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# Pressure Analysis of Composite Slider Bearing Under the Effect of Second Order Rotation of the Lubrication Theory

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**Abstract**

The initial and second order theory of rotation for lubrication fluid mechanics was supported the expressions obtained by holding the terms up to 1st and second powers of rotation number M among the extended generalized Reynolds equation of the classical Reynolds theory. Among this analysis, there are the derivations of the new equations for pressure beneath the results of second order rotation of lubrication fluid mechanics. The expression for the exponential and index variation of the pressure with respect M is obtained. The analysis provides some new glorious basic solutions with the assistance of geometrical figure, expressions, calculated table and graph for the composite slider bearings for second order rotation. The analysis of equations for pressure, table, and graph analyzes that the pressure will increase with increasing values of the rotation number. The pressure isn't independent of viscosity of fluid and varies with the viscosity of the fluid.

**Keywords:** Composite bearing, Pressure, Reynolds equation, Rotation number, Viscosity, Vorticity.

## 1. INTRODUCTION

The behavior of a fluid-film bearing depends on the boundary conditions at the interfaces between the liquid and also the solid bearing surfaces. For nearly all solid surfaces, the no-slip stipulation applies. However, variety of researchers has recently found that slip will occur with specially designed surfaces. Lubrication beneath high load and high speed is common in engineering apply these days. Within the thin film that separates the two rubbing surfaces a big quantity of heat is generated. Also, the surfaces are elastically distorted by the fluid film pressure and also the exaggerated temperature. Yet, on the far side the classical theory for equal and rigid fluid mechanics lubrication, very little is thought concerning the combined effects of temperature and solid distortion. Doubtless there's a necessity for an elementary study of the conditions that surround the fluid film [1].

Research into the physics of fluid film bearings was aroused by Fogg [2], United Nations agency discovered that parallel sliders will so carry loads. Later, Cope [3] and Cameron and Wood [4] coupled the energy balance equation within the fluid film with the momentum and continuity equations to get the temperature and also the pressure distributions. The result of prescribed thermal gradients was additionally studied for journal bearings [5], [6], [7], [8] yet as for slider bearings [9], [10]. Zienkiewicz [11] analyzed an infinitely long parallel slider bearing allowing temperature variations across the film. Afterwards, Hunter and Zienkiewicz [12] performed an analogous analysis for the one-dimensional slider bearing. In each analysis the thermal boundary conditions at the fluid-solid interface were assumed a priori. Dowson and Hudson [13], [14] reexamined a similar bearings allowing the heat transfer to the bearing solids to see the fluid film thermal boundaries. The result of thermal distortions of the bearing solids was thought-about by Otto Hahn and Kettleborough [15], [16], [17] in an analysis of the one-dimensional slider bearing.

They complete that whereas thermal distortions are accountable for the load carrying capability exhibited by parallel surface sliders, they need very little result on the performance of inclined sliders compared to facet run effects. McCallion, et al [18] conferred a thermohydrodynamic (THD) analysis of a finite bearing. The analysis indicated that the bearing load carrying capability is insensitive to heat transfer within the solids. It additionally showed that the thermohydrodynamic load isn't essentially finite by either the adiabatic or the equal solutions. Additional recently Ezzat and Rohde [19] analyzed the thermohydrodynamic behavior of a steady loaded inclined-plane slider, taking into consideration the finite breadth of the bearing and heat transfer to the solids. The results of elastic distortion on bearing performance are studied by Carl [20], Dowson, et al. [21], Higginson [22], O'Donoghue et al. [23], Benjamin' and Castelli [24], McCallion [25] and last, by American state and Huebner [26]. A number of these works are experimental in nature, some a mixture of experiment and theoretical analysis, and others, strictly analytical studies. None, however, have enclosed the results of temperature variation within the fluid film.

The composite bearings are made up of the combination of Tapered-Land and Flat-Land bearings. The geometry of the composite bearing is shown by the figure-1. The region $B_1$ consists of Tapered-Land and the region $B_2$ consists of the Flat-Land bearing.

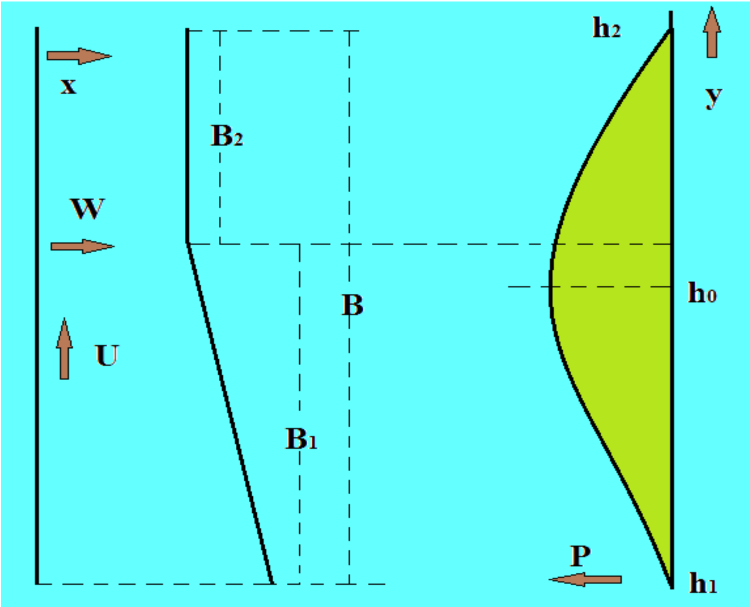


**Figure 1:** Geometry of Composite Slider Bearing

## 2. GOVERNING EQUATIONS OF THEORY OF ROTATION WITH BOUNDARY CONDITIONS

The two dimensional classical theory [27] of lubrication was first given by O. Reynolds. In 1886, at intervals the wake of a classical experiment by Beauchamp Tower [28], he developed associate equation celebrated as: Reynolds Equation. The formation and basic mechanism of fluid film was analyzed by that experiment by taking some assumptions that the film thickness is very small as compared to the axial and longitudinal dimensions of fluid film and if the lubricating substance layer is to transmit pressure between the shaft and thus the bearing, the layer ought to have variable thickness.

The rotation induces a section of vorticity at intervals the direction of rotation and additionally the consequences arising from it are predominant, for big Taylor's no., it winds up within the streamlines becoming confined to plane transverse to the direction of rotation [29]. The extended version of "Generalized Reynolds Equation" is claimed to be "Extended Generalized Reynolds Equation" given by Banerjee et. al., that takes into consideration of the implications of the uniform rotation concerning associate axis that lies across the fluid film and depends on the rotation no., M i.e., the square root of the classical Taylor's number. This generalization of the classical theory is known as a result of the "Rotatory Theory of Lubrication".

The "First order rotatory theory" and "Second order rotatory theory" of hydrodynamic Lubrication was given by Banerjee et.al. [30] on tenacious the terms containing up to first and second powers of M severally, and neglecting higher powers of M [31], [32]. This paper analyzes concerning the pressure at intervals the composite slider bearings with relevancy the impact of second order rotation and comparative analysis with relation to rotation number.

The Extended Generalized Reynolds Equation derived by Banerjee et al., [11], [12] in ascending powers of rotation no. M and by tenacious the terms containing up to second powers of M and neglecting higher powers of M, is written as equation (1). For the case of pure W*=0, and if the bearing is infinitely short then the pressure gradient in x-direction is much smaller than the pressure gradient in y-direction.

In y-direction the gradient $\partial_y P$ is of the order of (*P/L*) and within the *x*-direction, and is of order of *(P/B).* If $L << B$ then $P/L >> P/B$, so $\partial_x << \partial_y$. Then the terms containing $\partial_x$ can be neglected as compared to the terms $\partial_y$ containing in the expanded form of Generalized Reynolds Equation. Thus we've the equation as given:

$$\partial_y\left[F(h)\rho\partial_y\right] + \partial_x\left[G(h)\partial_y\right] = -\partial_x\left[\frac{\rho U}{2}\{h - M\,G(h)\}\right] - \partial_y\left[\frac{M\rho^2 U}{2}F(h)\right]$$
(1)

Where, $F(h) = \left[-\frac{h^3}{12\mu}\left(1 - \frac{17M^2\rho^2h^4}{1680\mu^2}\right)\right], G(h) = -\frac{M\rho^2h^5}{120\mu^2}\left(1 - \frac{31M^2\rho^2h^4}{3024\mu^2}\right)$
(2)

Where *x* and *y* area unit coordinates, *U* is that the sliding velocity, *P* is that the pressure, *ρ* is that the fluid density, *μ* is that the body.

Taking $h=h(y)$, $U=+U$, $P=P(y)$, $h=\alpha\overline{y}$, $h_1\text{-}h_2=\alpha\overline{B}_1$; for region $B_1$ and $h=h_2, y=B_2$; for region $B_2$.

The boundary conditions for the region $B_1$ are: $P=0$ *at* $h=h_1$ and $P=P_c$ *at* $h=h_2$.

The boundary conditions for the region $B_2$ are: $P=0$ *at* $y=0$ and $P=P_c$ *at* $y=B_2$.

The solution of the differential equation (1) under the boundary condition (2) gives the pressure for composite bearings is given by

$$P = -\frac{1}{2}M\rho U y + \frac{6\mu C}{\rho}\left[\frac{1}{\alpha\overline{y}^2} - \frac{17M^2\rho^2\,\overline{\alpha}}{1680\,\mu^2}y^2\right] + D \quad \text{(For region B}_1\text{)} \qquad (3)$$

Where, $C = \frac{B(h_1)-B(h_2)}{A(h_1)-A(h_2)}, D = \frac{A(h_1)B(h_2)-A(h_2)B(h_1)}{A(h_1)-A(h_2)}$ (4)

$$A(h) = \frac{12\mu}{\rho}\left(\frac{\overline{\alpha}}{h^2} - \frac{17M^2\rho^2h^2}{1680\,\mu^2\overline{\alpha}}\right), B(h) = \frac{1}{2}M\rho U\frac{h}{\overline{\alpha}} \qquad (5)$$

$$P = \frac{P_c}{B_2}y \quad \text{(For region B}_2\text{)} \qquad (6)$$

## 3. NUMERICAL SIMULATIONS

By taking the values of different mathematical terms in *C.G.S.* system as follows:

*e*=0.2, *μ*=0.0002, *C*=0.0067, *ρ*=1, *U*=80, *h*=0.02, $h_1$=00269 $h_2$=0.0167, $B_1$=1, $B_2$=0.5; the calculated values of pressure with respect *M* are given by table-1.

**Table-1**

| *S.NO.* | *M* | *P(Second Order Rotation)* |
|---|---|---|
| *1.* | *0.1* | *153.33617* |
| *2.* | *0.2* | *267.72894* |
| *3.* | *0.3* | *382.12381* |
| *4.* | *0.4* | *496.52868* |
| *5.* | *0.5* | *610.92878* |

## 4. CONCLUSIONS

The variation of pressure with respect to rotation number M by taking viscosity as constant; are shown by equations, table and graph. These show that within the second order rotation of fluid mechanics lubrication, the pressure will increase with increasing values of M, once viscosity is taken as arbitrary constant. The equations, tables and graphs for second order rotatory theory of fluid mechanics lubrication show that the pressure isn't independent of viscosity. The comparative exponential and logarithmic variations of pressure with relation to second order rotation are shown by the expressions:

$$P = 124.4\, e^{3.382M},\ P = 277 \log_e M + 754.7$$

$$\Rightarrow P \propto M,\ P \propto \mu$$

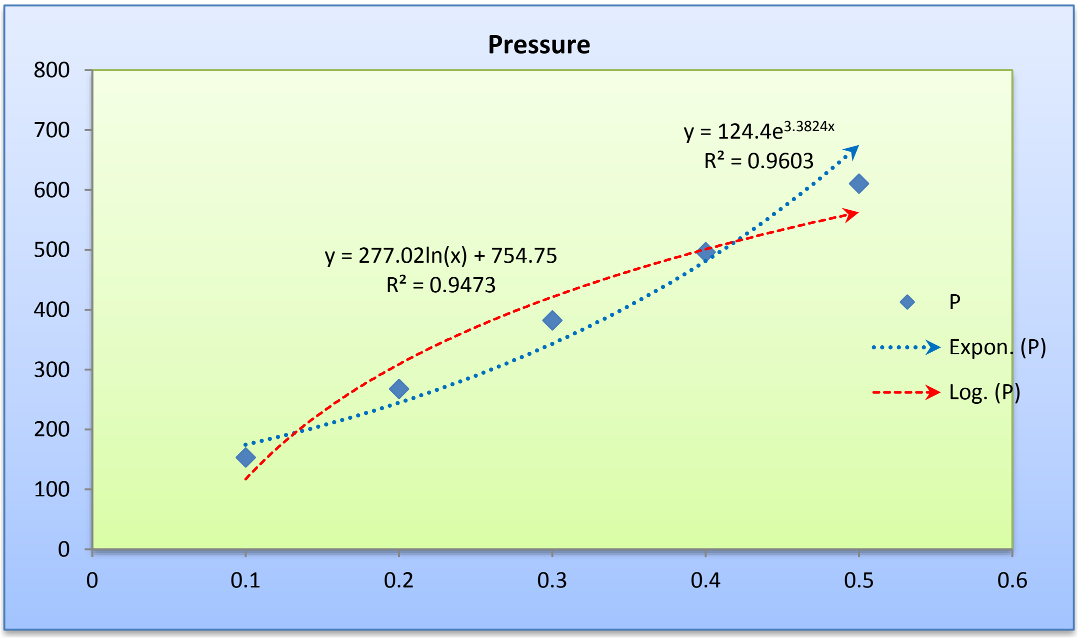


**Figure 2.** Variation of pressure with respect to *M*

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