Comparative Study of Different Analytical Method To Reynolds Equation with Experimental Method To Find out Maximum Pressure of Journal Bearing

Wakchaure Keshav Gulab
PG Student Mechanical Department.
BM Collage Of Technology,
Indore , India

Ravi Kumar Dwarpalli
Asst prof-Mechanical Department
BM Collage Of Technology,
Indore , India

***Abstract*— A comparative study of analytical method and Experimental method for pressure distribution in journal bearing is presented in this paper. In calculating the pressure distribution. Using both analytical method and Experimental method, pressure distribution in the bearing was calculated. Moreover, the effects of variations in operating variables such as eccentricity ratio, shaft speed and different load condition for different grade oil of the bearing were calculated. The analytical results and Experimental result were compared. In order to check the validity, these results were also compared with the available published results. IN comparison with the published results, generally rapid method, Analytical Method results showed better agreement than experimental results.**

Keywords-Journal Bearingt,Reynolds Equtions,Analytical Method,Experimental Method.

I. INTRODUCTION

In today's world we are highly dependent upon the mechanical devices or machines for our daily works. The operation of these mechanical systems involves the relative motion of some machine elements. To protect the wear and tear of these relatively moving surfaces under various types of external loadings and to have better and smooth operation, we use bearings, which is the heart of every mechanical system/device or machine. Fluid-film bearings play a key role in the design of turbo machinery systems . They are important components of turbines, compressors, and pumps that are widely used in aircraft, naval ship as well as petrochemical, power and petroleum industries. Because of wide application of bearing, continuous efforts are being carried out to improve the bearing mechanisms to have the proper relative motion of the parts by finding the suitably designed bearing with proper lubrication to get desired performance level.

The past conventional hydrostatic journal bearing were restricted to support heavy load at zero or low speed and also at low eccentricity required for precision machine. Also limitation of poor performance of hydrodynamic bearings at low speed compelled the designers to improve the performance of hydrostatic journal bearings.To develop alternative configurations in order to meet expanding industrial demands at different speeds. Therefore the hybrid journal bearings have been developed and used successfully in machines, which operates under high speed and heavy load conditions. A hybrid bearing combines the physical mechanisms of both hydrodynamic and hydrostatic bearings. An advantage of hybrid bearing over purely hydrostatic bearings is the ability to tolerate substantial loads over and above the normal design load. Also these bearings have advantages for withstanding heavy dynamic loadings which vary widely in the direction of rotation. In the plain hybrid bearings the recesses in the bearing surfaces are avoided in order to maximize the hydrodynamic effect.

The mobility method is an established means to predict the performance of fluid-film journal bearings subjected to dynamic loads and kinematics Although limited to circumferential symmetry, rigid surfaces, and journal-sleeve angular alignment, its relative simplicity allows the bearing designer a rapid means to provide first-order assessments of bearing performance without the need to write or run sophisticated finite element programs. For similar reasons, the mobility method is often used as the basis for bearing modules incorporated within large engine simulation models, where computational efficiency is of primary importance . Mobility maps have been constructed for short, long, and finite-length journal bearing lubrication models.

This work aims to use the analytical mobility method to study journal bearings subjected to dynamic loads, with the intent to include it in a general computational program, such as the DAP (Dynamic Analysis Program), that has been developed for the dynamic analysis of general mechanical systems. A simple journal bearing subjected to a dynamic load is chosen as a demonstrative example, in order to provide the necessary results for the comprehensive discussion of the model presented throughout this work.

The main objective of the present study is to analyze the pressure distribution in hydrodynamic journal bearing for various loading conditions and various operating parameters. The space between the shaft and the bearing is called lubrication gap and is filled with lubricant. Journal bearing test rig is used to test the long bearing. Test bearing is located between two antifriction bearings. The bearing is loaded mechanically. The bearing is tested under various parameters like type of lubricant, loading conditions, speeds etc… In the last, experimental results are compared with the theoretical results and results are found satisfactorily. Lots of research have been done on the maximum pressure and still are going on the basis of various parameters to find maximum pressure for various method. In this project work, the work is carried out to find out that analytical method of maximum pressure which is closest to experimental method. The work contains theoretical as well as experimental analysis of journal bearing including design of particular bearing of L/D ratio as 1. The experimental set up is supported by software Winducom 2013 through which various readings have been taken and graphs for it. The experimental results of pressure distribution for selected grades of an oils are validated with theoretical ones for stability purpose and are found to be satisfactory. Finally, the result of all work are accompanied by the results, discussion and conclusion of the project.

II. THEORETICAL BACKGROUND AND CALCULATION PROCEDURE.

A. Assumptions.

i) Left hand side terms describe flow rate to pressure gradients known as Poiseuille pressure induced flow rate. ii) First two terms on right hand side describe flow rate due to motion of surfaces or shear known as Couette velocity induced flow rates.

B. Theoretical Analysis Of Hydrodynamic Journal Bearing.

When a journal bearing which has an adequate supply of lubricant is carrying a load it normally runs with the geometric centers of the shaft and housing displaced so that a region of convergent flow is established. In this region large hydrodynamic pressures are set up within the oil film and these pressures when summated over the total bearing surface are found to completely support the load.

If bearing conditions change; for instance the load may vary, the displacement or "attitude" of the centers changes so that the new pressure distribution is sufficient to support the new load. Fig. 1 illustrates the basic principles. Various non-dimensional load parameters are used to assess the performance of a bearing. The parameters are formed from terms such as the oil viscosity and the load and speed of the bearing. For a given bearing it is found that there is a critical value of load parameter at which the convergent film is unable to support the total bearing load and the surfaces touch. Under these conditions the friction torque suddenly starts to rise and boundary lubrication occurs in the region of minimum film thickness. Figs. 2 illustrate these phenomena. Based on his theoretical investigation of cylindrical journal bearings, Professor Osborn Reynolds showed that oil, because of its adhesion to the journal and its resistance to flow (viscosity), is dragged by the rotation of the journal so as to form a wedge-shaped film between the journal and journal bearing.

Operating Parameter

Diameter of journal= Dj (in mm)
Length of bearing= L(in mm)
L/D ratio
Clearance=C(in mm)
Type of lubricant= SAE grade oils Viscosity of lubricant=μ(in Pa-sec)
Speed of journal=N(in rpm)
Load on journal bearing=W(in rpm)
Eccentricity ratio=ε
h Film thickness
U Surface speed of shaft
Re Reynolds number
L Bearing axial length
R Journal radius
C Radial clearance
N Shaft speed
e Eccentricity
ε Eccentricity ratio
Ω Domain of calculation
K_{ij} Stiffness matrix term at i , j
F_i Load matrix term
J (ξ ,η) Jacobian matrix
μ Lubricant viscosity

C. Reynolds Equation For Two Dimensional Flow From Navier-Strokes Equation.

For the study of lubricant films, the equations of motions for a viscous fluid are basics which are normally referred as Navier-Strokes Equations for a Newtonian fluid and can be given as below.

i. Inertia force per unit volume in x-direction:

$$\rho\frac{Du}{Dt} = \rho\ X- \frac{\partial p}{\partial x} - \frac{2}{3}\ \frac{\partial}{\partial x}(\mu*\Delta)+2\ \frac{\partial}{\partial x}(\mu*\frac{\partial u}{\partial x})+ \frac{\partial}{\partial y}\left[\mu\left(\frac{\partial u}{\partial y}+\frac{\partial v}{\partial x}\right)\right]+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial u}{\partial z}+\frac{\partial w}{\partial x}\right)\right] \quad (1)$$

ii. Inertia force per unit volume in y-direction:

$$\rho\frac{Dv}{Dt} = \rho\ Y- \frac{\partial p}{\partial y} - \frac{2}{3}\ \frac{\partial}{\partial y}(\mu*\Delta)+2\ \frac{\partial}{\partial y}(\mu*\frac{\partial v}{\partial y})+ \frac{\partial}{\partial z}\left[\mu\left(\frac{\partial v}{\partial z}+\frac{\partial w}{\partial y}\right)\right]+\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial v}{\partial x}+\frac{\partial u}{\partial y}\right)\right] \quad (2)$$

iii. Inertia force per unit volume in z-direction:

$$\rho\frac{Dw}{Dt} = \rho\ Z- \frac{\partial p}{\partial z}- \frac{2}{3}\ \frac{\partial}{\partial z}(\mu*\Delta)+2\ \frac{\partial}{\partial z}(\mu*\frac{\partial w}{\partial z})+ \frac{\partial}{\partial x}\left[\mu\left(\frac{\partial w}{\partial x}+\frac{\partial u}{\partial z}\right)\right]+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial w}{\partial y}+\frac{\partial v}{\partial z}\right)\right] \quad (3)$$

Where, $\frac{Du}{Dt}$, $\frac{Dv}{Dt}$ and $\frac{Dw}{Dt}$ are the total derivatives of x, y and z components of velocities of fluid respectively.(OR component of acceleration of fluid along x, y and z directions respectively).

X,Y and Z are components of body forces per unit mass,

$\Delta = \frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}$ is pressure viscosity coefficient

ρ is mass density.

u, v and w are velocity components of fluid in x,y and z directions respectively.

These equations are used to derive generalized Reynolds equation. Also the flow of lubrication must satisfy continuity requirements expressed by the continuity equations given below.

In cartesian co-ordinate system,

V≡v(x,y,z)

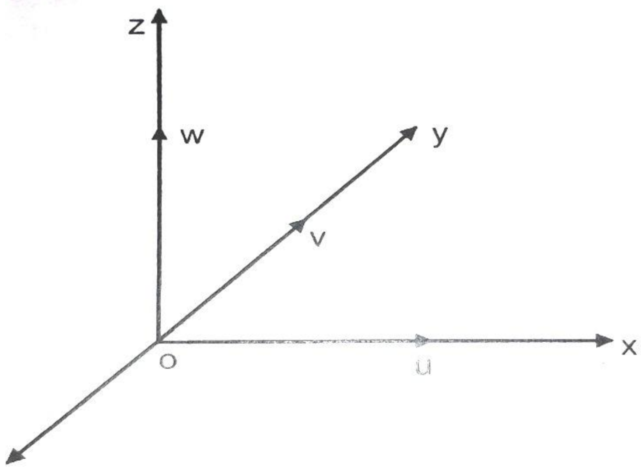
The continuity equation takes the following form:

$\frac{\partial \rho}{\partial t} \equiv \frac{\partial}{\partial x}(\rho u)+\frac{\partial}{\partial y}(\rho v) + \frac{\partial}{\partial z}(\rho w)=0$

Total change of velocity in x-direction in time interval dt is,

$Du=\frac{\partial u}{\partial x}dx + \frac{\partial u}{\partial y}dy + \frac{\partial u}{\partial z}dz+\frac{\partial u}{\partial t}$ dt

$$\frac{Du}{Dt} = \underbrace{\frac{\partial u}{\partial x}dx + \frac{\partial u}{\partial y}dy + \frac{\partial u}{\partial z}dz}+\underbrace{\frac{\partial u}{\partial t}} \qquad (4)$$



Based on assumptions made, reduced form of Navier-Strokes Equation is given by,

$\frac{\partial p}{\partial x}=\mu\frac{\partial^2 u}{\partial z^2}$

And

$$\frac{\partial p}{\partial y}=\mu\frac{\partial^2 v}{\partial z^2} \qquad (5)$$

As p is a function of x and y, on integrating above equations with respect to z, the general expression for velocity gradients treating viscosity constant can be given as,

$\int\frac{\partial p}{\partial x}.dz=\int \mu\frac{\partial^2 u}{\partial z^2}.dz$

$z.\frac{\partial p}{\partial x}+A= \mu\frac{\partial u}{\partial z}$

$$\frac{\partial u}{\partial z} = \frac{z}{\mu}\frac{\partial p}{\partial x}+\frac{A}{\mu} \qquad (6)$$

Similarly where A and C are constants of integration

$$\frac{\partial v}{\partial z} = \frac{z}{\mu}\frac{\partial p}{\partial y}+\frac{C}{\mu} \qquad (7)$$

Once again integrating above equation w .r.t. z,

$u= \frac{z^2}{2\mu}\frac{\partial p}{\partial x} + \frac{A.z}{\mu} + B$ where B and D are additional integration constants to be

$v= \frac{z^2}{2\mu}\frac{\partial p}{\partial y} + \frac{C.z}{\mu} + D$ determined using suitable boundary conditions.

Where, u_a, $v_a$and w_a are velocity components of upper surface in x,y and z directions respectively. u_b, $v_b$and w_b are velocity components of lower surface in x,y and z directions respectively.

Boundary conditions:

i. At z=0 $\begin{cases} u = u_b \\ v = v_b \end{cases}$

And

ii. At z=h $\begin{cases} u = u_a \\ v = v_a \end{cases}$

Using the Boundary conditions in above equations for velocities u and v, we have

i. $u_b=\frac{0^2}{2\mu}\frac{\partial p}{\partial x}+\frac{A.(0)}{\mu}+B$, $\quad B = u_b$

ii. $v_a=\frac{0^2}{2\mu}\frac{\partial p}{\partial y}+\frac{C.(0)}{\mu}+D$, $\quad D = v_b$

Substituting these constants in equations for u and v we get,

$u_b=\frac{Z^2}{2\mu}\frac{\partial p}{\partial x}+\frac{A.(Z)}{\mu}+u_b$

And

$v=\frac{z^2}{2\mu}\frac{\partial p}{\partial y}+\frac{C.(Z)}{\mu}+v_b$

$u_a=\frac{h^2}{2\mu}\frac{\partial p}{\partial x}+\frac{A.h}{\mu}+u_b$

$A=\left[(u_a - u_b) - \frac{h^2}{2\mu}\frac{\partial p}{\partial x}\right]\frac{\mu}{h}$

And

$v_a=\frac{h^2}{2\mu}\frac{\partial p}{\partial x}+\frac{C.h}{\mu}+v_a$

$C=\left[(v_a - v_b) - \frac{h^2}{2\mu}\frac{\partial p}{\partial x}\right]\frac{\mu}{h}$

Substituting constants A and C in above equations for u and v we get,

$$u=\frac{Z^2}{2\mu}\frac{\partial p}{\partial x}+\left[(u_a - u_b) - \frac{h^2}{2\mu}\frac{\partial p}{\partial x}\right]\frac{Z}{h}+u_b \qquad (8)$$

$=\frac{Z^2}{2\mu}\frac{\partial p}{\partial x}-\frac{z.h}{2\mu}\frac{\partial p}{\partial x}+\frac{Z}{h}u_a-\frac{Z}{h}u_b+u_b$

$$u=\underbrace{\frac{1}{2\mu}\frac{\partial p}{\partial x}(Z^2 - z.h)}_{\text{Poiseuille flow term}}+\underbrace{\left(1-\frac{Z}{h}\right)u_b+\frac{Z}{h}u_b}_{\text{coquette flow termV}} \qquad 9$$

$v==\frac{Z^2}{2\mu}\frac{\partial p}{\partial y}+\left[(v_a - v_b) - \frac{h^2}{2\mu}\frac{\partial p}{\partial y}\right]\frac{Z}{h}+v_b$

Actual volumetric flow rates in direction of sliding per unit width. Consider elemental fluid at a distance z from body b having thickness dz and unit width in y-direction. In x-direction.

$q_x = \int_0^h (dz * 1)$ .u

$$=\int_0^h \left(\frac{z^2}{2\mu}\frac{\partial p}{\partial x} + \left[(u_a-u_b) - \frac{h^2}{2\mu}\frac{\partial p}{\partial x}\right]\frac{z}{h} + u_b\right) dz$$

$$=\left\{\left(\frac{z^3}{6\mu}\frac{\partial p}{\partial x} + \left[(u_a-u_b) - \frac{h^2}{2\mu}\frac{\partial p}{\partial x}\right]\frac{z^2}{2h} + z.u_b\right)\right\}_0^h$$

$$= \frac{h^3}{6\mu}\frac{\partial p}{\partial x} + \left[(u_a-u_b) - \frac{h^2}{2\mu}\frac{\partial p}{\partial x}\right]\frac{h}{2h} + h.u_b$$

$$= \frac{\partial p}{\partial x}\left[\left(\frac{h^3}{6\mu} - \frac{h^3}{4\mu}\right)\right] + \left[(u_a-u_b)\right]\frac{h}{2} + h.u_b$$

$$q_x = -\frac{h^3}{12\mu}.\frac{\partial p}{\partial x} + \frac{(u_a+u_b)}{2}.h \quad (10)$$

similarly we can write volume flow rate in y-direction as,

$$q_y = -\frac{h^3}{12\mu}.\frac{\partial p}{\partial y} + \frac{(v_a+v_b)}{2}.h \quad (11)$$

now consider continuity equation,

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) + \frac{\partial}{\partial z}(\rho w) = 0 \quad (12)$$

Integrating above equation over the film thickness w.r.t. dz

$$\int_0^h \left[\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) + \frac{\partial}{\partial z}(\rho w)\right] dz = 0$$

Let,

$$I = \int_0^h \frac{\partial \rho}{\partial t}.dz = \frac{\partial \rho}{\partial t}.h = h.\frac{\partial \rho}{\partial t}$$

$$II = \int_0^h \frac{\partial}{\partial x}(\rho u)dz$$

Using general form of Leibnitz's integration rule.

$$\int_0^h \frac{\partial}{\partial x}F(x,y,z)dz = -F(x,y,z)\frac{\partial h}{\partial x} + \frac{\partial}{\partial x}\int_0^h f(x,y,z)dz$$

Now, $II = \int_0^h \frac{\partial}{\partial x}(\rho u)dz = -\rho u\frac{\partial h}{\partial x} + \frac{\partial}{\partial x}\int_0^h (\rho u)dz$

$= -\rho u\frac{\partial h}{\partial x} + \frac{\partial}{\partial x}\rho \left[\int_0^h (u)dz\right]\}$ at z=h

$u = u_a$

$III = \int_0^h \frac{\partial}{\partial x}(\rho v)dz = -\rho v_a\frac{\partial h}{\partial y} + \frac{\partial}{\partial y}\rho\left[\int_0^h (v)dz\right]\}$ at z = h

$v = v_a$

$IV = \int_0^h \frac{\partial}{\partial z}(\rho w)dz = \rho\int_{wb}^{wa}\partial w = \rho(w_a - w_b)$

Substituting I, II, III, IV in above equation.

$$h\frac{\partial \rho}{\partial t} - \rho u_a\frac{\partial h}{\partial x} + \frac{\partial}{\partial x}\left[\rho\int_0^h (u)dz\right] - \rho v_a\frac{\partial h}{\partial y} + \frac{\partial}{\partial y}\left[\rho\int_0^h (v)dz\right] - \rho v_a\frac{\partial h}{\partial y} + \frac{\partial}{\partial y}\left[\rho\left(-\frac{h^3}{12\mu}.\frac{\partial p}{\partial y} + \frac{(v_a+v_b)}{2}.h\right)\right] + \rho(w_a-w_b) = 0$$

$$h\frac{\partial \rho}{\partial t} - \rho u_a\frac{\partial h}{\partial x} + \frac{\partial}{\partial x}\left[\rho\left(-\frac{h^3}{12\mu}.\frac{\partial p}{\partial x} + \frac{(u_a+u_b)}{2}.h\right)\right]\rho(w_a-w_b) = 0$$

$$\frac{\partial}{\partial x}\left[\left(\frac{\rho h^3}{12\mu}.\frac{\partial p}{\partial x}\right)\right] + \frac{\partial}{\partial y}\left[\left(\frac{\rho h^3}{12\mu}.\frac{\partial p}{\partial y}\right)\right] = \frac{\partial}{\partial x}\left[\left(\frac{\rho h(u_a+u_b)}{2}\right)\right] + \frac{\partial}{\partial y}\left[\left(\frac{\rho h(v_a+v_b)}{2}\right)\right] + \rho(w_a-w_b) - \rho u_a\frac{\partial h}{\partial x} - \rho v_a\frac{\partial h}{\partial y} + h.\frac{\partial \rho}{\partial t}$$

(13)

Last four terms on right hand side describe the net flow rates due to squeeze motion and local compression.

Neglecting side leakage (i.e. pressure is constant in y-direction). Reynolds equation can be written as.

$$\frac{\partial}{\partial x}\left[\left(\frac{\rho h^3}{12\mu}.\frac{\partial p}{\partial x}\right)\right] = \frac{\partial}{\partial x}\left[\left(\frac{\rho h(u_a+u_b)}{2}\right)\right] - \rho u_a\frac{\partial h}{\partial x} + \rho(w_a-w_b) + h.\frac{\partial \rho}{\partial t} \quad (14)$$

In infinitely short (narrow) hydrodynamic journal bearings, the length of bearing in axial direction is very short. Hence, there is considerable flow of fluid in an axial direction o Z-direction.

In such bearings, the pressure gradient in an axial direction or Z-direction $\partial p/\partial z$ is much higher than the pressure gradient in X-direction $\partial p/\partial z$,

That is $\frac{\partial p}{\partial z} >> \frac{\partial p}{\partial x}$

Hence in infinitely short (narrow) hydrodynamic journal bearings, the pressure gradient in X-direction is neglected. i.e. $\frac{\partial p}{\partial x} \approx 0$.

The journal bearing with length to diameter ratio (l/d ratio)less than 0.5 are considered as infinitely short (narrow) hydrodynamic journal bearings.

III. ANALYTICAL METHOD(RAPID METHOD)

By using mathematical modeling, develop & modify the relationship between the average pressure, load capacity, minimum oil film thickness & stiffness, so, as to minimize the effect of oil whirl or whip & instability in the rotor system.

By using experimental set up of JBTR 60, investigate & to determine the stability of hydrodynamic journal bearing, by observing distribution in the journal bearing.

Thus compare the analytical method with experimental method by using Winducom 2013 software, analyze the pressure distribution.

A. Combined graph of maximum pressure value for different speed and load condition for Oil 20W40

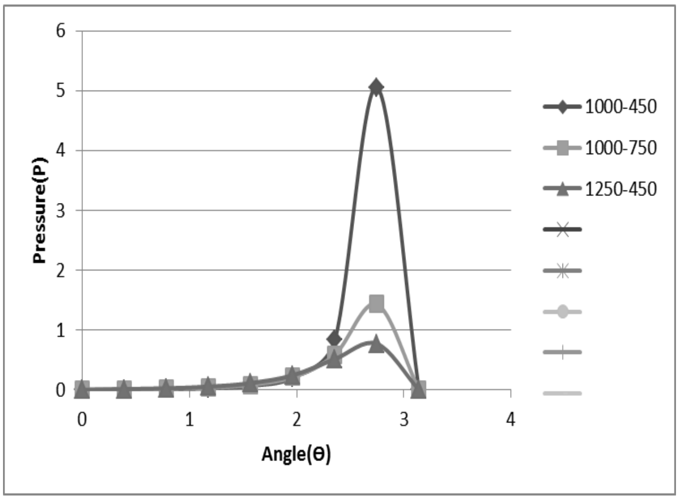


Fig. 1: Variation of pressure with angular position and load condition at 1000,1250 rpm and 450,750 N Load Grade Oil SAE 20W40

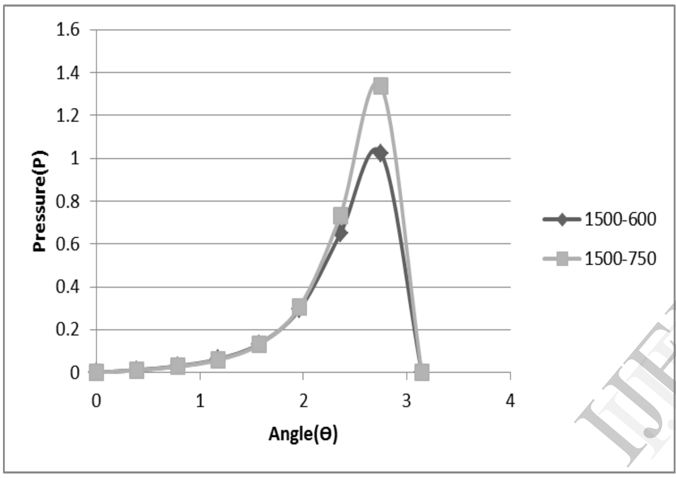


Fig. 2: Variation of pressure with angular position and load condition at 1500 rpm and 600,750 N Load Grade Oil SAE 20W40.

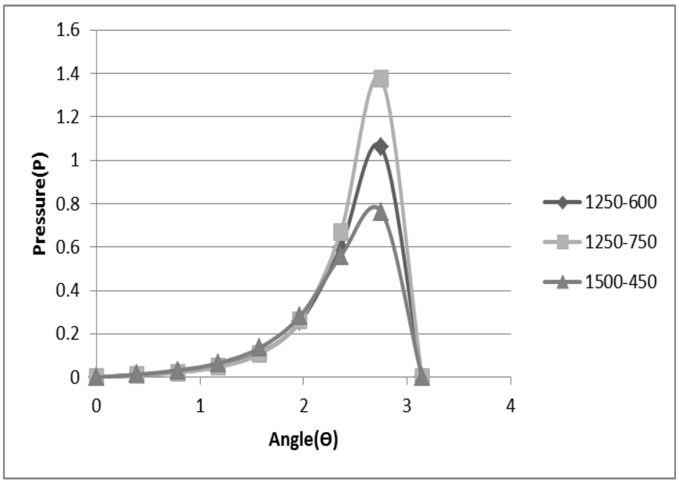


Fig. 3: Variation of pressure with angular position and load condition at 1250,1500 rpm and 450,600,750 N Load Grade Oil SAE 20W40.

B . Combined graph of maximum pressure value for different speed and load condition for Oil 15W30.

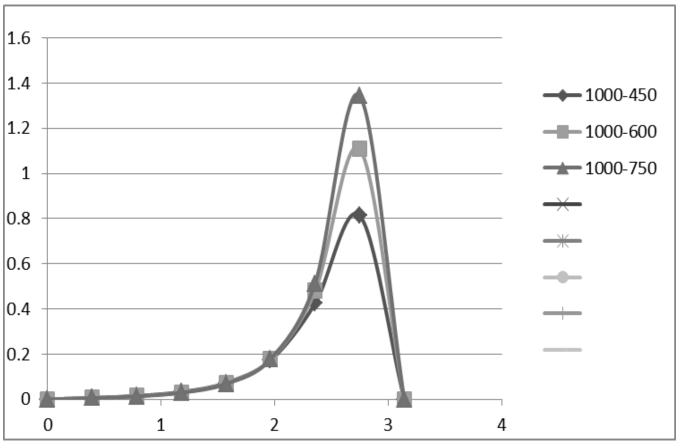


Fig. 4: Variation of pressure with angular position and load condition at 1000 rpm and 400, 600,750 N Load Grade Oil SAE 15W30.

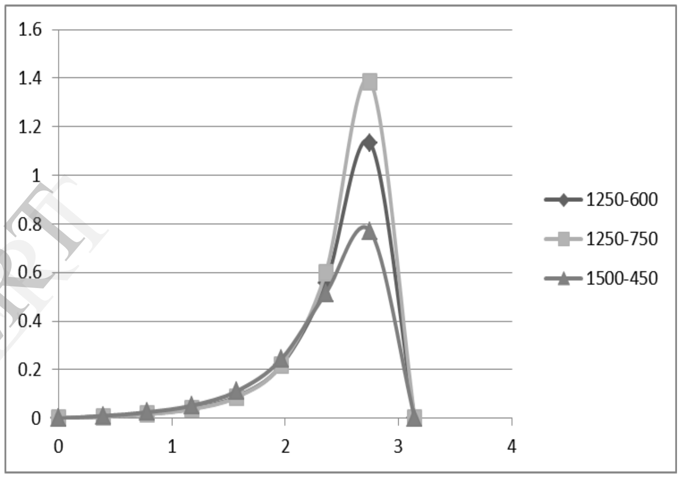


Fig. 5: Variation of pressure with angular position and load condition at 1250,1500 rpm and 400, 600,750 N Load Grade Oil SAE 15W30

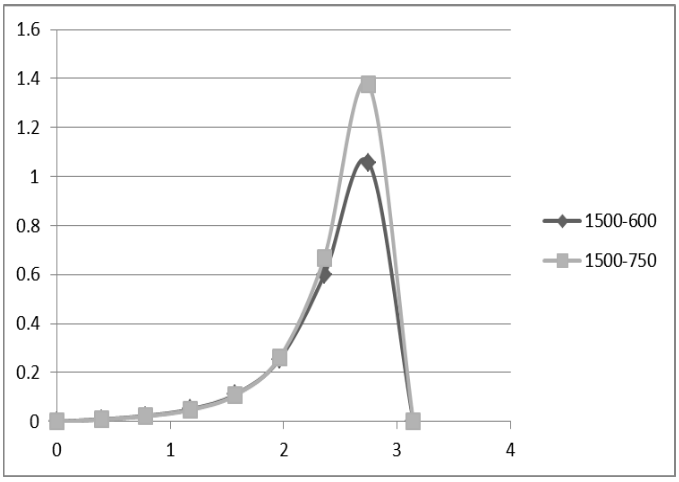


Fig. 6: Variation of pressure with angular position and load condition at 1500 rpm and 600,750 N Load Grade Oil SAE 15W30

C. Combined graph of maximum pressure value for different speed and load condition for Oil 10W30.

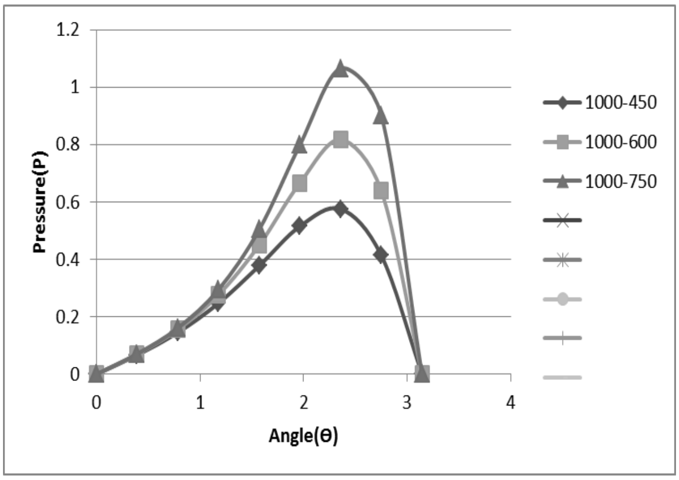


Fig. 7: Variation of pressure with angular position and load condition at 1000 rpm and 450, 600,750 N Load Grade Oil SAE 10W30

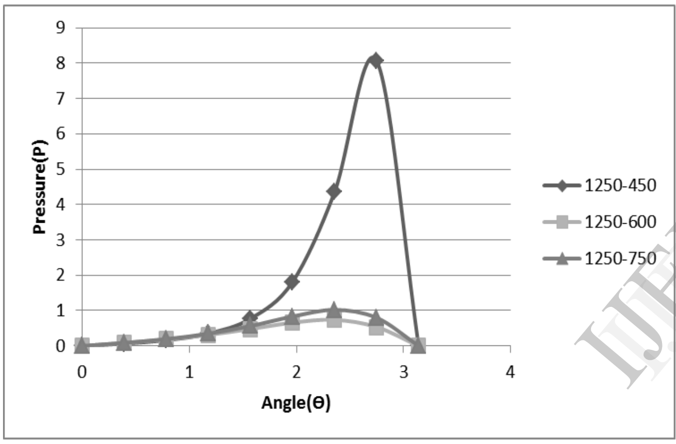


Fig. 8: Variation of pressure with angular position and load condition at 1250 rpm and 450, 600,750 N Load Grade Oil SAE 10W30

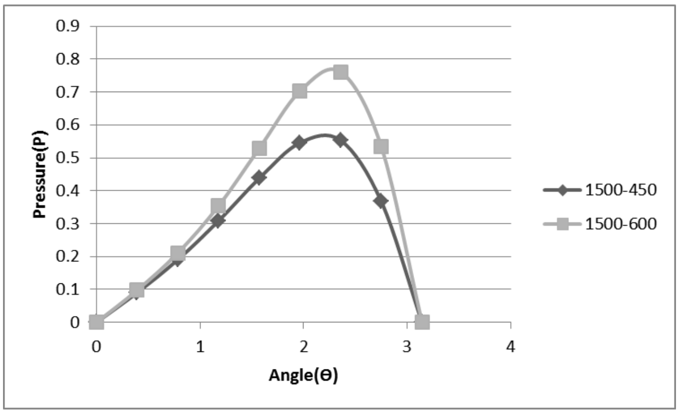


Fig. 9: Variation of pressure with angular position and load condition at 1500 rpm and 450, 600 N Load Grade Oil SAE 10W30.

All graphs shows pressure distribution is in increasing order with respect to an angle. Then it comes to zero as angle reaches to $180^{0.}$

IV. EXPERIMENTAL ANALYSIS OF PRESSURE DISTRIBUTION AT DIFFERENT CONDITION

According to above test procedure, experimentation is carried out for three different grades of oils SAE20W40, SAE15W40, SAE 10W30. At different conditions are calculated theoretically for various loading conditions. Theoretical pressure are validated experimentally on test rig JBTR-60 with the help of winducom software. From the plot we can see that the pressure distribution.

A. Oil Grade- SAE 10W30.

Load-6000N RPM-1250,L/D=1 Viscosity of lubricant, μ=00.06Pa-sec

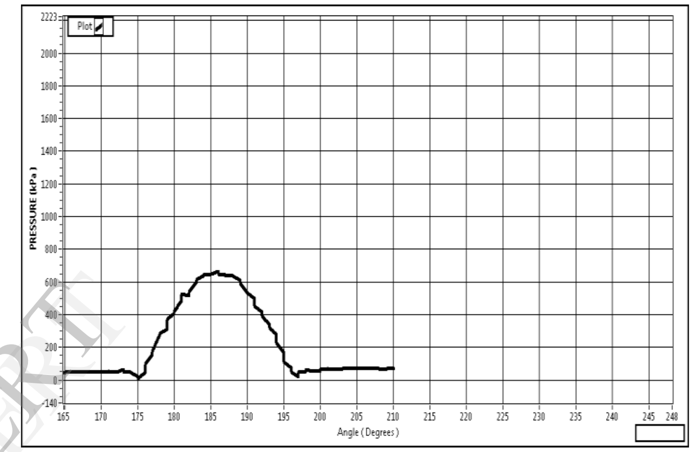


Fig. 10: Variation of pressure with angular position and load condition at IN Experimental method of Grade Oil SAE 10W30

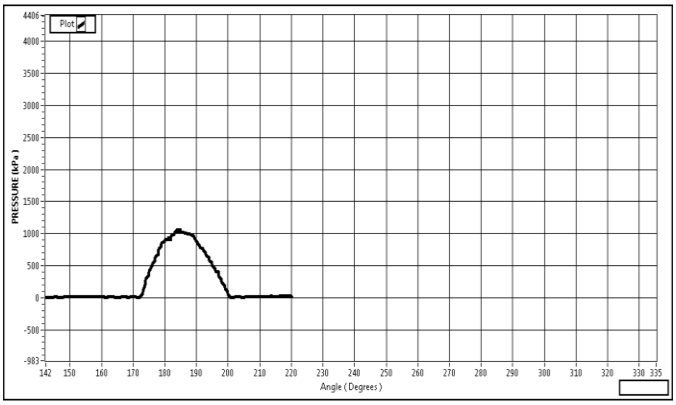


Fig. 11: Variation of pressure with angular position and load condition at in Experimental method of Grade Oil SAE 10W30

B. Oil Grade- SAE 15W40

Load-600N RPM-1250,L/D=1 Viscosity of lubricant, μ=0.0819Pa-sec

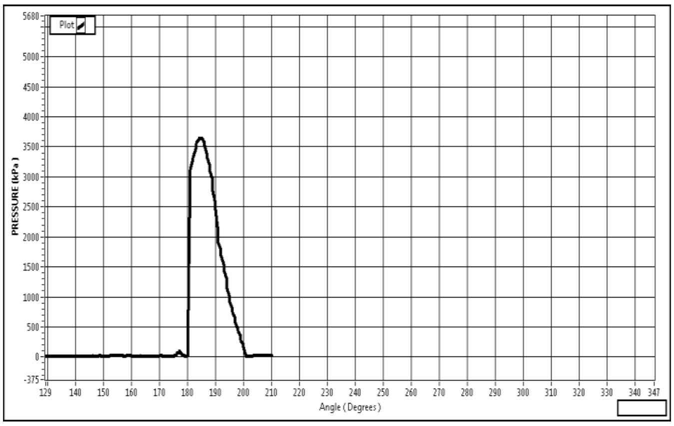


Fig. 12: Variation of pressure with angular position and load condition at in Experimental method of Grade Oil SAE 15W40.

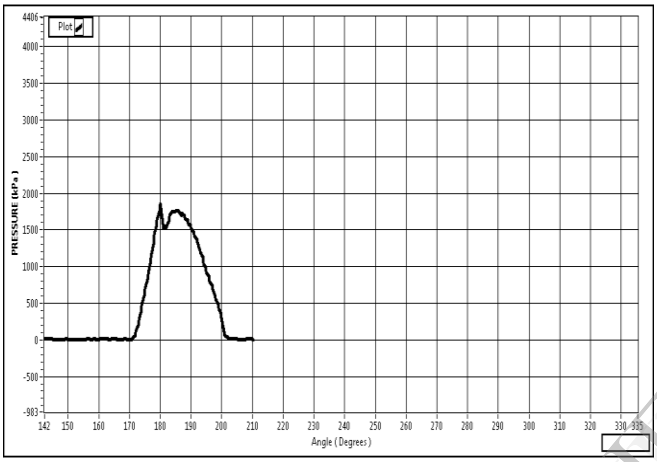


Fig. 13: Variation of pressure with angular position and load condition at in Experimental method of Grade Oil SAE 15W40

C.Oil Grade- SAE 20W40

Load-600N RPM-1250,L/D=1 Viscosity of lubricant, μ=0.0981Pa-sec

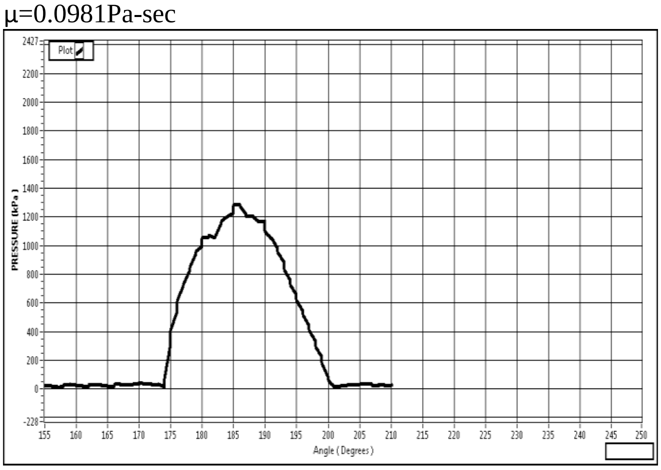


Fig. 14: Variation of pressure with angular position and load condition at in Experimental method of Grade Oil SAE 20W40.

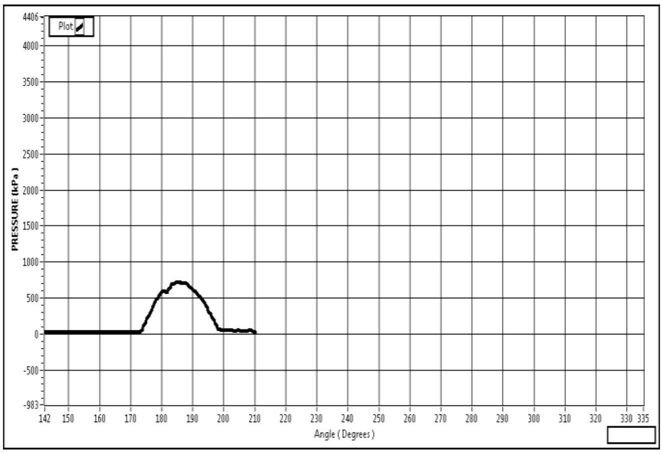


Fig. 15: Variation of pressure with angular position and load condition at in Experimental method of Grade Oil SAE 20W40.

V. RESULT AND DISCUSSION

1. For SAE 10W30 from theoretical analysis and experimental results at 600N,therotically and experimentally the pressure distribution at 135°is 0.738 (N-sec/mm2) and 0.662 (N-sec/mm2) respectively.
2. For SAE 15W30 from theoretical analysis and experimental results at 600N,therotically and experimentally the pressure distribution at 160°is 1.131 (N-sec/mm2) and 1.117(N-sec/mm2) respectively
3. For SAE 20W40 from theoretical analysis and experimental results at 750N,therotically and experimentally the pressure distribution at 157°is 1.37 (N-sec/mm2) and 1.281(N-sec/mm2) respectively

OIL GRADE-SAE10W30 *μ=00.06Pa-sec*				
Load N	Angle Θ	Therotical Pressure (N-sec/mm2)	Experimental Pressure (Nsec/mm2)	Percentage Error (%)
600	135	0.738	0.662	11.48
OIL GRADE-SAE20W40 *μ=0.0981Pa-sec*				
750	157	1.37	1.28	4.6
OIL GRADE-SAE15W40 *μ=0.0819Pa-sec*				
600	160	1.131	1.117	1.81

VI. CONCLUSION

i It is found that when comparing different analytical method with experimental method in which the results of rapid method which is good agreement with experimental method to find the pressure distribution of Reynolds equations.

ii Theoretical results are validated by experimentation on test rig JBTR-60 with support of Winducom 2013 software, which shows that the readings of pressure distribution are good agreement with theoretical ones at various Loads and speed.

iii The fluid film pressure developed in hydrodynamic journal bearing increases from 0^0 up to certain angle, then again decreases as an angle reaches to 180^0.

REFERENCES

[1] Zhang Yongqi1,Tan Qingchang1,Zhang Kuo1,Li Jiangang2 "Analysis of Stress and Strain of the Rolling Bearing by FEA method" Physics procedia 24 (2014) pp. 19-24

[2] D. C. Lo, D. L. Young, and K. Murugesan "GDQ Method For Natural Convection In A Square Cavity Using Velocity–Vorticity Formulation" Numerical Heat Transfer, Part B, 47: 321–341, 2005

[3] Tao He, Xiqun Lu, and Jingzhi Zhu "Mobility Method Applied to Calculate the Lubrication Properties of Bearing under Dynamic Loads" Hindawi Publishing Corporation ISRN Mechanical Engineering Volume 2013, Article ID 461472, 5 pages

[4] Biao Yu And Jerzy T. Sawicki* "Comparison of Mobility Method and Mass Conservation Method in a Study of Dynamically Loaded Journal Bearings". International Journal of Rotating Machinery, 8(1): 71±79, 2002.

[5] V. K. Dwivedi, Sanjeev Kumar Gupta "Finite Difference Method Analysis of Hybrid Bearing" 5th International Conference on Advances in Mechanical Engineering (ICAME-2011), June 06-08, 2011

[6] U. Ghia, K. N. Ghia, And C. T. Shin "High-Re Solutions for Incompressible Flow Using the Navier-Stokes Equations and A Multigrid Method*"Journal Of Computational Physics 48, 387-411 (1982)

[7] M. Senthil Kumar, P.R. Thyla, E. Anbarasu " Numerical Analysis Of Hydrodynamic Journal Bearing Under Transient Dynamic Conditions" ISSN 1392 - 1207. Mechanika. 2010. Nr.2(82)

[8] Paulo Flores1, JC Pimenta Claro, Jorge Ambrósio "Journal Bearings Subjected To Dynamic Loads: The Analytical Mobility Method"

[9] Dimitris Sfyris and Athanasios Chasalevris "An Analytical Solution Of The Reynolds Equation For The Finite Journal Bearing And Evaluation Of The Lubricant Pressure"

[10] C. Shu*, L. Wang And Y.T. Chew "Comparative Studies of Three Approaches for GDQ Computation of Incompressible Navier–Stokes Equations in Primitive Variable Form" International Journal of Computational Fluid Dynamics, July 2004 Vol. 18 (5), pp. 401–412.

[11] D. Benasciutti, M. Gallina, M. Gh. Munteanu, F. Flumian "A Numerical Approach for Static and Dynamic Analysis of Deformable Journal Bearings" World Academy of Science, Engineering and Technology 67 2012.

[12] T. K. Sengupta and A. Dipankar "A Comparative Study of Time Advancement Method for Solving Navier–Stokes Equations" Journal of Scientific Computing, Vol. 21, No. 2, October 2004

[13] Arto Sorsimo, Mika Juntunen, Rolf Stenberg and Juha Videman "Finite element analysis of the Reynolds lubrication equation with cavitation" (Journal of Structural Mechanics) Vol. 45, No 4, 2012, pp. 188 -200

[14] A. T. Prata, R. T. S. S. Ferriera,D. E. B. Mile and M. G. D. Bortoli "Dynamically Loaded Journal Bearings: Finite Volume Method Analysis"