



Pressure Analysis in Infinitely Short Bearings with Nanolubricants

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Abstract: The analysis of infinitely short bearing is conferred supported the thought of journal bearings and takes into thought of fluid film layer's thickness and consistence of nanoparticles additive fluid. Within the present paper, the results of additives on the nano lubricants have been investigated and reportable, aiming for improvement and additionally the development of fluid bearing performance. The comparative dimensionless pressure distribution with the improved viscosity over the fluid film for rotatory theory and classical theory has been calculated. The results show an increase within the pressure distribution in response to extending among the viscosity of the additives with respect to the numerous values of the rotation number. The non-dimensional pressure distribution is obtained from integration of Extended Generalized Reynolds Equation underneath the consequences of second order rotation by exploitation the Reynolds boundary conditions. The nano stuff Aerosil is used for the analysis with 5% volume fraction of nanoparticles. The fluid film properties and structure significantly influence the film fluid hydraulics lubrication for the nano lubricants underneath the consequences of second order rotation.

Keywords: Film thickness, Nano-lubricants, Pressure, Rotation number, Reynolds equation, Viscosity.

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I. Introduction

Over the previous few years, many researchers are specializing in the results of nanolubricant in fluid mechanics bearing. Nanoparticles within the oil significantly scale back the constant of friction and can increase load bearing capability. Kind of lubrication mechanisms are planned for nanoparticle suspension in oil like bearing result, protective film, mending result and sharpening results [1]. In general, for low loading, the losses attributable to friction and wear are negligible on contact surfaces. But below serious loading, the fluid film generated between the bearing and journal are getting to be intensely squeezed, and will not be able to defend contact surfaces. Some researches disclosed that nanoparticle additives enhance the supporting forces. Therefore, friction and wear are typically reduced by adding nanoparticle to the fabric. Many studies are being found to review the properties of nanolubricant and their application among the past decade. Varied forms of nanoparticles square measure used as material additives and their performance in skinny film lubrication are studied [2-5]. Of those studies showed reduced wear and friction in tribo-surfaces with the use of nanoparticle material additives. However, there is a serious gap in analysis regarding theoretical simulation of nanolubricant in fluid film bearing. Offered literature suggests improvement in static and dynamic performance characteristics by considering entirely variable viscosity analysis of nanoparticle additive [6-8].

In the thin film lubrication, the effective body of adsorbate layer below relative motion is far larger than the bulk Newtonian consistence. The adsorbate molecular layer thickness and body area unit the foremost essential factors in skinny film lubrication. The analysis on the body of nanofluids urged by Einstein supported dilute suspended spherical particles in viscous fluids. Einstein's pioneering formula is valid for spherical particles with low nanoparticles volume fraction [9]. Brinkman contributed to increase the Einstein equation considering the impact of moderate spherical nanoparticles suspensions in viscous fluids [10]. Batchelor planned the second order formula for the body of nanofluids considering motion impact of spherical nanoparticles additives and their interactions [11]. Each these models predict body of nanoparticles suspensions from the bottom fluid body and volume fraction of nanoparticles [10-11]. Krieger and Dougherty derived the shear body equation considering particle concentrations [12]. Tichy and Qingwen et al. studied the impact of lubricator molecular structure and developed skinny film model of solid surfaces adhered high body layer. Supported skinny film lubrication analysis, as body and thickness of adsorbate layer at solid surfaces will increase; results indicate increase in Load capability and cut back in constant of friction [13-14]. Nabhan et al. put together investigated binary fluid film lubricated fluid mechanics bearing. Nanoparticles additive lubricators increase the pressure distribution of fluid film bearings as results of increase in lubricator body [15]. Chen et al. derived changed Krieger and Dougherty

equation to predict high shear body of nanofluids supported mixture nanoparticles structure considering the results of variable packing fraction [16]. They conferred classification of nanofluids supported the volume fraction of nanoparticles (V_ϕ) betting on nanoparticles concentration and structure as: dilute ($0 < V_\phi \leq 0.001$), semi-dilute ($0.001 < V_\phi \leq 0.05$), semi-concentrated ($0.05 < V_\phi \leq 0.1$), and concentrated ($V_\phi \geq 0.1$).

Meurisse and Espejel along have given a Generalized Reynolds Equation for a three-layered film model. Analysis of bearing load capability and constant of friction in three-layered film is influenced by thickness of fluid film [9]. Nair et al. given characteristics of statically loaded bearing operational lubricants with nanoparticles additives [10]. Duangthongsuk and Wongwises planned correlations for thermal natural phenomenon and consistency of nanofluids supported experimental results. The consistency of nanofluids decreases considerably with increasing temperature and may increase with increasing particle volume concentration [11]. Szeri investigated composite film configuration for increase in load capability (high consistency lubricant) and cut back in friction (low consistency lubricant). The composite film bearing consists of upper and lower consistency fluid layers adjacent to bearing surface and journal surface, severally [12]. Hosseini et al. given associate empirical model of consistency of nanoparticles suspensions i.e., derived from consistency of the bottom fluid, particle volume fraction, particle size, properties of the wetter layer, and temperature [13, 14]. Shenoy et al. have given the influence of API-SF engine oil with nanoparticles additives on the characteristics of associate outwardly adjustable statically loaded fluid film bearing [15]. Babu et al. gave the impact lubricants with nanoparticles additives on static and dynamic characteristics of bearing. The lubricants with compound, metal compound, and mineral nanoparticles are used among the analysis. The viscosity models for nanoparticles additive lubricants were developed victimization on the suitable experimental analysis [16]. Mahbubul et al. have given review of theoretical models of consistency for suspensions and pictured consistency correlations for volume concentrations, temperature, and particle diameter of nanofluids [17].

Beauchamp Tower's exposition of fluid mechanics behavior of journal bearings in 1880's and his observations play some valuable results for playwright Sir Joshua Reynolds to carryout theoretical analysis. This has resulted throughout associate elementary equation for hydrodynamic lubrication. This has provided a robust foundation and basis for the planning of hydro-dynamic greased bearings [18, 19].

In his theoretical analysis, Reynolds thought of the assumptions as: The fluid is Newtonian, incompressible. The viscosity is constant throughout the film. The pressure does not vary at intervals the axial direction. The bearing and journal extend infinitely at intervals the z direction i.e., no lubricating flow at intervals the z direction. The film pressure is constant at intervals the y direction. Thus the pressure depends on the x coordinate exclusively. The speed of particle of lubricant at intervals the film depends exclusively on the coordinates x and y. The impact of physical phenomenon and gravitation is neglected. The fluid flow is laminar.

Later Sir Joshua Reynolds himself derived associate improved version of Reynolds Equation familiar with as: "Generalized Reynolds Equation" [19] that depends on density, viscosity, and film thickness, surface and transverse velocities. The rotation of fluid film regarding the axis that lies across the film offers some new finishes up in lubrication issues with hydrodynamics. The origin of rotation is usually derived by some general theorems relating to vorticity among the rotating fluid dynamics. The rotation induces an area of vorticity among the direction of rotation of fluid film then the implications arising from it are predominant, for giant Taylor's number, it finishes up within the streamlines changing into confined to plane transverse to the direction of rotation of the film.

The new extended version of "Generalized Equation" is thought to be "Extended Generalized Reynolds Equation" [21] that takes into thought of the implications of the uniform rotation concerning associated axis that lies across the fluid film and depends on the rotation number (M), i.e. the root of the Taylor's number. The generalization of the classical theory of fluid dynamics lubrication is thought as results of the "Rotatory Theory of Hydrodynamic Lubrication" [22].

The "First Order Rotatory Theory of Hydrodynamic Lubrication" and therefore the "Second Order Rotatory Theory of Hydrodynamic Lubrication" [22, 23] was given by retentive the terms containing up to first and second powers of (M) severally by neglecting higher powers of (M) [24, 25].

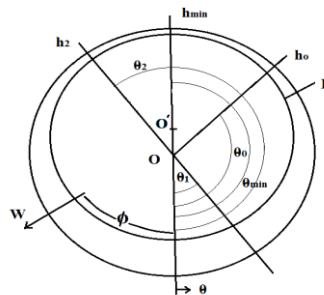


Figure-1 (Geometry of Journal Bearing)

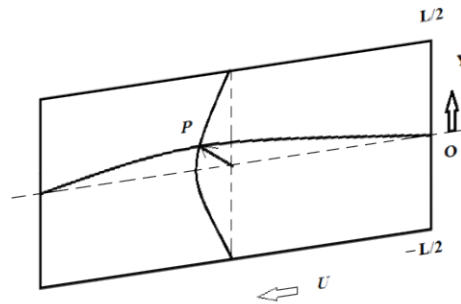


Figure-2 (Geometry of Short Journal Bearing)

II. Governing Equations and Boundary Conditions

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 [25-27] can be written as:

$$\begin{aligned} & \frac{\partial}{\partial x} \left[-\frac{h^3}{12\mu} \left(1 - \frac{17M^2\rho^2h^4}{1680\mu^2} \right) \rho \frac{\partial P}{\partial x} \right] + \frac{\partial}{\partial y} \left[-\frac{h^3}{12\mu} \left(1 - \frac{17M^2\rho^2h^4}{1680\mu^2} \right) \rho \frac{\partial P}{\partial y} \right] \\ & + \frac{\partial}{\partial x} \left[-\frac{M\rho^2h^5}{120\mu^2} \left(1 - \frac{31M^2\rho^2h^4}{3024\mu^2} \right) \frac{\partial P}{\partial y} \right] - \frac{\partial}{\partial y} \left[-\frac{M\rho^2h^5}{120\mu^2} \left(1 - \frac{31M^2\rho^2h^4}{3024\mu^2} \right) \frac{\partial P}{\partial x} \right] \\ & = -\frac{\partial}{\partial x} \left[\frac{\rho U}{2} \left\{ h - \frac{M^2\rho^2h^5}{120\mu^2} \left(1 - \frac{31M^2\rho^2h^4}{3024\mu^2} \right) \right\} \right] \\ & - \frac{\partial}{\partial y} \left[\frac{M\rho^2U}{2} \left\{ -\frac{h^3}{12\mu} \left(1 - \frac{17M^2\rho^2h^4}{1680\mu^2} \right) \right\} \right] \end{aligned} \quad (2.1)$$

Where x , y and z are coordinates, ρ is the fluid density, U is the sliding velocity, P is the pressure, μ is the viscosity and W^* is fluid velocity in z -direction. If the bearing is infinitely short, then the pressure gradient in x -direction is too small than the pressure gradient in y -direction. In y -direction the gradient $\partial p/\partial y$ is of the order of (P/L) and in x -direction, and is of order of (P/B) . If $L \ll B$ then

$$\frac{P}{L} \gg \frac{P}{B}, \text{ so } \frac{\partial P}{\partial x} \ll \frac{\partial P}{\partial y} \quad (2.2)$$

Then the terms containing $\frac{\partial P}{\partial x}$ can be neglected as compared to the terms $\frac{\partial P}{\partial y}$ containing in the expanded form of Generalized Reynolds Equation. Then in view of (48) the equation (47) takes the form:

$$\begin{aligned} & \frac{\partial}{\partial y} \left[-\frac{h^3}{12\mu} \left(1 - \frac{17M^2\rho^2h^4}{1680\mu^2} \right) \rho \frac{\partial P}{\partial y} \right] + \frac{\partial}{\partial x} \left[-\frac{M\rho^2h^5}{120\mu^2} \left(1 - \frac{31M^2\rho^2h^4}{3024\mu^2} \right) \frac{\partial P}{\partial y} \right] \\ & = -\frac{\partial}{\partial x} \left[\frac{\rho U}{2} \left\{ h - \frac{M^2\rho^2h^5}{120\mu^2} \left(1 - \frac{31M^2\rho^2h^4}{3024\mu^2} \right) \right\} \right] \\ & - \frac{\partial}{\partial y} \left[\frac{M\rho^2U}{2} \left\{ -\frac{h^3}{12\mu} \left(1 - \frac{17M^2\rho^2h^4}{1680\mu^2} \right) \right\} \right] \end{aligned} \quad (2.3)$$

Taking

$$h=h(x), U=-U, P=P(y) \quad (2.4)$$

The film thickness ' h ' and ' y ' can be taken as:

$$h=C(1+\cos \theta), x=R\theta \quad (2.5)$$

Where θ being measured from x -direction.

For the determination of pressure the boundary conditions are as follows:

$$P=0, y=\pm \frac{L}{2} \quad (2.6)$$

III. Determination of Pressure

The solution of the differential equation (2.3) under the boundary condition (2.6) gives the pressure for infinite short journal bearing under the effects of second order rotatory theory of hydrodynamic lubrication as follows:

$$P = \frac{\rho U M e \sin \theta (L^2 y - 4y^3)}{8CR(1 + e \cos \theta)^2} + \frac{U e \sin \theta \rho^2 C^2}{16R(1 + e \cos \theta)} \left[\frac{e^2 M^2 \sin^2 \theta}{R^2} \left(y^4 - \frac{L^2 y^2}{2} + \frac{L^2}{16} \right) - \frac{53M^2}{35} (1 + e \cos \theta)^2 \left(y^2 - \frac{L^2}{4} \right) \right] \frac{1}{\mu} + \frac{(3U e \sin \theta)(L^2 - 4y^2)}{4C^2(1 + e \cos \theta)^3 R} \mu \quad (3.1)$$

On neglecting the term containing M^2 in equation (3.1), we get the pressure equation for first order rotatory theory of hydrodynamic lubrication i.e., as given

$$P = \frac{\rho M U e \sin \theta (L^2 y - 4y^3)}{8CR(1 + e \cos \theta)^2} + \frac{(3U e \sin \theta)(L^2 - 4y^2)}{4C^2(1 + e \cos \theta)^3 R} \mu \quad (3.2)$$

On neglecting the term containing M^2 and M in equation (3.2), we get the pressure equation for classical Reynolds theory of hydrodynamic lubrication i.e., as given

$$P = \frac{(3U e \sin \theta)}{C^2(1 + e \cos \theta)^3 R} \left(\frac{L^2}{4} - y^2 \right) \mu \quad (3.3)$$

IV. Numerical Analysis

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

$e=0.2$, $C=0.0067$ cm., $\theta=30^\circ$, $y=50$ cms., $L=200$ cms., $\rho_{\text{Aerosil}} = 1.7$ gm./cm³ (1.6 gm./cm³ $\leq \rho_{\text{Aerosil}} \leq 1.9$ gm./cm³) [27-28], $h=0.00786$ cm., $M = 0.1$, $U=10^2$ cm./sec.; the calculated values of pressure with respect to μ by taking $M=0.1$ are given by table-1.

The Aerosil nanolubricants have been used with viscosity 0.1 gm/cm. sec. at 40°C, 0.077 gm/cm. sec. at 60°C and 0.055 gm/cm. sec. at 80°C with a 5% of volume fraction i.e., semi diluted lubricant [8] of nanolubricants [27-28]. The nondimensional pressure is calculated using eq. (3.1). The variation of pressure (P) of nanoparticles additive couple stress fluid lubricated for porous journal bearing with respect to viscosity are shown in the table-1 and the graphical variation is shown by figure-3.

Table-1: Pressure Distribution with respect to viscosity

μ	0.055	0.077	0.1
P for second order rotation; $M=0.1$	6.13056 X 10 ⁷	8.17003 X 10 ⁷	10.30220 X 10 ⁷
P for first order rotation; $M=0.1$	6.13056 X 10 ⁷	8.17003 X 10 ⁷	10.30220 X 10 ⁷
P for classical solution; $M=0.0$	5.09867 X 10 ⁷	7.13813 X 10 ⁷	9.27031 X 10 ⁷

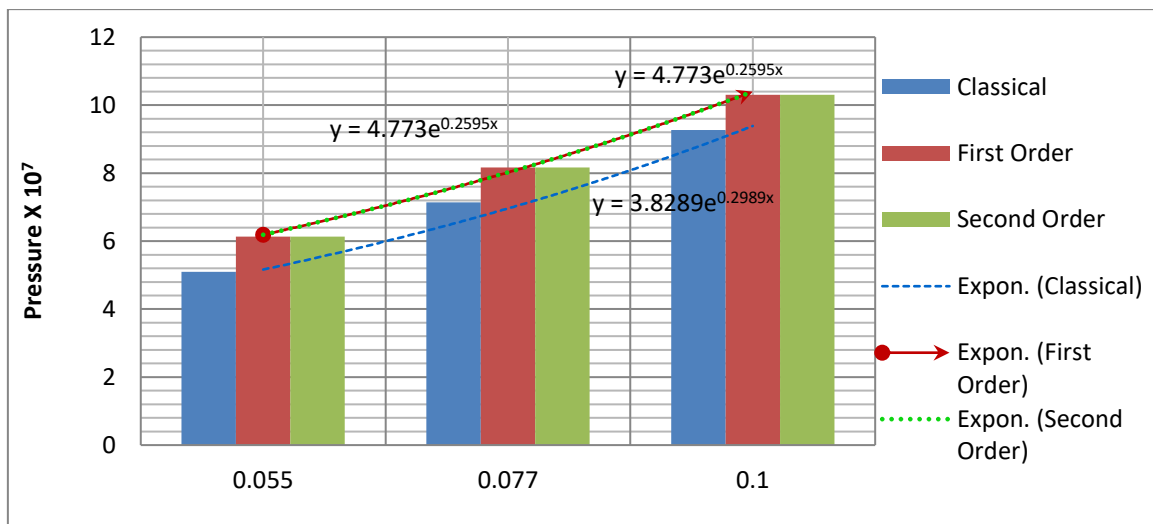


Figure-3. Variation of pressure with respect to μ with exponential trendline

V. Conclusion

The non-dimensional pressure distribution is obtained from integration of Extended Generalized Reynolds Equation underneath the impact of second order rotation by victimization the Reynolds boundary conditions with nanolubricant. The nanolubricant Aerosil is employed for the analysis with 5% volume fraction of nanoparticles. The fluid film properties and structure considerably influence the thin film fluid mechanics lubrication for the nanolubricants underneath the impact of rotation. For the rotatory theory the pressure is

$P = 4.773 e^{0.259\mu} \times 10^7$, whereas the classical pressure is $P = 3.828 e^{0.298\mu} \times 10^7$. The results reveal a rise in pressure distribution of bearings operative on nanoparticles dispersions as compared to plain lubricants. The pressure distribution will increase with the viscosity of the nanofluids. The results obtained are excellent as compared to plain lubricants. The change in pressure for rotatory theory is 20.24% at 80°C, 14.45% at 60°C and 11.13% at 40°C as comparison to classical pressure theory with nanolubricant.

VI. References

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