International Journal of Multidisciplinary and Current Research

Research Article

ISSN: 2321-3124
Available at: http://ijmcr.com

Analysis of Fitted Bearings under Second Order Rotatory Theory of Hydrodynamic Lubrication

Mohammad Miyan

Department of Mathematics, Shia P.G. College, Lucknow, India

Accepted 05 Aug 2015, Available online 10 Aug 2015, **Vol.3 (July/Aug 2015 issue)**

Abstract

The second order rotatory theory of hydrodynamic lubrication was founded on the expression obtained by retaining the terms containing first and second powers of rotation number in the extended generalized Reynolds equation. In the present paper, there are derivations of new equations for pressure and load capacity and some new excellent fundamental solutions with the help of geometrical figures, expressions, calculated tables and graphs for the fitted bearings in the second order rotatory theory of hydrodynamic lubrication. The analysis of equations for pressure and load capacity, tables and graphs reveal that pressure and load capacity are not independent of viscosity and increases slightly with viscosity. Also the pressure and load capacity both increases with increasing values of rotation number. In the absence of rotation, the equation of pressure and load capacity gives the classical solutions of the classical theory of hydrodynamic lubrication. The relevant tables and graphs confirm these important investigations in the present paper.

Keywords: *Continuity, Rotation number, Taylor's number, Reynolds equation, Film thickness, Permeability.*

1. Introduction

The hydrodynamic bearings can be divided in to four categories generally:

(a) Rolling element bearings for example; ball, cylindrical, spherical or tapered roller and needle etc.
(b) Dry bearings for example; plastic bushings, coated metal bushings etc.
(c) Semi-lubricated bearings for example; oil-impregnated bronze bushings etc.
(d) Fluid film bearings for example; crankshaft bearings etc.

Except from some radial-configuration aircraft engines, almost all piston engines use fluid film bearings. This is true for the crankshaft and sometimes in the camshaft, although often the later runs directly in the engine structure. Here we have to discuss the working of the fluid film working and to demonstrate how engine designers are reducing friction losses through bearing technology given by *Hori et al.* [[10]] in 2005. *Dowson et al.* [7] in 1962, had already given that the fluid film bearings operate by generating, as a by-product of the relative motion between the shaft and the bearing, a very thin film of lubricant at a sufficiently high pressure to match the applied load, as long as that load is within the bearing capacity. Fluid film bearings represent a form of scientific process, by virtue of providing very large load carrying capabilities in a compact, lightweight implementation, and unlike the other classes, in most cases can be designed for infinite life. The fluid film bearings operate in any of the three modes:

(a) Fully-hydrodynamic
(b) Boundary
(c) Mixed.

In fully hydrodynamic or" full-film" lubrication, the moving surface of the journal is completely separated from the bearing surface by a very thin film of lubricant, was given by *Cameron* [4] in 1981. The applied load causes the centerline of the journal to be displaced from the centerline of the bearing. This eccentricity creates a circular "wedge" in the clearance space.

The lubricant, by virtue of its viscosity, clings to the surface of the rotating journal, and is drawn into the wedge, creating a very high pressure, which acts to separate the journal from the bearing to support the applied load.

The bearing eccentricity is expressed as the centerline displacement divided by the radial clearance. The bearing eccentricity increases with applied load and decreases with greater journal speed and viscosity. The hydrodynamic pressure has no relationship at all to the engine oil pressure, except that if there is insufficient engine oil pressure to deliver the required copious volume of oil into the bearing, the hydrodynamic pressure mechanism will fail and the bearing and journal will be destroyed. The pressure distribution in the hydrodynamic region of a fluid film bearing increases from quite low in the large clearance zone to its maximum at the point of minimum film thickness for the incompressible fluid like oil is pulled into the converging "wedge" zone of the bearing, given by *Cameron* [4], [5]. However, this radial profile does not exist homogeneously across the axial length of the bearing. If the bearing has sufficient width, the profile will have a nearly flat shape across the high-pressure region.

The second mode of bearing operation is boundary lubrication. In boundary lubrication, the "peaks" of the sliding surfaces i.e., journal and bearing, are touching each other, but there is also an extremely thin film of the lubricant only a few molecules thick which is located in the surface "valleys". That thin film tends to reduce the friction from what it would be if the surfaces were completely dry.

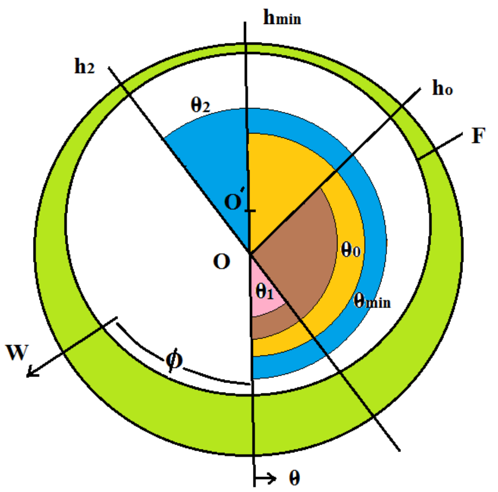

Figure-1(a) (Hydrodynamic journal bearing)

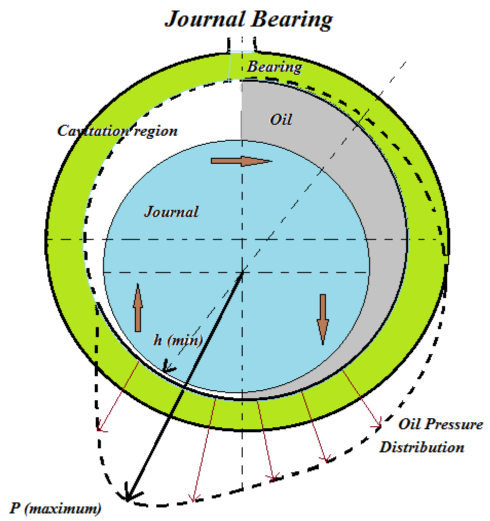

Figure-1(b) (Motion of fluid in Hydrodynamic journal bearing)

The mixed mode is a region of transition between boundary and full-film lubrication. The surface peaks on the journal and bearing surfaces partially penetrate the fluid film and some surface contact occurs, but the hydrodynamic pressure is starting to increase. When motion starts, the journal tries to climb on the wall of the bearing due to the metal- to-metal friction between the two surfaces.

If there is an adequate supply of lubricant, the motion of the journal starts to drag the lubricant into the wedge area and hydrodynamic lubrication begins to occur along with the boundary lubrication. If we assume that the load and viscosity remain relatively constant during this startup period, then as revolution per minute increases, the hydrodynamic operation strengthens until it is fully developed and it moves the journal into its steady state orientation. The direction of the eccentricity and the minimum film thickness, do not occur in line with the load vector and are angularly displaced from the load.

According to *Hori* [10], there are also some other form of fluid-film lubrication, which includes the squeeze-film lubrication i.e., the piston engine etc. Squeeze-film action is based on the fact that a given amount of time is required to squeeze the lubricant out of a bearing axially, thereby adding to the hydrodynamic pressure, and therefore to the load capacity. Since there is little or no significant rotating action in the wrist-pin bores, squeeze-film hydrodynamic lubrication is the prevailing mechanism which separates wrist pins from their bores in the rods and pistons.

The figure-1 shows a hydrodynamic journal bearing and a journal, are rotating in the clockwise direction. The rotation of the journal causes pumping of the lubricant that flows around the bearing in the direction of rotation. If there is no force applied to the journal then its position remains unaltered and concentric to the bearing position. The loaded journal moves from the concentric position and forms converging gap between the journal surfaces and bearing. Now the movement of journal forced the lubricant to squeeze through the gap generating the pressure. *Hori et al.* [10]had said that the pressure falls to the cavitations pressure i.e., closer to the atmospheric pressure in the gap in which the cavitations forms. In general the two types of cavitations are form in the journal bearing.

(a) **Gaseous cavitations:** This is associated with air and gases mixed with lubricant. If the pressure of lubricant falls below the atmospheric pressure then the gases come out to form the cavitations.

(b) **Vapors cavitations:** This is formed when the load applied to the bearing fluctuates at the high frequency. The pressure of fluid falls rapidly and causes the cavitations due to fast evaporation.

Now the fluid pressure creates the supporting force which separates the journal from the surface of the bearing.

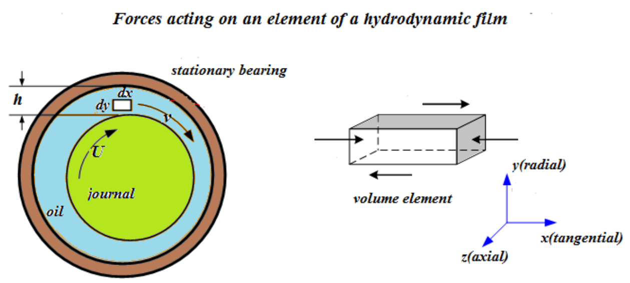

Figure-2 (Forces acting on a unit volume of an element of hydrodynamic lubrication film)

The hydrodynamic force of friction and force of fluid pressure counterbalance the external load. So the position of journal can be determined by these forces. In the hydrodynamic regime, the journal climbs in the rotational direction. If the working of journal is in the boundary and mixed lubrication then the hydrodynamic pressure ends and the journal climbs in the opposite to the rotational direction.

In the theory of hydrodynamic lubrication, two dimensional classical theories were first given by *Osborne Reynolds* [11]. In 1886, in the wake of a classical Beauchamp Tower's experiment given by *Reynolds* [13], he formulated an important differential equation, which was known as: Reynolds Equation given by *Reynolds* [12] in 1886. The formation and basic mechanism of fluid film was analyzed by that experiment on taking some important assumptions given as:

(a) The fluid film thickness is very small as compare to the axial and longitudinal dimensions of fluid film.
(b) If the lubricant layer is to transmit pressure between the shaft and the bearing, the layer must have varying thickness.

Later *Osborne Reynolds* himself derived an improved version of Reynolds Equation known as: "Generalized Reynolds Equation" [12], [13], which depends on density, viscosity, film thickness, surface and transverse velocities. The concept of rotation was discussed by *Banerjee et al.* [1], [2] in 1981 that the rotation of the fluid film which lies across the film gives some new results in lubrication problems of fluid mechanics. The origin of rotation can be traced by certain general theorems related to vorticity in the rotating fluid dynamics. The rotation induces a component of vorticity in the direction of rotation of fluid film and the effects arising from it are predominant, for large Taylor's Number, it results in the streamlines becoming confined to plane transverse to the direction of rotation of the film.

The new extended version of "Generalized Reynolds Equation" is said to be "Extended Generalized Reynolds Equation" given by *Banerjee et al.* [1], [2], which takes into account of the effects of the uniform rotation about an axis that lies across the fluid film and depends on the rotation number M , i.e. the square root of the conventional Taylor's Number. The generalization of the classical theory of hydrodynamic lubrication is known as the "Rotatory Theory of Hydrodynamic Lubrication" given by *Banerjee et al.* [1], [2]. The "First Order Rotatory Theory of Hydrodynamic Lubrication" and the "Second Order Rotatory Theory of Hydrodynamic Lubrication" was given by retaining the terms containing up to first and second powers of M respectively by neglecting higher powers of M, was given by *Banerjee et al.* [1], [2], [3].

The bearings having its diameter equal to the journal are known as fitted bearings or non clearance bearings. In these bearings the radial clearance is zero. The figure-3 shows the motion of fluid in fitted bearing.

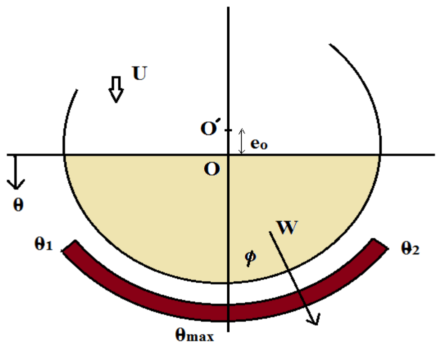

Figure-3 (Geometry of Fitted Bearing)

2. Governing Equations

In the second order rotatory theory of hydrodynamic lubrication the "Extended Generalized Reynolds Equation" [7]is given as:

$$
\begin{aligned}
&\frac{\partial}{\partial x}\left[-\sqrt{\frac{2\mu}{M\rho}}\frac{1}{M}\left(\frac{\sinh h\sqrt{\frac{M\rho}{2\mu}}-\sin h\sqrt{\frac{M\rho}{2\mu}}}{\cosh h\sqrt{\frac{M\rho}{2\mu}}+\cos h\sqrt{\frac{M\rho}{2\mu}}}\right)\frac{\partial P}{\partial x}\right]\\
&+\frac{\partial}{\partial y}\left[-\sqrt{\frac{2\mu}{M\rho}}\frac{1}{M}\left(\frac{\sinh h\sqrt{\frac{M\rho}{2\mu}}-\sin h\sqrt{\frac{M\rho}{2\mu}}}{\cosh h\sqrt{\frac{M\rho}{2\mu}}+\cos h\sqrt{\frac{M\rho}{2\mu}}}\right)\frac{\partial P}{\partial y}\right]\\
&+\frac{\partial}{\partial x}\left[-\frac{h}{M}+\sqrt{\frac{2\mu}{M\rho}}\frac{1}{M}\left(\frac{\sinh h\sqrt{\frac{M\rho}{2\mu}}+\sin h\sqrt{\frac{M\rho}{2\mu}}}{\cosh h\sqrt{\frac{M\rho}{2\mu}}+\cos h\sqrt{\frac{M\rho}{2\mu}}}\right)\frac{\partial P}{\partial y}\right]\\
&-\frac{\partial}{\partial y}\left[-\frac{h}{M}+\sqrt{\frac{2\mu}{M\rho}}\frac{1}{M}\left(\frac{\sinh h\sqrt{\frac{M\rho}{2\mu}}+\sin h\sqrt{\frac{M\rho}{2\mu}}}{\cosh h\sqrt{\frac{M\rho}{2\mu}}+\cos h\sqrt{\frac{M\rho}{2\mu}}}\right)\frac{\partial P}{\partial x}\right]\\
&=\\
&-\frac{U}{2}\frac{\partial}{\partial x}\left[\rho\sqrt{\frac{2\mu}{M\rho}}\left(\frac{\sinh h\sqrt{\frac{M\rho}{2\mu}}+\sin h\sqrt{\frac{M\rho}{2\mu}}}{\cosh h\sqrt{\frac{M\rho}{2\mu}}+\cos h\sqrt{\frac{M\rho}{2\mu}}}\right)\right]-\\
&\frac{U}{2}\frac{\partial}{\partial y}\left[-\rho\sqrt{\frac{2\mu}{M\rho}}\left(\frac{\sinh h\sqrt{\frac{M\rho}{2\mu}}-\sin h\sqrt{\frac{M\rho}{2\mu}}}{\cosh h\sqrt{\frac{M\rho}{2\mu}}-\cos h\sqrt{\frac{M\rho}{2\mu}}}\right)\right]-\rho W^{*} \qquad (1)
\end{aligned}
$$

Where *x, y* and *z* are coordinates, *P* is the pressure, *ρ* is the fluid density, *μ* is the viscosity and $W^*$is fluid velocity in *z*-direction. The Extended Generalized Reynolds Equation in view of second order rotatory theory of hydrodynamic lubrication, in ascending powers of rotation number *M*and by retaining the terms containing up to second powers of *M* and neglecting higher powers of *M*, can be written as:

$$\frac{\partial}{\partial x}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial x}\right]+\frac{\partial}{\partial y}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial y}\right]+\frac{\partial}{\partial x}\left[-\frac{M\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\frac{\partial P}{\partial y}\right]-\frac{\partial}{\partial y}\left[-\frac{M\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\frac{\partial P}{\partial x}\right]=-\frac{\partial}{\partial x}\left[\frac{\rho U}{2}\left\{h-\frac{M^2\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\right\}\right]-\frac{\partial}{\partial y}\left[\frac{M\rho^2U}{2}\left\{-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\right\}\right]-\rho W^* \quad (2)$$

For the case of pure sliding $W^*=0$, so we have the equation as given:

$$\frac{\partial}{\partial x}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial x}\right]+\frac{\partial}{\partial y}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial y}\right]+\frac{\partial}{\partial x}\left[-\frac{M\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\frac{\partial P}{\partial y}\right]-\frac{\partial}{\partial y}\left[-\frac{M\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\frac{\partial P}{\partial x}\right]=-\frac{\partial}{\partial x}\left[\frac{\rho U}{2}\left\{h-\frac{M^2\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\right\}\right]-\frac{\partial}{\partial y}\left[\frac{M\rho^2U}{2}\left\{-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\right\}\right] \quad (3)$$

3. Formulation of Problem

Let we assume the bearing to be infinitely long in *x*-direction, which implies that the variation of pressure in *y*-direction is very small as compared to the variation of pressure in *x*-direction i.e., $\frac{\partial P}{\partial x}\gg\frac{\partial P}{\partial y}$, then the equation (3) will be

$$\frac{\partial}{\partial x}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial x}\right]-\frac{\partial}{\partial y}\left[-\frac{M\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\frac{\partial P}{\partial x}\right]=-\frac{\partial}{\partial x}\left[\frac{\rho U}{2}\left\{h-\frac{M^2\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\right\}\right]-\frac{\partial}{\partial y}\left[\frac{M\rho^2U}{2}\left\{-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\right\}\right] \quad (4)$$

Taking *h, U, P* has given

$$h = h(x),\ U = -U,\ P = P(x) \quad (5)$$

The film thickness in angular coordinates is given as:

$$h = e_0\cos\vartheta \quad (6)$$

Here $e_0$is the eccentricity.

By rotating the angular coordinate 90°, in the direction of motion, we have

$$h= e_0\sin\vartheta,\ x = R\vartheta \quad (7)$$

In view of above conditions, the equation (4) can be written as:

$$\frac{\partial}{\partial x}\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\frac{\partial P}{\partial x}\right]=-\frac{\partial}{\partial x}\left[\frac{\rho U}{2}\left\{h-\frac{M^2\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\right\}\right] \quad (8)$$

4. Discussion and Results

For the determination of pressure distribution excluding the negative regions, the boundary conditions are as follows:

$$P=K\frac{dP}{d\theta}=0,\ \text{at } \vartheta=\vartheta_2,\ K=\text{constant} \quad (9)$$

Where ϑ_1 and ϑ_2 are connected by the condition that

$$P=0 \text{ at } \vartheta=\vartheta_1. \quad (10)$$

On integrating and using the boundary conditions (9), (10) the equation of pressure after neglecting the higher powers of *M* and retaining the terms up to M^2 is given as:

$$P=\frac{3\mu UR}{{e_0}^2}[F_1(\theta_1)-F_1(\theta)]+\frac{M^2\rho^2{e_0}^2\,RU}{280\mu}[F_2(\theta_1)-F_2(\theta)] \quad (11)$$

Where $F_1(\vartheta)$ and$F_2(\vartheta)$ are given by expressions:

$$F_1(\theta)=2cot\theta-sin\theta_2\left(cosec\theta\ cot\theta-logtan\frac{\theta}{2}\right) \quad (12)$$

$$F_2(\theta)=17\left(\frac{sin2\theta}{4}-\frac{\theta}{2}-sin\theta_2\ cos\theta\right)-7\left\{\begin{matrix}sin^5\theta_2\left(logtan\frac{\theta}{2}-cosec\theta\ cot\theta\right)\\-\theta+\frac{sin2\theta}{2}\end{matrix}\right\} \quad (13)$$

The load capacity for porous bearing is given by

$$W=\sqrt{W_x^2+W_y^2} \quad (14)$$

Here W_x and W_y are the components of the load capacity in *x*-direction and *y*-direction respectively.

$$W_x = \int_{\theta_1}^{\theta_2} LRP\ sin\theta\ d\theta \quad (15)$$

$$W_y = \int_{\theta_1}^{\theta_2} LRP\ cos\theta\ d\theta \quad (16)$$

heW_x and W_y in the increasing values of M are given by

$$W_x = \frac{3\mu ULR^2}{{e_0}^2}[F_1(\theta_1)cos\theta_1 - F_1(\theta_1)cos\theta_2 + G_1(\theta_1) - G_1(\theta_2)] + \frac{M^2\rho^2{e_0}^2R^2\ UL}{280\mu}[F_2(\theta_1)cos\theta_1 - F_2(\theta_1)cos\theta_2 + G_2(\theta_1) - G_2(\theta_2)] \quad (17)$$

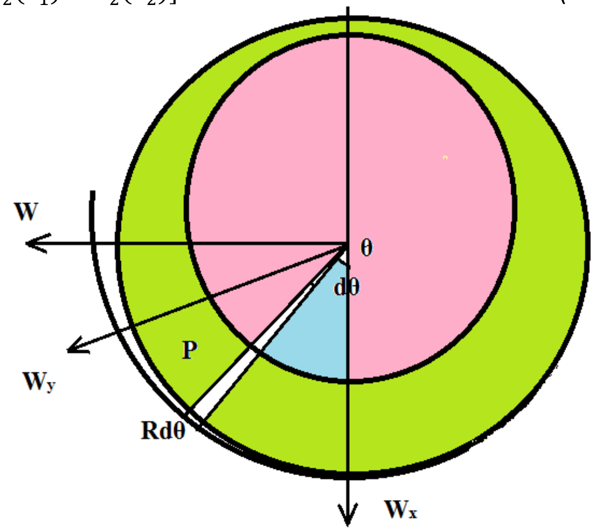

Figure-3 Geometry for Components of Load Capacity

$$W_y = \frac{3\mu ULR^2}{{e_0}^2}[F_1(\theta_1)sin\theta_2 - F_1(\theta_1)sin\theta_1 + H_1(\theta_1) - H_1(\theta_2)] + \frac{M^2\rho^2{e_0}^2R^2\ UL}{280\mu}[F_2(\theta_1)sin\theta_2 - F_2(\theta_1)sin\theta_1 + H_2(\theta_1) - H_2(\theta_2)] \quad (18)$$

Where $G_1\ (\vartheta)$, $G_2\ (\vartheta)$, $H_1\ (\vartheta)$ and $H_2\ (\vartheta)$ are given by the expressions:

$$G_1(\theta) = -2sin\theta + sin\theta_2\ log\ sin\theta - sin\theta_2\left(log\ sin^2\frac{\theta}{2} - 2cos^2\frac{\theta}{2}\ log\ tan\frac{\theta}{2}\right) \quad (19)$$

$$G_2(\theta) = \frac{3}{2}(sin\theta - \theta\ cos\theta) - \frac{1}{2}sin^3\theta - \frac{17}{4}sin\theta_2\ cos2\theta + 7sin^5\theta_2\left(log\ sin^2\frac{\theta}{2} - 2cos^2\frac{\theta}{2}\ log\ tan\frac{\theta}{2} - log\ sin\theta\right) \quad (20)$$

$$H_1(\theta) = -2\left(cos\theta + log\ tan\frac{\theta}{2}\right) - sin\theta_2\left(cot\theta + sin\theta\ log\ tan\frac{\theta}{2}\right) \quad (21)$$

$$H_2(\theta) = \frac{3}{2}(cos\theta + \theta\ sin\theta) + \frac{1}{2}cos^3\theta + \frac{17}{2}sin\theta_2\left(\theta + \frac{sin2\theta}{2}\right) \quad (22)$$

5. Numerical Simulation

Let us take the values of mathematical terms in C.G.S. system as follows:

ρ = 0.9, μ = 0.0002, e_0 =0.3, R = 3.35, U = 500, ϑ_1 =20^0, ϑ =50^0, ϑ_2 =160^0.

The calculated values of pressure and load capacity are given by the table-1.

Table-1 (The variation of pressure and load capacity with respect to rotation number M)

S.No.	M	P	W
1	0.1	80.85488142	045449.4097
2	0.2	327.5560264	296635.8569
3	0.3	738.7246014	427591.0792
4	0.4	1314.360606	761970.4658
5	0.5	2054.464041	1191886.946
6	0.6	2959.034906	1717340.766
7	0.7	4028.073201	2338331.607
8	0.8	5261.578926	3054859.437
9	0.9	6659.552081	3866924.387
10	1.0	8221.992666	4774526.387

Again, let us take the values of mathematical terms in C.G.S. system as follows:

ρ = 0.9, M = 0.1, e_0=0.3, R = 3.35, U = 500, ϑ_1 =20^0, ϑ =50^0, ϑ_2 =160^0.

The calculated values of pressure and load capacity are given by the table-2.

Table-2 (The variation of pressure and load capacity with respect to viscosity μ)

S.No.	μ	P	W
1	0.0001	40.42744071	22724.70485
2	0.0002	80.85488142	45449.4097
3	0.0003	121.2823221	68174.11455
4	0.0004	161.7097628	90898.8194
5	0.0005	202.1372036	113623.5243
6	0.0006	242.5646443	136348.2291
7	0.0007	282.992085	159072.934
8	0.0008	323.4195257	181797.6388
9	0.0009	363.8469664	204522.3437
10	0.0010	404.2744071	227247.0485

With the help of calculated data, we can say that the pressure and load capacity both increases with the increasing values of the rotation number M. The graphical representation of these values is shown in figure-4 and figure-5.

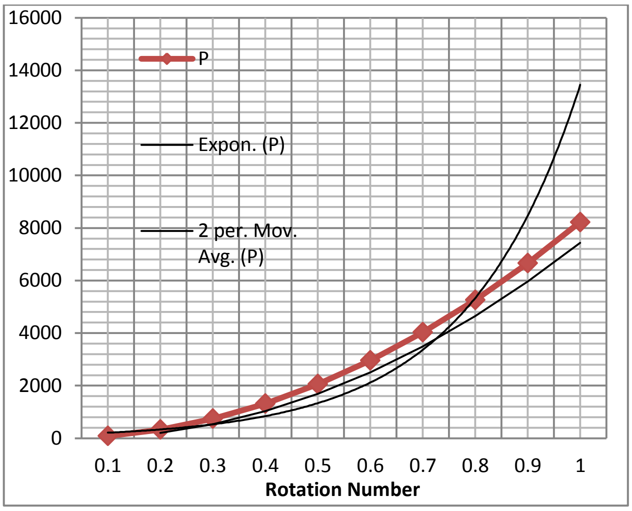

Figure 4. Variation of Pressure with respect to Rotation Number

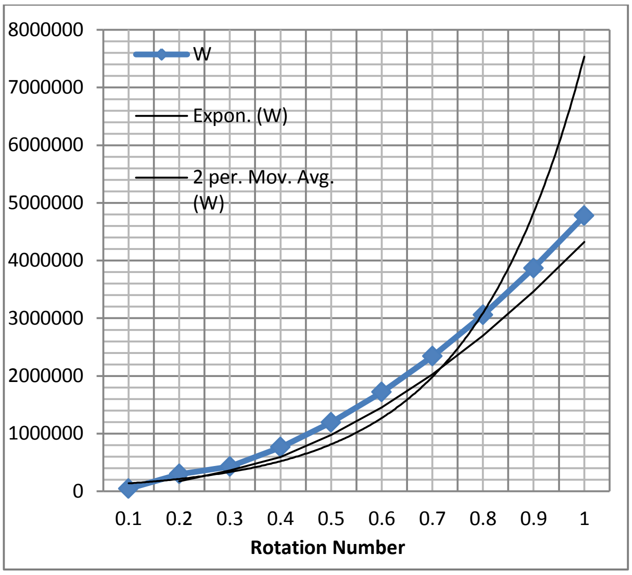

Figure 5. Variation of Load Capacity with respect to Rotation Number

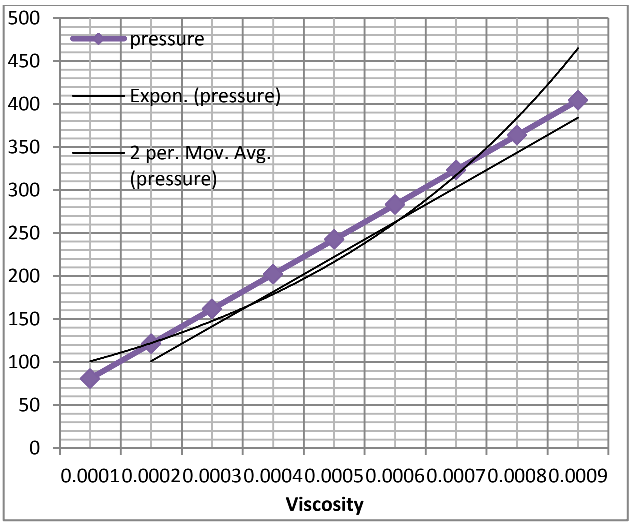

Figure 6. Variation of Pressure with respect to Viscosity μ

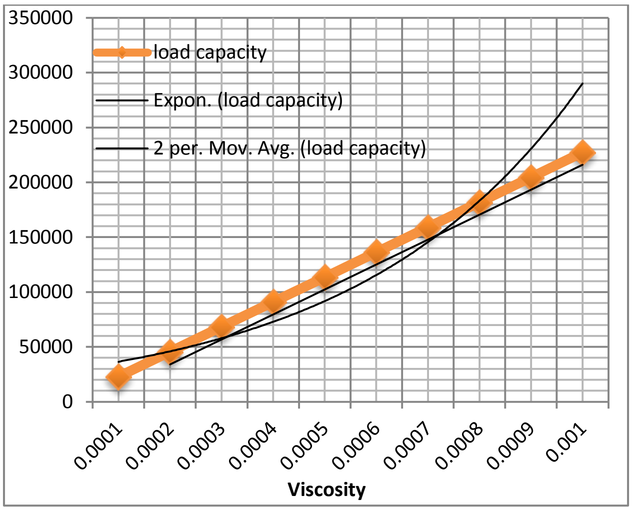

Figure 7. Variation of Load capacity with respect to Viscosity μ

The figure-6 and figure-7 show the variation of pressure and load capacity with respect to viscosity μ with exponential trends and moving averages. The figures show that the pressure and load capacity both increase slightly with the increasing values of viscosity μ for the constant value of M=0.1.

Conclusions

The derived equations of pressure and load capacity are given by equation (11) and equation (14). The graphical representation for the variation of pressures and load capacities is also shown by figure-4 to figure-7. The equations and graphs show that the pressure and load capacity both increase with the increasing values of the rotation number M for the constant value of μ for the fluid of small rotation. The equations and graphs also show that the pressure and load capacity are not independent of viscosity. These increase slightly with increase in the value of viscosity of the fluid.

References

[1]. Banerjee, M.B., Gupta, R.S. and Dwivedi, A.P. (1981) The Effects of Rotation in Lubrication Problems, *WEAR* 69:205-220.

[2]. Banerjee, M.B., Chandra, P. and Dube, G.S. (1981) Effects of Small Rotation in Short Journal Bearings, *Nat. Acad. Sci. Letters*, Vol. 4(9):112-125.

[3]. Banerjee, M.B., Dube, G.S., Banerjee, K. (1982) The Effects of Rotation in Lubrication Problems: A New Fundamental Solutions, *WEAR* 79:311-323.

[4]. Cameron, A. (1981) *Basic Lubrication Theory*, Ellis Harwood Limited, Coll. House, Watergate, Chicester, p. 45-162.

[5]. Cameron, A. (1958) The Viscous Wedge Trans., *ASME* (1): 248-255.

[6]. Chandrasekhar, S. (1970) *Hydrodynamic and Hydro magnetic Stability*, Oxford University Press, London p: 83-87.

[7]. Dowson, D. (1962) A Generalized Reynolds Equations for Fluid Film Lubrication, *Int. J. Mech. Sci*. 4: 159.
[8]. Dube, G.S. and Chatterjee, A. (1988) *Proc. Nat. Acad. Sci. India*, (58) I: 79-85.
[9]. Halling, J. (1975) *Principles of Tribology*, The Macmillan Press Ltd., London, p: 369-379.
[10]. Hori, Y.(2005) *Hydrodynamic Lubrication,* Springer Science & Business Media p: 23-31.
[11]. Pinkus, O. and Sternlicht, B. (1961) *Theory of Hydrodynamic Lubrication,* Mc. Graw Hill Book Company, Inc. New York, p: 5-64.
[12]. Reynolds, O.1886. *Phil. Trans. Roy. Soc. London*, Part I: 177-189.
[13]. Reynolds, O. (1886) On the Theory of Lubrication and its Application to Mr. Beauchamp Tower's Experiment, *Phil. Trans. Roy. Soc. London*, 177(I): 157-167.
[14]. Saibel, E. A. and Macken, N.A. (1973) *Annual Review of Fluid Mechanics*, Inc. Palo. Alto, Cal. Vol.5: 20-56.
[15]. Shaw, M.C. and Macks, E.F. (1949) *Analysis and Lubrication of Bearings*, Mc. Graw Hill Book Company, Inc., New York p.155-165