Load Capacity for Porous Journal Bearing with Nanolubricant under Second Order Rotation

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Abstract

The analysis of infinitely short porous bearing is conferred supported the concept of journal bearings and takes into thought of fluid film thickness and viscosity of nanoparticles additive fluid. In the present paper, the results of additives on the nanolubricants have been investigated and according, aiming for improvement and also the development of hydrodynamic bearing performance. The nanolubricant Aerosil is used with 3%, 3.5%, 4% and 4.5% of volume concentration and viscosity was considered as mingling achieved in oil based hydrophilic Aerosil 200 and numerous hydrophobic Aerosil types. A dimensionless Load Capacity with viscosity over the fluid film has been calculated. The results show an increase on the Load Capacity in response to increase in the viscosity of the additives with percent volume concentration and relevance the rotation number (M). The non-dimensional Load Capacity is obtained from integration of Pressure Equation under the impact of second order rotation by using the Reynolds boundary conditions. The fluid film properties and structure significantly influence the film fluid lubrication for the nanolubricants with respect to the impact of second order rotation.

Keywords: Bearing Load Capacity; Nanolubricants; Porous journal bearing, Second order rotation.

2010 Mathematics Subject Classification: 76D08

Significance Statement:

The hydrodynamic analysis of the porous bearing with nanolubricants is analyzed in this analysis. The Load Capacity of the bearing primarily depends upon the viscosity of the fluid used. The addition of the nanoparticles with the base fluid may enhance the viscosity of lubricating fluid and in turn changes the performance characteristics of the motion of fluid. The analysis of porous bearing takes into consideration of fluid film thickness and viscosity of nanoparticles additive fluid under the effect of second order rotatory theory of hydrodynamic lubrication. The non-dimensional load capacity is obtained from integration of Pressure Equation by using the Reynolds boundary conditions. The results reveal an increase in the load capacity of bearings operative on nanoparticles dispersions as compared to plain lubricants. It increases with the viscosity of the nanofluids and also with the volume concentration. The results obtained are wonderful as compared to plain lubricants.

1. Introduction

Over the previous couple of years, several researchers are focusing on the result of nanolubricant in hydrodynamic bearing. Nanoparticles in the lubricating oil considerably reduce the coefficient of friction and will increase load bearing capability. Form of lubrication mechanisms are planned for nanoparticle suspension in oil like bearing result, protecting film, mending result and sharpening results (Shah et al. 2015). In general, for low loading, the losses owing to friction and wear are negligible on contact surfaces. However beneath serious loading, the fluid film generated between the bearing and journal are going to be intensely squeezed, and should not be ready to defend contact surfaces. Some researches disclosed that nanoparticle additives enhance the supporting forces. Therefore, friction and wear are often reduced by adding nanoparticle to the material. Several studies are being found to review the properties of nanolubricant and their application within the past decade. Varied sorts of nanoparticles are used as material additives and their performance in thin film lubrication are studied (Li et al. 2011), (Rico et al. 2007), (Peng et al.2007). Of these studies showed reduced wear and friction in tribo-surfaces with the utilization of nanoparticle material additives. However, there's a serious gap in analysis relating to theoretical simulation of nanolubricant in fluid film bearing. Offered literature suggests improvement in static and dynamic performance characteristics by considering solely variable viscosity analysis of nanoparticle additive (Wu et al. 2007), (Shenoy et al. 2012), (Yathish et al. 2015).

In the thin film lubrication, the effective viscosity of adsorbed layer below relative motion is way larger than the majority Newtonian consistence. The adsorbed molecular layer thickness and viscosity are the foremost essential factors in thin film lubrication. The analysis on the viscosity 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 (Einstein, 1906). Brinkman contributed to extend the Einstein equation considering the impact of moderate spherical nanoparticles suspensions in viscous fluids (Brinkman, 1952). Batchelor planned the second order formula for the body of nanofluids considering pedesis impact of spherical nanoparticles additives and their interactions. Every these models predict viscosity of nanoparticles suspensions from the base fluid viscosity and volume fraction of nanoparticles (Batchelor, 1977). Krieger and Dougherty derived the shear body equation considering particle concentrations (Krieger et al. 1959).Tichy and Qingwen et al. studied the impact of lubricant molecular structure and developed thin film model of solid surfaces adhered high body layer. Supported thin film lubrication analysis, as body and thickness of adsorbed layer at solid surfaces can increase; results indicate increase in Load Capacity and reduce in coefficient of friction (Tichy, 1995), (Kingwen et al. 1998). Nabhan et al. collectively investigated binary fluid film greased hydrodynamic bearing. Nanoparticles additive lubricators increase the pressure distribution of fluid film bearings as results of increase in lubricant viscosity (Nabhan et al. 1997). Chen et al. derived modified Krieger and Dougherty equation to predict high shear viscosity of nanofluids supported mixture nanoparticles structure considering the results of variable packing fraction (Chen et al. 2007). They conferred classification of nanofluids supported the volume fraction of nanoparticles (V_ϕ) betting on nanoparticles concentration and structure into the followings:

a) dilute ($0 < V_\phi \leq 0.001$),
b) semi-dilute ($0.001 < V_\phi \leq 0.05$),
c) semi-concentrated ($0.05 < V_\phi \leq 0.1$), and
d) concentrated ($V_\phi \geq 0.1$).

Meurisse and Espejel together have given a Generalized Reynolds Equation for a three-layered film model. Analysis of bearing load Capacity and coefficient of friction in three-layered film is influenced by thickness of fluid film (Meurisse et al. 2007). Nair et al. given characteristics of statically loaded bearing operational lubricants with nanoparticles additives (Nair et al. 2009). Duangthongsuk and Wongwises planned correlations for thermal physical phenomenon and viscosity of nanofluids supported experimental results. The viscosity of nanofluids decreases significantly with increasing temperature and can increase with increasing particle volume concentration (Duangthongsuk et al. 2009). Szeri investigated composite film configuration for increase in load capacity (high viscosity lubricant) and scale back in friction (low viscosity lubricant). The composite film bearing consists of higher and lower viscosity fluid layers adjacent to bearing surface and journal surface, severally (Szeri, 2009). Hosseini et al. given an empirical model of viscosity of nanoparticles suspensions i.e., derived from viscosity of the base fluid, particle volume fraction, particle size, properties of the wetter layer, and temperature (Hosseini et al. 2011). Shenoy et al. have given the influence of API-SF engine oil with nanoparticles additives on the characteristics of an externally adjustable statically loaded fluid film bearing (Shenoy et al. 2012). 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 utilized among the analysis. Viscosity models for nanoparticles additive lubricants were developed victimization on the acceptable experimental analysis (Babu et al. 2012).

Mahbubul et al. have given review of theoretical models of viscosity for suspensions and pictured viscosity correlations for volume concentrations, temperature, and particle diameter of nanofluids (Mahbubul et al. 2012).

Beauchamp Tower's exposition of hydrodynamic behavior of journal bearings in 1880's and his observations play some valuable results for Osborne Reynolds to carryout theoretical analysis. This has resulted during an elementary equation for fluid mechanics lubrication. This has provided a powerful foundation and basis for the look of hydro-dynamic lubricated bearings (Reynolds, 1886), (Gopinath et al. 2017).

In his theoretical analysis, Reynolds created the subsequent assumptions:

a) The fluid is Newtonian.
b) The fluid is incompressible.
c) The consistence is constant throughout the film.
d) The pressure doesn't vary within the axial direction.
e) The bearing and journal extend infinitely within the z direction i.e., no lubricating substance flow within the z direction.
f) The film pressure is constant within the y direction. Therefore the pressure depends on the x coordinate solely.
g) The rate of particle of lubricating substance within the film depends solely on the coordinates x and y.
h) The impact of mechanical phenomenon and gravitation is neglected.
i) The fluid expertise laminar flow.

Later Reynolds himself derived an improved version of Reynolds Equation acquainted as: "Generalized Reynolds Equation" (Pinkus, 1961), (Cameron, 1981) that depends on density, viscosity, and film thickness, surface and transverse velocities. The rotation of fluid film concerning the associated axis that lies across the film offers some new ends up in lubrication problems with hydrodynamics. The origin of rotation is commonly derived by some general theorems related to vorticity among the rotating fluid dynamics. The rotation induces a region of vorticity among the direction of rotation of fluid film and so the consequences arising from it are predominant, for large Taylor's number, it ends up in the streamlines becoming confined to plane transversal to the direction of rotation of the film.

The new extended version of "Generalized Equation" is known to be "Extended Generalized Reynolds Equation" (Cameron, 1981) that takes into thought of the implications of the uniform rotation relating to associated axis that lies across the fluid film and depends on the rotation number (M), i.e. the square root of the Taylor's number. The generalization of the classical theory of fluid dynamics lubrication is known as a result of the "Rotatory Theory of Hydrodynamic Lubrication" (Banerjee et al. 1981).

The "First Order Rotatory Theory of Hydrodynamic Lubrication" and thus the "Second Order Rotatory Theory of Hydrodynamic Lubrication" was given by retaining the terms containing up to first and second powers of (M) severally by neglecting higher powers of (M) (Banerjee et al. 1982).

The present paper analyzes relating to the pressure within the porous bearings with reference to the impact of second order rotation. These bearings are created by porous material and thus the stuff flows out of the bearing surface with a particular speed. These bearings are usually utilized in many useful devices, like vacuum cleaners, extractor fans, motor automobile starters, hair dryers etc. These bearings are infinitely short in nature. The geometrical description of the bearings is delineating in figure-1 (Miyan, 2013), of solid shaft in mildew metal bush and of shaft and bush opened up.

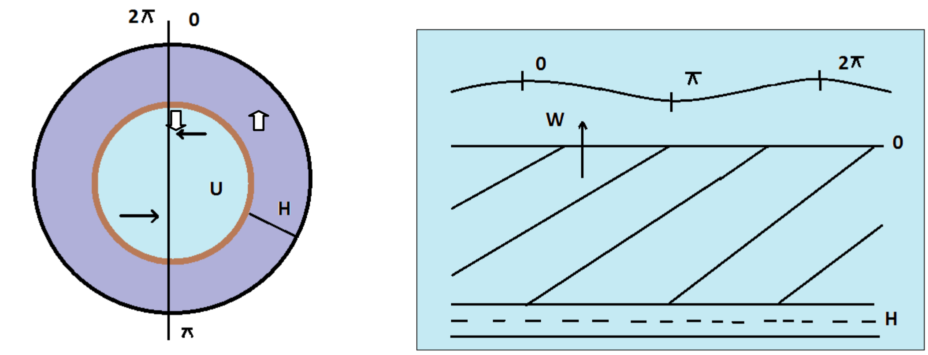

Figure (1.1) *Geometry of Porous Bearing*

1.1 Additive Aerosil Types

The Aerosil i.e., treated silica; conjointly known as pyrogenic compound as results of its created throughout a flame, consists of microscopic droplets of amorphous compound fused into branched, chainlike, three-dimensional secondary particles that then agglomerate into tertiary particles. The following powder has particularly low bulk density and high surface area. Its three-dimensional structure winds up in viscosity-increasing, thixotropic behavior once used as a material or reinforcing filler. Aerosil contains a very strong thickening result. Primary particle size is 5–50 nm. The particles are non-porous and have a locality of 50–600 m^2/g. The density is 160–190 kg/m^3. The Aerosil suspensions provide higher viscosities over others (Mondragón et al. 2012). The kinds of Aerosil nanolubricant i.e., Aerosil 200, Aerosil R274, Aerosil R972 and Aerosil R805 are used for this analysis. The viscosity of the nanofluids will increase with reference to the percent volume concentration (Evonik, 2017).

2. Governing Equations and Boundary Conditions

If the bearing is infinitely short, then the pressure gradient in x-direction is far smaller than the pressure gradient in y-direction. In y-direction the gradient $\partial P/\partial y$ is of the order of (P/L) and at intervals the x-direction, and is of order of (P/B). If $L << B$ then $P/L >> P/B$, so $\partial P/\partial x << \partial P/\partial y$ Then the terms containing $\partial P/\partial x$ is also neglected as compared to the terms $\partial P/\partial y$ containing at intervals the distended sort of Generalized Reynolds Equation.

Where;

B: Total breadth of bearing parallel to the direction of motion; *L*: Bearing length traditional to the direction of motion; *P*: Pressure; *x*: Co-ordinate on span of the bearing system; *y*: Co-ordinate on length of the bearing system.

The “Extended Generalized Reynolds Equation” with relevance second order turn theory of hydrodynamic lubrication is written as:

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

Where;

C : Radial clearance; D : Diameter of the bearing; h : Film thickness; M : Rotation number; P : Pressure; R : Radius of the bearing; U : Sliding velocity ; W^*: Fluid velocity in z-direction; μ : Absolute viscosity of fluid; ρ : Density of fluid.

Taking

$$h=h(x),\ U=-U,\ P=P(y) \text{ and } W^* = -\partial P/\partial z\big|_{z=0}\ \phi/\mu \tag{2.2}$$

Where; z: Co-ordinate across the fluid film; $\partial P/\partial z$ is the pressure gradient at the bearing surface; and ϕ is the property called permeability, which varies with porosity and size of pores. From the requirements of continuity, we have for the porous matrix;

$$\phi/\mu\ \nabla W^* = \nabla^2 P=0 \text{ i.e., } \nabla^2 P=0 \tag{2.3}$$

The problem then is to solve the governing equation (2.1) for the pressures in lubricant film simultaneously with that of Laplace for the porous matrix with a common pressure gradient $\partial P/\partial z$ at the boundary, we have

$$\frac{\partial^2 P}{\partial x^2}+\frac{\partial^2 P}{\partial y^2}+\frac{\partial^2 P}{\partial z^2}=0 \tag{2.4}$$

We have two assumptions to solving the equations (2.1) and (2.4) as follows:

a) The bearing is infinitely short.
b) $\partial P/\partial z$ is linear across the matrix and is zero at the outer surface of the porous bearing shell.

From (2.2), (2.3), we have

$$\frac{\partial^2 P}{\partial x^2}=0,\ \frac{\partial^2 P}{\partial z^2}=K \text{ (constant)},\ \frac{\partial^2 P}{\partial y^2}=-K \tag{2.5}$$

From (2.2), we have

$$\frac{\partial P}{\partial z}\Big|_{Z=0} = KH = \frac{\partial^2 P}{\partial y^2}\Big|_{Z=0} H \tag{2.6}$$

Where; H: Wall thickness of porous bearing.

Now the equation (2.1) becomes

$$\left[-\frac{h^3}{12\mu}\left(1-\frac{17M^2\rho^2h^4}{1680\mu^2}\right)\rho\right]\frac{d^2P}{dy^2}+\left[\frac{M\rho^2}{120\mu^2}\frac{d}{dx}\left(h^5-\frac{31M^2\rho^2h^9}{3024\mu^2}\right)\right]\frac{dP}{dy}=\frac{d}{dx}\left[\frac{\rho U}{2}\left\{h-\frac{M^2\rho^2h^5}{120\mu^2}\left(1-\frac{31M^2\rho^2h^4}{3024\mu^2}\right)\right\}\right]-\rho\left(-\frac{dP}{dz}\Big|_{z=0}\frac{\phi}{\mu}\right) \tag{2.7}$$

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

$$h=C\ (1+e\cos\theta),\ y=R\ \theta \tag{2.8}$$

Where; θ: Angular co-ordinates (bearing angle), θ being measured from x-direction; e: Eccentricity ratio.

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

P=0, $y= \pm L/2$ (2.9)

The solution of the differential equation (2.1) under the boundary condition (2.9) gives the pressure for porous bearing as follows:

$$P = \left[\begin{array}{c} \left(3\frac{CU\alpha}{4Rh^3}\right)\mu + \left(\frac{3KH\phi}{h^3} + \rho C\alpha U\frac{yM}{8Rh^2}\right) + \left(\frac{KH\phi yM}{2Rh^2} + 53U\frac{\rho^2 C\alpha h}{2240R}M^2\right)\frac{1}{\mu} \\ -\frac{17KH\phi\rho^2 hM^2}{560\mu^2}\frac{1}{\mu^2} \end{array} \right](L^2 - 4y^2) \quad (2.10)$$

3. Load Capacity

The load capacity for porous bearing is given by

$$W = \sqrt{W_x^2 + W_y^2}$$

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

$$W_x = -2\int_0^{\pi}\int_0^{L/2} P\, Cos\theta\, R\, d\theta\, dy$$

$$W_y = 2\int_0^{\pi}\int_0^{L/2} P\, Sin\theta\, R\, d\theta\, dy$$

The W_x and W_y in the increasing values of M are given by

$$W_x = -\frac{\mu U e^2 L^3}{C^2(1-e^2)^2} - \frac{KH\phi R\pi e L^3}{(1-e^2)^{\frac{5}{2}}} + \frac{\rho CU}{64}\left\{\frac{1}{e}log\frac{1+e}{1-e} - \frac{2}{1-e^2}\right\}L^4 M + 4\frac{\rho CKH\phi}{16\mu}\left\{\frac{1}{e}log\frac{1+e}{1-e} - \frac{2}{1-e^2}\right\}L^4 M$$
$$+\frac{106Ue^2\rho^2C^2L^3M^2}{13440\mu} - \frac{204RCKH\phi\rho^2\pi eL^3M^2}{13440\mu^2} \quad (3.1)$$

$$W_y = \frac{\mu U\pi e L^3}{4C^2(1-e^2)^{\frac{3}{2}}} + \frac{4KH\phi RL^3}{(1-e^2)^2} + \frac{U\pi e\rho CL^4 M}{128(1-e^2)^{\frac{3}{2}}} + \frac{4KH\phi\pi e\rho CL^4 M}{128\mu(1-e^2)^{\frac{3}{2}}}$$

$$+\frac{53U\rho^2C^2\pi eL^3M^2}{4480\mu} - \frac{272RCKH\phi\rho^2L^3M^2}{4480\mu^2} \quad (3.2)$$

4. Result and Discussions

The values of different mathematical terms are taken in C.G.S. system and are as follows:

$\rho_{Aerosil}$ = 1.7 gm. /cm^3 ($1.6\ gm./cm^3 \leq \rho_{Aerosil} \leq 1.9\ gm./cm^3$), e = 0.5, L =2 cm., ϕ=0.0025 cm., C=0.0067 cm., h=0.00786 cm., H=0.05 cm., M = 0.1, R=3.35 cm., U=10^2 cm./sec. , y=1 cm.

The differences in the viscosity of mineral oil (PKWF 4/7 from Dow, printing ink oil) with respect to percent volume concentration obtained with hydrophilic Aerosil 200 and various hydrophobic Aerosil types (Evonik, 2017); are shown in the table (4.1) and figure (4.1).

The nondimensional Load Capacity is calculated using eqs. (3.1) and (3.2). The variation of Load Capacity (W) of nanoparticles additive couple stress fluid lubricated for porous journal bearing with respect to viscosity (μ) are shown in the table (4.2) and figure (4.2) with respect to (M) =0.1.

Table (4.1)

Volume Concentration of nanolubricants %; (V_ϕ)		3%	3.5%	4%	4.5%
Viscosities of Nanolubricants in mineral oil (μ)	Aerosil 200	40 P	200 P	320 P	380 P
	Aerosil R274	30 P	60 P	110 P	190 P
	Aerosil R972	20 P	40 P	50 P	90 P
	Aerosil R805	20 P	20 P	20 P	20 P

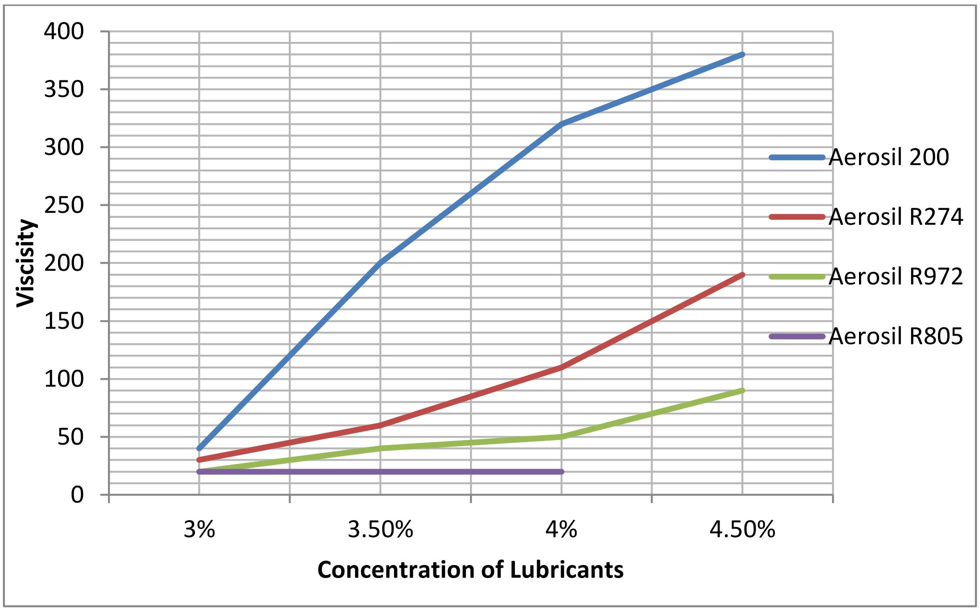

Figure (4.1): *Viscosity increase achieved in mineral oil using hydrophilic Aerosil 200 and various hydrophobic Aerosil types* [35]

Table (4.2)

Volume Concentration of Nanolubricants %; (V_ϕ)		3%	3.5%	4%	4.5%
Load Capacity (W) x10^8	Aerosil 200	06.5595	32.7975	52.4759	62.3152
	Aerosil R274	04.9196	09.8392	18.0386	31.1576
	Aerosil R972	03.2797	06.5595	08.1994	14.7589
	Aerosil R805	03.2797	03.2797	03.2797	03.2797

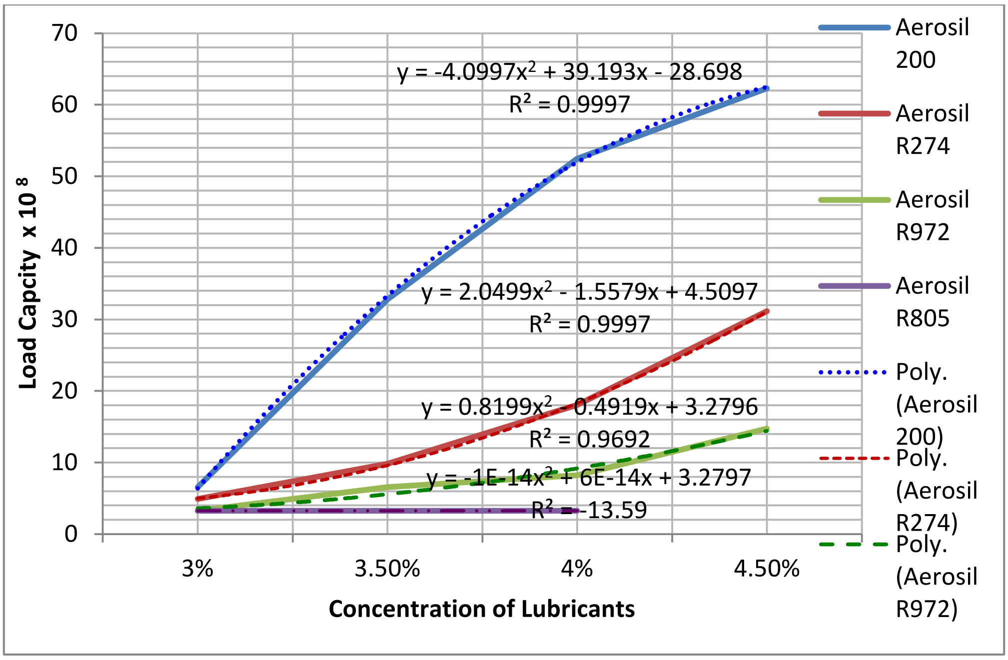

Figure (4.2) *Variation of Load Capacity for concentration % of nanolubricants in mineral oil using hydrophilic Aerosil 200 and various hydrophobic Aerosil types with polynomial trend lines*

The graph (4.2) shows that the Load Capacity increases for increasing the percent volume concentration and also with viscosity of nanolubricants. The results show that the Aerosil 200 has much viscosity at 4.5% volume concentration, so has much load carrying capacity with the polynomial trend line $y = -4.009\ x^2+39.19\ x-28.69;\ R^2=0.999$; as compared to other Aerosil types. The results are excellent as compared to the analysis of bearing with normal lubricants.

5. Conclusions

The dynamic analysis of the hydrodynamic porous bearing operative with nanolubricants is analyzed throughout this analysis. The Load Capacity of the bearing primarily depends upon the viscosity of the material used. The addition of the nanoparticles in the base fluid may enhance the viscosity of lubricating fluid and in turn changes the performance characteristics. The analysis of porous bearing i.e., infinitely short is given supported the speculation of journal bearings and takes into thought of fluid film layer's thickness and viscosity of nanoparticles additive fluid and also the rotation number. The fluid film layers in porous bearing are assumed to be Newtonian. The non-dimensional load capacity is obtained from integration of Pressure Equation under the effect of the second order rotation by using the Reynolds boundary conditions. The assorted sorts of Aerosil nanolubricant is used for the analysis with 3%, 3.5%, 4% and 4.5% of volume concentration of nanoparticles. The fluid film properties and structure significantly influence the thin film hydrodynamic lubrication for the nanolubricants under the effect of the second order rotation. Results reveal an increase in the load capacity of bearings operative on nanoparticles dispersions as compared to plain lubricants. The Load capacity will increase with the viscosity of the nanofluids and also with the volume concentration. The results obtained are wonderful as compared to plain lubricants.

6. Suggestions and Scope for Future Work

- Further study is created on the intense Load Capacity characteristics by using the nanolubricant.
- Conjointly optimum nanoparticle yet because the dispersant to be further added within the lubricant so as to get better performance.
- Study is created with the addition of the dispersant or the surface-active agent.
- Totally different base oils and nanoparticles are studied with constant procedure so as to improve the wear resistance properties.

- Any study is created on the categories of different nanoparticles to be accustomed compare the lubrication performance.

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