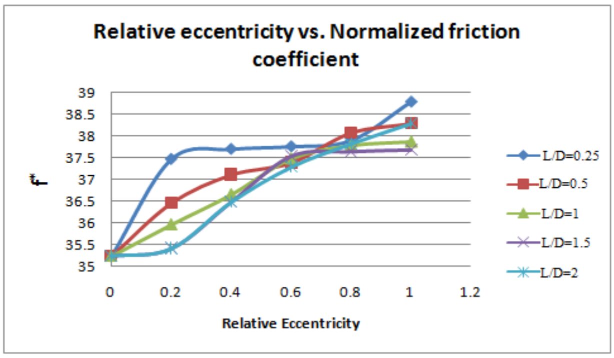
D. Bearing Characteristics Number

Sommerfeld number which is the main variable in design of journal bearing and its values are found out for different L/D ratios at eccentricity varying from 0 to 1 and presented in the graph below.



E. Normalized Friction Coefficient

The graph is plotted for Normalized friction coefficient and relative eccentricity for different L/D ratios. The variation of normalized friction coefficient is from a minimum value of 37.5 to 38.88.



F. Turbulence Viscosity

Turbulent viscosity variation of the lubricant is presented in the Fig. The maximum and minimum turbulent viscosity variations for different L/D ratios are presented in the Table III.

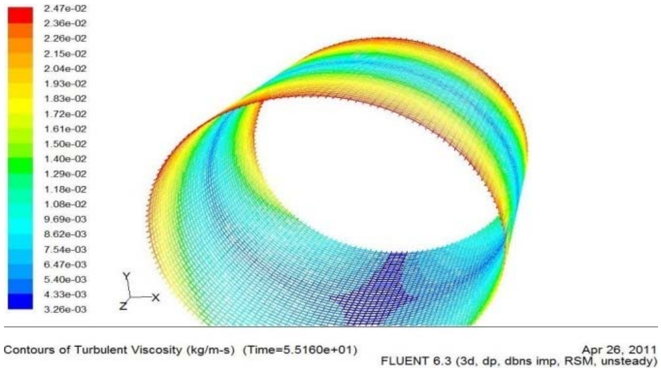


Fig. 11. Countors of Turbulent viscosity

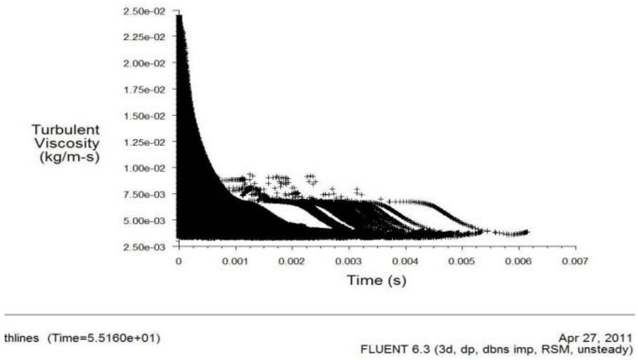


Fig. 12. Path lines of Turbulent viscosity

TABLE III: MIN. AND MAX. TURBULENT VISCOSITY

L/D ratio	Min. Turbulent Viscosity (kg-m/s)	Max. Turbulent Viscosity (kg-m/s)
0.25	0.00565	0.0235
0.5	0.0047	0.02422
1	0.00325	0.02469
1.5	0.00173	0.0247
2	0.00087	0.002484

VI. CONCLUSION

Transient dynamic behavior of thin film lubricated journal bearing system have been studied and presented. From pressure plots, it is observed that the maximum pressure, the bearing can withstand is increasing with increase in L/D ratio. The maximum pressure is noted at minimum oil film thickness. It can be observed that the shear stress developed on the walls of journal and bearing is steady after 0.04seconds and it is decreasing with increase in L/D ratio. The dimensionless load carrying capacity is observed to be good with L/D ratio 1. Turbulent viscosity of the lubricant increases with increase in L/D ratio and the level of turbulence at the minimum oil film thickness is low.

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