Vol. 46, No. 2 (2024) 198-209, DOI: 10.24874/ti.1476.04.23.10

Tribology in Industry

www.tribology.rs

Numerical Investigation of Short Bearing Lubricated with Copper Oxide Nanolubricants

Abhijeet G. Chavan[a,*] and Y.P. Reddy[a]

[a]*Department of Mechanical Engineering, Sinhgad College of Engineering, Vadgaon, Pune, Maharashtra, India*

Keywords:

Short journal bearings
Copper oxide (CuO) nanolubricants
Finite difference method
Couple stress parameter

* Corresponding author:

Abhijeet G. Chavan
E-mail:
abhijeeetchavan101@gmail.com

Received: 24 April 2023
Revised: 25 July 2023
Accepted: 1 October 2023

A B S T R A C T

Performance of short hydrodynamic journal bearing lubricated with Copper oxide (CuO) nanoparticle suspensions in base oil (nanolubricants) is investigated numerically. Dimensionless pressure, load carrying capacity and friction parameter are calculated by numerical solution of the modified Reynolds equation by means of finite difference method. The particular values of couple stress parameter for nanolubricants are evaluated and used for finding pressure distribution. Modified Krieger Dougherty equation is used to predict the viscosity of nanolubricant. Two short bearing configurations with aspect ratio 0.25 and 0.5 are analyzed by evaluating the pressure profile, load carrying capacity and friction parameter. The comparative results between plain lubricants and nanolubricants are presented. The results reveal improved pressure and load carrying capacity accompanied by decrease in friction parameter of short bearing. The shorter bearing configuration with aspect ratio of 0.25 shows better results than the others.

1. INTRODUCTION

The demand for sophisticated machinery capable of performing exceptionally well and efficiently in diverse operating conditions has led to the research towards reducing the friction and stabilizing high speed machine components. Bearings are key machine elements which account for smoother and efficient operation of machinery. Hydrodynamic journal bearings found their place in particular applications where higher reliability and continuous operation is the major requirement. A carefully designed bearing supports the rotating shaft by formation of hydrodynamic lubricant film in the clearance space and thus prevents metal to metal contact. The challenging task of improving the performance of journal bearing has been undertaken by many researchers and optimum values of clearance, bearing dimensions and geometry, operating speeds are already identified.

One of the ways to enhance the performance of journal bearings further is by altering the properties of lubricating oils. One of the methods to improve properties like viscosity, friction and wear performance of lubricant is addition of nanoparticles in lubricant. Nanolubricants are the suspensions obtained when base lubricants are suspended with small fractions of nanoparticles.

Many researchers [1-3] have revealed that addition of ultrafine particles in lubricants can lead to better performance of hydrodynamic journal bearing. In recent times, metal oxide nanoparticles like zinc oxide (ZnO), silicon dioxide (SiO_2), aluminum oxide (Al_2O_3), titanium dioxide (TiO_2), Copper oxide (CuO) are preferred to be used as additives, because of their chemically neutral behavior towards lubricating oil and as a result they can provide a stable colloidal suspension of nanolubricant [4]. For preparing nanolubricants, Tarasov et al. [5] added copper oxide nanoparticles in commercially available SAE 30 motor oil. The friction and wear performance tests were conducted on the prepared nanolubricant suspensions. They reported prominent reduction in friction at operating conditions involving higher values of speeds and loads. Wu et al. [6] performed experiments for assessment of rheology of nanolubricants prepared by suspending nano-diamond, TiO_2 and CuO nanoparticles in engine oil and CuO is found to have preeminent anti wear and anti friction characteristics amongst other additives. Ingole et al. [7] conducted friction and wear tests on TiO_2 based nanolubricants where mineral oil is used as base lubricant. They found reduced coefficient of friction and wear because of abrasive action. Many studies reported improved performance of journal bearings which are operating with nanolubricants. Binu et al. [8] and Suryawanshi and Pattiwar [9] examined the static characteristics of journal bearing lubricated with TiO_2 nanolubricant and revealed significant rise in load carrying capacity of journal bearing. Solghar [10] reported higher load carrying capacity and friction coefficient of journal bearings using Al_2O_3 based nanolubricants. Mahdi et al. [11] used the couple stress fluid model for studying cavitation effects in bearing. Sadabadi and Nezhad [12] simulated the journal bearing used in heavy machinery which is lubricated with Tungsten disulfide (WS_2) based nanolubricants and reported a maximum of 20% rise in load carrying capacity. Dang et al. [13] assessed the thermal performance of elliptical journal bearing which is lubricated with TiO_2 and CuO nanoparticle based nanolubricants.

Majority of these studies are concerned with finite journal bearings and square bearings. When the aspect ratio $\lambda \ll 1$, the bearings are identified as short journal bearings. For general practical applications, the aspect ratio λ of short bearing ranges from 0.2 to 0.8. The short bearings consume less space and further the misalignment effects are lower. The short bearings are preferred in crucial applications like internal combustion engines, aerospace applications, rocker arms etc. because of their superior heat dissipation ability as compared to long bearing. There are very few studies on performance of short bearings operating on nanolubricants. Further, majority of the studies are based on application of modified viscosity of nanolubricants. These studies have not considered the interaction of nanoparticles with each other and walls of bearing and shaft. These interactions can be included in the analysis by using the couple stress approach. In view of this, further analysis of short journal bearing lubricated with nanolubricants is essentially required.

The present study attempts to evaluate the performance of short bearing configurations with reference to the non dimensional values of pressure, load carrying capacity and friction parameter. The short bearing configurations under consideration are lubricated with nanolubricants. Although short bearing approximation is preferred for performance evaluation of short journal bearing, this study uses the two dimensional form of modified Reynolds equation, which does not neglect circumferential pressure gradient. Stokes couple stress theory [14] provides the model for anticipating the flow of fluids containing particles, which is used for predicting the flow of nanolubricants in the journal bearing. The couple stress parameter for nanolubricant suspensions is calculated for different aggregate sizes of CuO nanoparticles in the base oil. The dimensionless pressure distribution and load carrying capacity in case of short bearing configuration is evaluated by using the calculated couple stress parameter values of nanolubricants.

2. THEORY

Hydrodynamic journal bearings are broadly classified as long bearings, square bearings and short bearings, they work on the hydrodynamic lubrication principle. The load carrying capacity is developed by eccentrically rotating the journal, which pumps the lubricating fluid in the wedge shaped region between bearing and journal. The schematic diagram of a typical short journal bearing is shown in Fig. 1. In this diagram x, y and z represents Cartesian coordinates while R, θ represents polar coordinates. The classical Reynolds equation is presented in generalized form by Dowson [15]. For a steady state, finite journal bearing using incompressible fluid as lubricant the equation is given by (1)

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\mu U\frac{\partial h}{\partial x} \tag{1}$$

The equation (1) is in the two dimensional form where change in pressure along the y direction that is, along film thickness is neglected. In equation (1), p is the pressure developed in the oil film, h is the film thickness, μ is the dynamic viscosity of the lubricant and U is the linear velocity of journal.

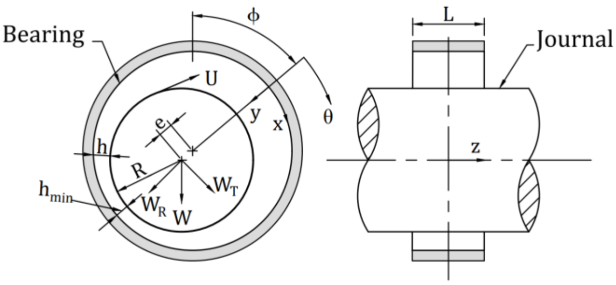


Fig. 1. Schematic of hydrodynamic journal bearing.

For short bearing application, the equation (1) is modified and solved by Dubois and Ocvirk [16] by assuming the circumferential pressure gradient as negligible i.e. $\frac{\partial p}{\partial x}=0$, which gives the final form of equation for a short journal bearing as

$$\frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right)=6\mu U\frac{\partial h}{\partial x} \tag{2}$$

The pressure distribution is obtained by solving equation (2), which is given by the expression

$$p=\frac{3\mu U}{RC^2}\left(\frac{L^2}{4}-z^2\right)\frac{\varepsilon\sin\theta}{(1+\varepsilon\cos\theta)^3} \tag{3}$$

In equation (3), the length of bearing is given by L, the distance from center plane of the bearing is z, radius of journal is R, eccentricity ratio is ε and the radial clearance between bearing and journal is represented by C. From pressure distribution equation, the overall load carrying capacity in terms of radial and tangential force components is calculated by integrating the pressure values over the area of bearing considering the positive pressure values. The vector addition of radial and tangential force components gives the load capacity of the bearing W and attitude angle ϕ which are given in equation (4) and (5).

$$W=\frac{\mu UL^3}{4C^2}\frac{\varepsilon}{(1-\varepsilon^2)^2}\sqrt{[\pi^2(1-\varepsilon^2)+16\varepsilon^2]} \tag{4}$$

$$\phi=\tan^{-1}\left[\frac{\pi}{4}\frac{\sqrt{1-\varepsilon^2}}{\varepsilon}\right] \tag{5}$$

This bearing theory developed by Dubois and Ocvirk is suitable only for short bearing applications. In present work, modified two dimensional form of Reynolds equation is used which considers the variation of pressure in circumferential direction. The dimensionless load capacity obtained by short bearing theory and modified two dimensional Reynolds equation is compared for validation purpose in section 5.3.

3. VISCOSITY OF NANOLUBRICANT

The viscosity of lubricant is one of the key parameters for pressure generation in case of hydrodynamic journal bearing. Majority of conventional lubricants are Newtonian but numerous studies reveal non-Newtonian behavior of nanolubricants [17]. Thus conventional approach of analyzing Newtonian lubricant needs to be modified for solving the Reynolds equation. There are multiple theories for addressing the viscosity of fluid containing ultrafine particles. Modified Krieger-Dougherty model proposed by Chen et al. [18] is used by many researchers [11,13,19] for predicting the viscosity of nanolubricants. The viscosity model presented by Selvakumar et al. [20] is developed by particle size distribution and it is used for predicting the effective viscosity of nanoparticle suspensions. This model simulates the viscosity of suspension by aggregation and interfacial layer formation characteristics. The current work also employs modified Krieger- Dougherty model given by Chen et al. [18] for relative viscosity $\bar{\mu}$ of nanofluids, described as

$$\bar{\mu}=\frac{\mu_n}{\mu_b}=\left(1-\frac{\phi}{\phi_m}\left(\frac{a_a}{a}\right)^{3-D}\right)^{-\eta\phi_m} \tag{6}$$

The equation (6) is valid for the suspension where volume fraction of nano sized particles is between 0.001 and 0.05 in the base fluid, which is termed as semi dilute suspension. In equation (6), D is fractal index, which denotes the degree to which the packing percentage varies from the centre to the edge of the aggregates. Fractal index of nanoparticles is determined by two dimensional imaging of nanoparticles using techniques like dynamic light scattering or electron microscopy. In the present work, typical value of fractal index is taken as 1.8, which is reported by [17-19,21]. Intrinsic viscosity η is a dimensionless value representing the capability of an additive to enhance the viscosity of fluid. For nanofluid with hard spherical suspended particles, this value is 2.5. At high shear rates,the maximum value of particle packing fraction ϕ_m is 0.605 [22]. After inserting the values of fractal index, intrinsic viscosity and particle packing fraction in the equation (6), the modified Krieger Dogherty equation takes the following form

$$\bar{\mu} = \frac{\mu_n}{\mu_b} = \left(1 - \frac{\emptyset}{0.605}\left(\frac{a_a}{a}\right)^{1.2}\right)^{-1.5125} \quad (7)$$

The equation (7) is used in present analysis for estimating the relative viscosity $\bar{\mu}$ of nanolubricants. Change in the viscosity of nanolubricants due to aggregation effect in the case of CuO nanoparticles is addressed by Kole and Dey [23]. They reported the dimension of average aggregate particle as around 7.15 times the size of primary nanoparticle using dynamic light scattering analysis. This value is adopted in current research as the cluster of aggregates of CuO nanoparticles remains unchanged after application of shear stress [23].

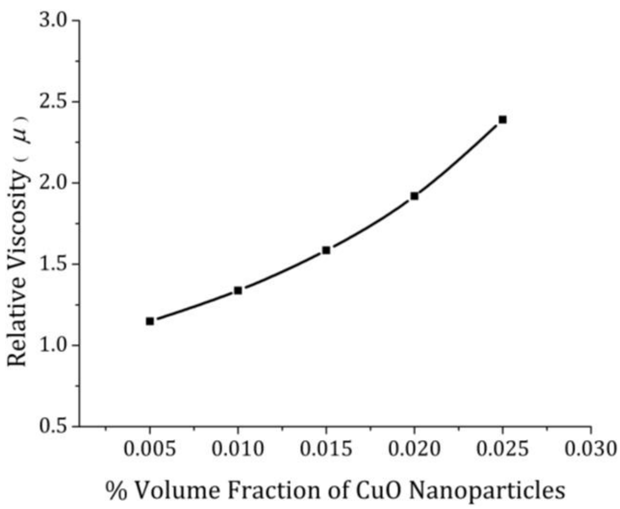


Fig. 2. Relative viscosity of nanolubricant containing CuO nanoparticles.

The average particle size (APS) and average size of the aggregates of particular nanoparticles plays a crucial role in deciding the effective viscosity of nanofluids. Fig. 2 depicts the values of relative viscosities for average particle size range of 20 to 200 nm at different volume fraction of CuO nanoparticles. The dimensionless relative viscosity values are estimated for volume fraction of 0.005, 0.01, 0.015, 0.02 and 0.025 using equation (7) which is based on modified Krieger- Dougherty approach. The commercially available CuO nanoparticle sizes vary from 20 nm to 200 nm. Majority of the researchers have reported the average particle size ranging between 40 to 80 nm.

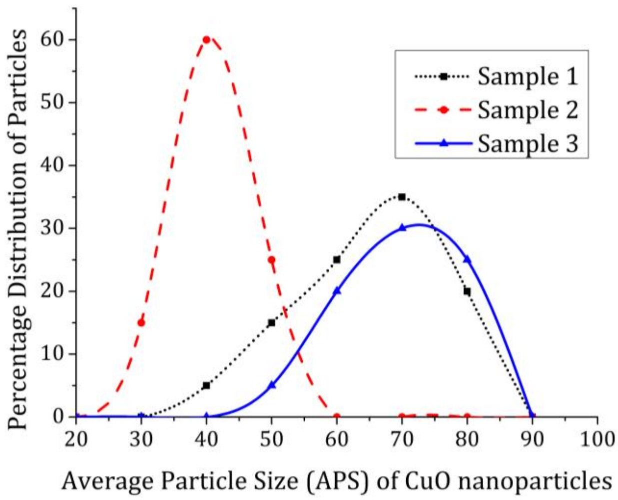


Fig. 3. APS of commercially available CuO nanoparticles.

In Fig. 3 average particle size of CuO nanoparticles is presented for three different commercially available samples. The CuO nanoparticles having an average particle size of 40 nm and 80 nm is chosen for the study in this research.

4. COUPLE STRESS MODEL

The modified Krieger-Dougherty model predicts effective viscosity based on volume fraction and dimensions of aggregates. Although the accuracy of the model is good, this model cannot be used directly for hydrodynamic bearing application. In addition, mutual collision of nanoparticles and collision with the walls of bearing and shaft must be considered. Stokes couple stress theory can be utilized effectively to address this issue. According to stokes theory, couple stress parameter l is defined as [14]

$$l = \sqrt{\frac{\eta_0}{\mu_b}} \quad (8)$$

Here η_0 is material constant and μ_b is viscosity of base fluid. The dimensionless form of the couple stress parameter is expressed by using the bearing clearance value C [19]. The dimensionless couple stress parameter is defined as

$$\bar{l} = \frac{l}{C} \quad (9)$$

For CuO nanoparticles having average aggregate size of 500, 570 and 775 nm, dimensionless couple stress parameter $\bar{l}$ is calculated by using equation (9), as 0.025, 0.0285 and 0.039 respectively. These values are adopted in present analysis for including the effect of CuO nanoparticles in lubricating oils. The selected values provide more realistic approach as they are based on experimental data [19,22,23].

5. NUMERICAL PROCEDURE

Many researchers have [1,19,24-26], reported finite difference method (FDM) as suitable method for solving the two dimensional Reynolds equation because of its accuracy, simplicity and faster convergence. Therefore, FDM is adopted for numerical solution of Reynolds equation modified for couple stress flow of lubricant containing copper oxide nano particles. In the following section, modified Reynolds equation and its non dimensional form are presented which are governing equations for hydrodynamic journal bearing.

5.1 Governing equations

The Reynolds equation for fluids containing ultrafine particles is expressed as the modified form of equation (1) provided by Lin [27]

$$\frac{\partial}{\partial x}\left(f(h,l)\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(f(h,l)\frac{\partial p}{\partial z}\right)=6\mu U\frac{\partial h}{\partial x} \quad (10)$$

Here, $f(h,l)$ is a function of film thickness h and couple stress parameter l, given by

$$f(h,l)=h^3-12l^2\left\{h-2l\tanh\left(\frac{h}{2l}\right)\right\} \quad (11)$$

The equation (10) is used for both plain lubricant and nanolubricants by adopting specific values of the couple stress parameter l. For plain lubricant, $l=0$, which implies no additives in the base lubricant, all other terms in the equation (11) will become zero except h^3 thus giving the classical Reynolds equation for plain lubricants.

In the first step of solution, modified Reynolds equation is brought to non-dimensional form by using the following dimensionless variables [27]

$$\theta=\frac{x}{R},\qquad \bar{z}=\frac{z}{L},\qquad \lambda=\frac{L}{D},\qquad \varepsilon=\frac{e}{C},$$

$$\bar{p}=\frac{pC^2}{\mu UR},\quad \bar{h}=\frac{h}{C},\quad \bar{l}=\frac{l}{C} \quad (12)$$

Here θ and $\bar{z}$ are dimensionless bearing coordinates, dimensionless value of pressure is given by $\bar{p}$, the dimensionless film thickness is $\bar{h}$. After introducing these dimensionless terms in equation (10), we get modified Reynolds equation in non dimensional form

$$\frac{\partial}{\partial\theta}\left(\bar{f}\left(\bar{h},\bar{l}\right)\frac{\partial\bar{p}}{\partial\theta}\right)+\frac{1}{4\lambda^2}\frac{\partial}{\partial\bar{z}}\left(\bar{f}\left(\bar{h},\bar{l}\right)\frac{\partial\bar{p}}{\partial\bar{z}}\right)=6\frac{d\bar{h}}{d\theta} \quad (13)$$

Equation (13) is solved numerically for getting the dimensionless pressure of short hydrodynamic bearing. The particular advantage of using this modified dimensionless form of Reynolds equation is, the same equation can be used for plain lubricant and nanolubricants by adopting various specific calculated values of the couple stress parameter $\bar{l}$. Further, load carrying capacity of the bearing, expressed as $\bar{W}$, is calculated by integrating pressure term. For this pressure values which are acting on shaft are considered. The two components of $\bar{W}$ are calculated, equation (14) and (15) represents the radial $\bar{W}_R$ and tangential $\bar{W}_T$ components.

$$\bar{W}_R=\bar{W}\cos\psi=-\int_0^1\int_0^{\theta_c}\bar{p}\cos\theta\,d\theta\,d\bar{z} \quad (14)$$

$$\bar{W}_T=\bar{W}\sin\psi=\int_0^1\int_0^{\theta_c}\bar{p}\sin\theta\,d\theta\,d\bar{z} \quad (15)$$

The total dimensionless load carrying capacity $\bar{W}$ is calculated by the adding two components $\bar{W}_R$ and $\bar{W}_T$ and attitude angle ϕ is given by equation (16) and (17) respectively.

$$\bar{W}=\sqrt{(\bar{W}_R)^2+(\bar{W}_T)^2} \quad (16)$$

$$\phi=\tan^{-1}\left(\frac{\bar{W}_T}{\bar{W}_R}\right) \quad (17)$$

The friction force F_f is evaluated by integrating the shear stress about the circumferential area of journal. Equation (18) gives non dimensional values of friction force [19].

$$\bar{F}_f=\int_0^1\int_0^{2\pi}\left[\frac{\bar{\mu}}{\bar{h}}+\frac{\bar{h}}{2}\frac{\partial\bar{p}}{\partial\theta}-\bar{l}\,\frac{\partial\bar{p}}{\partial\theta}\tanh\left(\frac{\bar{h}}{2\bar{l}}\right)\right]d\theta\,d\bar{z} \quad (18)$$

Friction can be conveniently expressed in the form of dimensionless friction parameter $\bar{f}$ which is normalized coefficient of friction and generally used in the design tables of journal bearing and is given in equation (19)

$$\bar{f}=\frac{\bar{F}_f}{\bar{W}} \quad (19)$$

For solving modified Reynolds equation (10), the following boundary conditions for pressure and pressure gradient are defined

$$\bar{p} = 0 \text{ at } \bar{z} = 0 \text{ and } 1 \tag{20a}$$

$$\bar{p} = 0 \text{ at } \theta = 0 \tag{20b}$$

$$\frac{\partial \bar{p}}{\partial \theta} = 0 \text{ at } \theta = \theta_c \tag{20c}$$

$$\bar{p} = 0 \text{ at } \theta_c \leq \theta \leq 2\pi \tag{20d}$$

Equations (20a) and (20b) represent the pressure boundary conditions for the finite journal bearing operating on incompressible lubricant. Equations (20c) and (20d) are used for addressing the cavitation in journal bearing. Using these conditions, negative pressures are set to zero, as per the algorithm provided by Christopherson [28], thus neglecting the cavitation effect.

5.2 Solution methodology

The two dimensional Reynolds equation can have exact solution by using either long bearing assumption or short bearing assumption. In long bearing assumption, bearing is considered as infinitely long and in short bearing assumption, it refers to infinitely short. The numerical methods are an efficient way for solving the Reynolds equation, where long or short bearing assumption is not required. In recent times, with advanced computational facilities, faster and more accurate solutions are possible. The dimensionless form of the modified Reynolds equation (13) is discretized by means of central difference scheme in this research. A MATLAB code is prepared to solve the equation using Gauss-Seidel successive over relaxation scheme. The lubricant film domain is divided into a mesh grid of $m \times n$ points along θ and $\bar{z}$ respectively. Fig. 4 shows the grid mesh used.

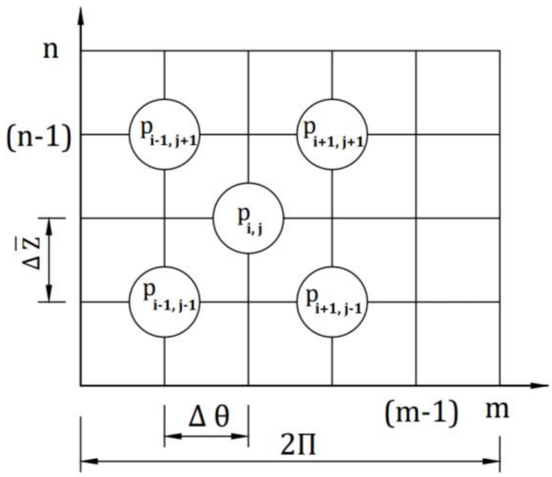


Fig. 4. Schematic of mesh grid adopted for oil film.

The values of $m = 90$ and $n = 48$ are used by considering the number of iterations and grid independency. The values of m and n are optimum for solution accuracy and minimum number of iterations, and decided by conducting exhaustive calculations for a standard case by using different combinations of m, n and over relaxation factor. The iterations are performed until the predefined convergence condition of 0.00001 is satisfied. The pressure distribution obtained is integrated to calculate the dimensionless value of load capacity $\bar{W}$ and attitude angle ϕ.

The pressure distribution and load capacity is calculated by considering $\bar{l}$ as zero for plain lubricant, whereas $\bar{l}$ is taken as 0.025, 0.0285 and 0.039 for calculating couple stress parameter in nanolubricants as described in section 4.

5.3 Validation

The finite difference algorithm is used for solution of modified Reynolds equation. A provision is made to include couple stress effect in the form of dimensionless couple stress parameter in the modified Reynolds equation.

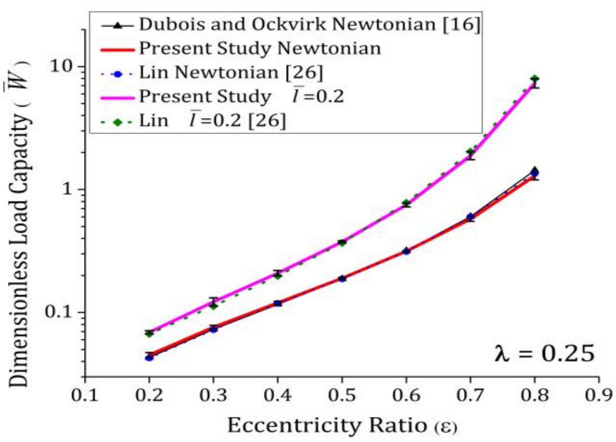


Fig. 5. Dimensionless load carrying capacity of various lubricants for λ=0.25.

The computed results are validated by comparing with the results depicted by Dubois and Ocvirk [16] for Newtonian lubricants and with Lin [27] for couple stress fluid. The dimensionless values of pressure profile and load carrying capacity are plotted in Fig. 5 which represents bearing configuration with aspect ratio as 0.25. The results obtained by numerical solution of modified Reynolds equation are compared with literature results and they are in good agreement, hence the validity of the solution of the modified equation is justified.

6. RESULTS AND DISCUSSION

For different concentration of CuO nanoparticles, the couple stress parameter is calculated by considering the aggregate sizes as 500, 570 and 775 nm and clearance of bearing as 20 microns [24]. Three dimensionless couple stress values of 0.025, 0.0285 and 0.039 are used for analysis. The pressure distribution for short bearing configuration $\lambda = 0.25$ and $\lambda = 0.5$ is calculated by using the MATLAB program developed in this research. The dimensionless pressure is integrated for obtaining the value of non dimensional load carrying capacity. The dimensionless load carrying capacity and friction parameter $\bar{f}$ of bearing lubricated with nanolubricants, at various eccentricity ratios is evaluated and compared with the values of bearing operating on plain oil. Calculated values of dimensionless load carrying capacity and friction parameter are shown in Table 1 and Table 2. It is observed that variation in load carrying capacity is more significant at higher eccentricity ratios because the smaller wedge shaped region at higher eccentricity ratios results in higher interaction of nanoparticles, whereas highest dimensionless load carrying capacity is observed for $\bar{l}$= 0.039 which represents the proportional relationship between couple stress parameter and pressure development due to improved viscosity.

Table 1. Dimensionless load carrying capacity $\bar{W}$ and friction parameter $\bar{f}$ for λ= 0.5.

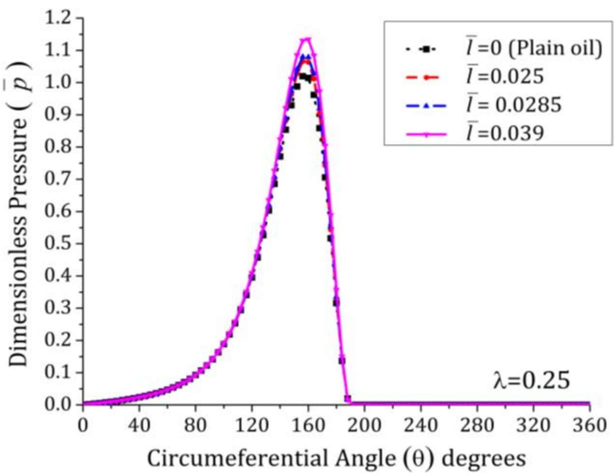
Eccentricity Ratio (ε)	Dimensionless Parameters							
	$\bar{l}$= 0.0 (Plain Oil)		$\bar{l}$= 0.025		$\bar{l}$=0.0285		$\bar{l}$= 0.039	
	$\bar{W}$	$\bar{f}$	$\bar{W}$	$\bar{f}$	$\bar{W}$	$\bar{f}$	$\bar{W}$	$\bar{f}$
0.2	0.167	38.25	0.176	36.411	0.176	36.326	0.178	36.012
0.3	0.279	23.508	0.292	22.396	0.293	22.335	0.296	22.113
0.4	0.432	15.704	0.452	14.982	0.454	14.932	0.46	14.751
0.5	0.666	10.685	0.694	10.245	0.697	10.201	0.709	10.041
0.6	1.05	7.256	1.074	7.085	1.081	7.042	1.105	6.884
0.7	1.81	4.632	1.83	4.58	1.848	4.535	1.913	4.374
0.8	3.686	2.624	3.902	2.471	3.972	2.425	4.231	2.268
0.9	10.791	1.156	13.124	0.931	13.727	0.886	15.978	0.752

Table 2. Dimensionless load carrying capacity $\bar{W}$ and friction parameter $\bar{f}$ for λ=0.25.

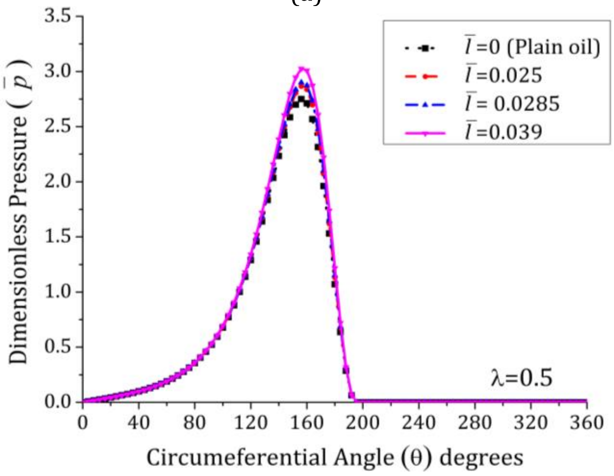
Eccentricity Ratio (ε)	Dimensionless Parameters							
	$\bar{l}$= 0.0 (Plain Oil)		$\bar{l}$= 0.025		$\bar{l}$= 0.0285		$\bar{l}$= 0.039	
	$\bar{W}$	$\bar{f}$	$\bar{W}$	$\bar{f}$	$\bar{W}$	$\bar{f}$	$\bar{W}$	$\bar{f}$
0.2	0.045	141.944	0.047	134.737	0.0478	134.42	0.048	133.251
0.3	0.076	86.661	0.08	82.366	0.08	82.141	0.081	81.309
0.4	0.12	56.893	0.127	53.968	0.127	53.784	0.129	53.11
0.5	0.19	37.927	0.201	35.982	0.201	35.819	0.205	35.225
0.6	0.316	24.649	0.334	23.321	0.336	23.168	0.344	22.617
0.7	0.577	15.017	0.605	14.316	0.612	14.163	0.636	13.621
0.8	1.294	7.884	1.416	7.193	1.446	7.042	1.557	6.53
0.9	4.66	2.927	6.038	2.235	6.401	2.104	7.743	1.728

Increase in couple stress parameter $\bar{l}$ results in reduction of dimensionless friction parameter $\bar{f}$. At lower eccentricity ratios, both friction parameter and load carrying capacity are changed by insignificant amount. At higher eccentricity ratios, boost in load carrying capacity is observed while friction parameter is decreased. At higher eccentricity ratio wedge shaped region is smaller and there is more interaction between the nanoparticles which results in increased pressure and load capacity.

For the bearing, working at eccentricity ratio of 0.7, the dimensionless pressure profile at centre plane of the short bearing against the bearing angle θ is shown in Fig. 6 (a) and (b). The peak value of dimensionless pressure is obtained when $\bar{l}$ is at 0.039. The attitude angle ϕ is also changing with change in couple stress parameter, but the difference is very small and insignificant in present case.



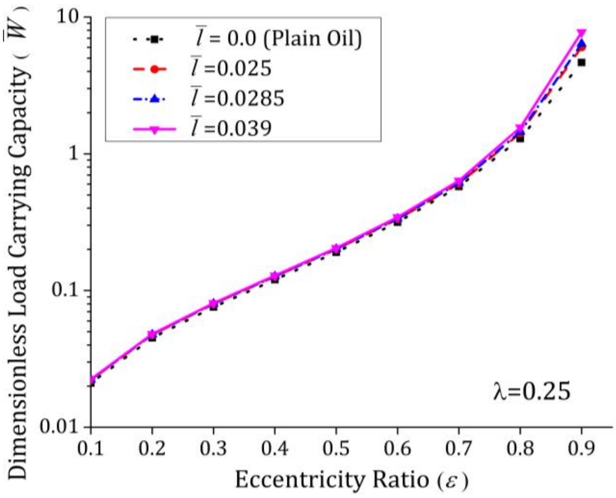
(a)



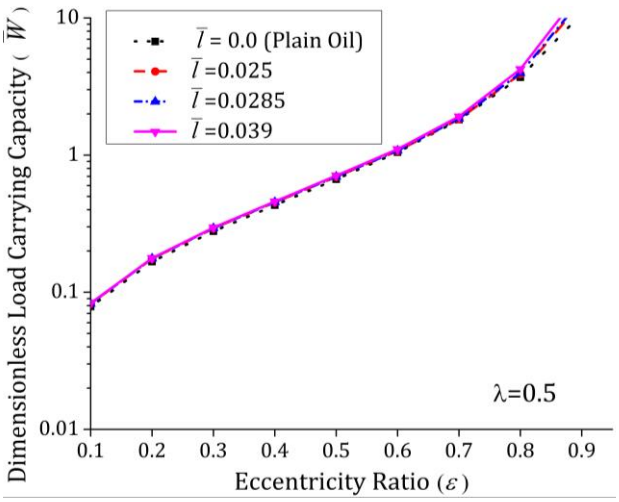
(b)

Fig. 6. Dimensionless pressure at centre plane of bearing for eccentricity ratio 0.7 a) λ =0.25; b) λ=0.5.

Fig. 7 (a) and (b) shows the gradual rise in dimensionless load carrying capacity of short bearing in consequence of increased couple stress parameter. The variation effect in load carrying capacity is very small at low eccentricity ratio and it is more dominant at higher eccentricity ratio. For quantitative investigation and performance comparison of plain oil with nanolubricants, the percentage rise in the dimensionless load carrying capacity of short bearing operating on nanolubricants over plain lubricant is plotted for various eccentricity ratios.



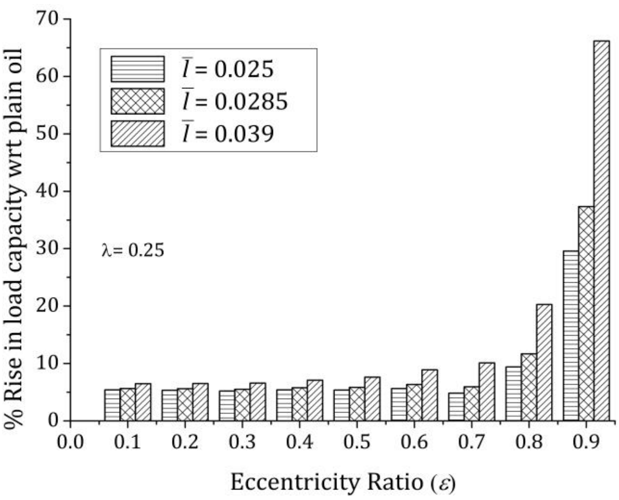
(a)



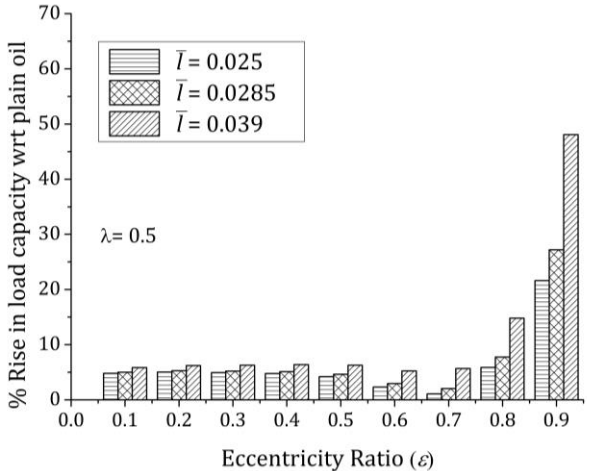
(b)

Fig. 7. Dimensionless load capacity at various eccentricity ratios (a) λ =0.25; (b) λ=0.5.

Fig. 8 (a) and (b) shows percentage rise in the dimensionless load carrying capacity of bearings for aspect ratio 0.25 and 0.5 respectively.

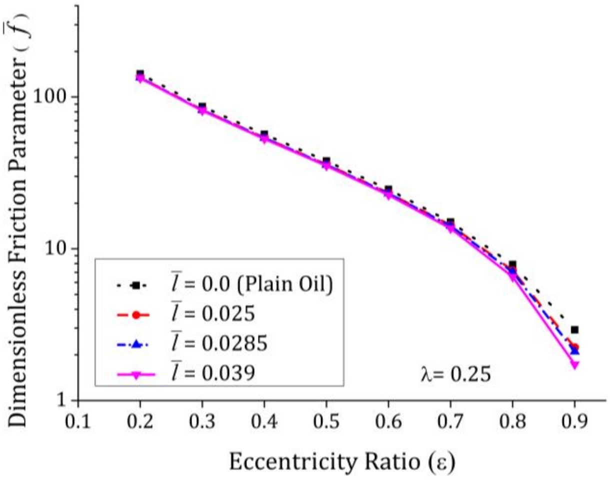


(a)

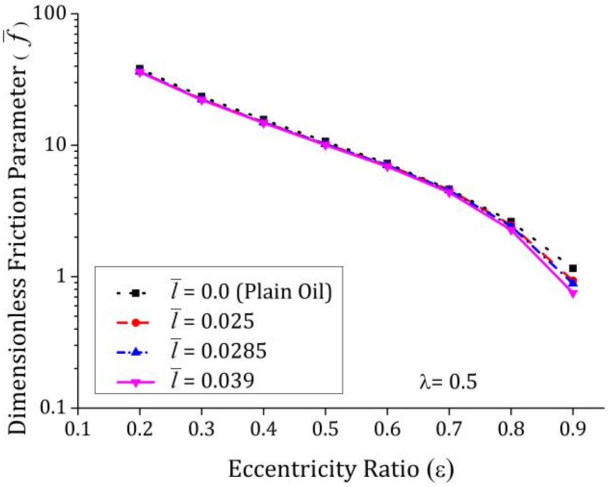


(b)

Fig. 8. Percentage rise in the load carrying capacity as compared to plain oil for various eccentricity ratios (a) λ =0.25; (b) λ=0.5.



(a)

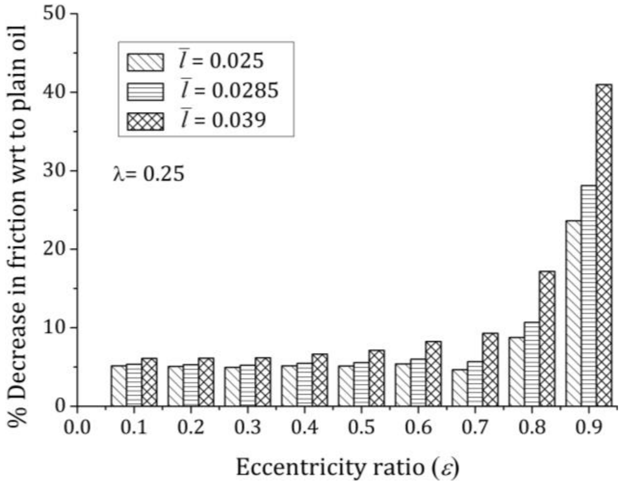


(b)

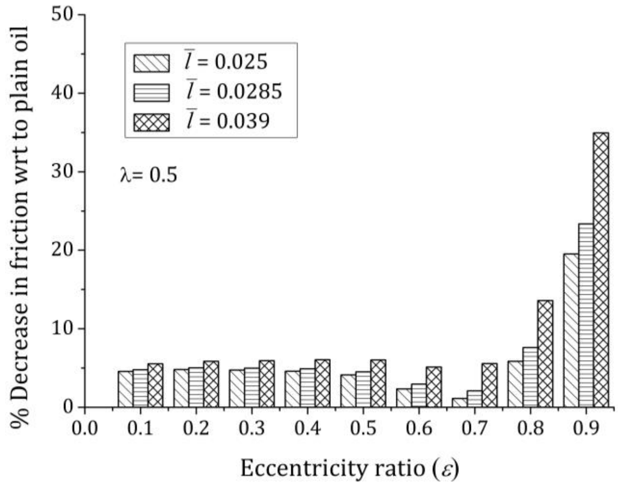
Fig. 9. Dimensionless friction parameter at various eccentricity ratios.

Both the configurations shows exponential rise in the load carrying capacity with increased eccentricity ratio. For the Bearing with aspect ratio $\lambda = 0.5$, maximum rise of 46.5% and for $\lambda = 0.25$ maximum rise of 61.33% in dimensionless load capacity is found. This variation is attributed to improved viscosity of nanolubricants in comparison with plain oil.

Dimensionless value of the friction parameter against eccentricity ratio is plotted for various lubricants for both bearing configurations $\lambda = 0.25$ and $\lambda = 0.5$.



(a)



(b)

Fig. 10. Percentage decrease in friction parameter as compared to plain oil for various eccentricity ratios.

As shown in Fig. 9 (a) and (b), for both bearing configurations, dimensionless friction parameter decreases with increase in eccentricity ratio and couple stress parameter. The variation effect is distinct at higher eccentricity ratio. Fig. 10 (a) and (b) shows the graph for percentage decrease in dimensionless friction parameter at specific values of the couple stress parameter with respect to the plain lubricant. Maximum decrement of 39% for $\lambda = 0.25$ and 33.91% for

$\lambda = 0.5$ is obtained for dimensionless friction parameter. As a result of introduction of couple stress parameter for short journal bearing, load carrying capacity is increased and friction is reduced for both the configurations. The effects are small and insignificant at low eccentricity ratio, but significant when bearing is operating at higher eccentricity ratio. Further, in case of load capacity and friction parameter, the configuration $\lambda = 0.25$ gives better results than $\lambda = 0.5$.

7. CONCLUSION

The numerical investigation of short journal bearing lubricated with nanolubricants is carried out by considering Reynolds 2D equation instead of short bearing approximation. The couple stress parameter is introduced to get the modified form of Reynolds equation. This couple stress parameter accounts for nanoparticle addition in the base lubricant. The specific values of the couple stress parameter are evaluated from the average particle size of commercially available CuO nanoparticles. A MATLAB program for the finite difference solution of the modified Reynolds equation is written. The dimensionless values of pressure, load carrying capacity and friction parameter are calculated for two short bearing configurations having aspect ratio of 0.25 and 0.5.

The numerical investigation shows a significant rise in the load carrying capacity of short bearing when operating on nanolubricants in comparison to a case where plain lubricant is used. There is rise in non-dimensional oil film pressure which results in increased load carrying capacity which ranges between 2% to 61.3 %. The dimensionless friction parameter has significant reduction in case of nanolubricants with maximum reduction in friction as 39% for $\lambda = 0.25$. The improvement in load carrying capacity is because of improved viscosity of lubricant which is enhanced by added nanoparticles and reduced friction is because of rolling action of nanoparticles in the small wedge shaped region of bearing. The CuO nanoparticle suspensions in the base oil (nanolubricants) show a superior load carrying capacity than plain oil and can be considered as potential substitutes for plain oils. The improvement in both friction parameter and load carrying capacity is better for smaller aspect ratios. In order to get more insights in this research, other characteristics like temperature -viscosity and pressure- viscosity relationship, thermal capacity of nanolubricants, needs to be addressed during the analysis.

REFERENCES

[1] M. M. Khonsari and D. E. Brewe, "On the Performance of Finite Journal Bearings Lubricated with Micropolar Fluids," *Tribology Transactions*, vol. 32, no. 2, pp. 155–160, Jan. 1989, doi: 10.1080/10402008908981874.

[2] S. Das, S. K. Guha, and A. K. Chattopadhyay, "Linear stability analysis of hydrodynamic journal bearings under micropolar lubrication," *Tribology International*, vol. 38, no. 5, pp. 500–507, May 2005, doi: 10.1016/j.triboint.2004.08.023.

[3] A. Kornaev, L. Savin, E. Kornaeva, and A. Fetisov, "Influence of the ultrafine oil additives on friction and vibration in journal bearings," *Tribology International*, vol. 101, pp. 131–140, Sep. 2016, doi: 10.1016/j.triboint.2016.04.014.

[4] W. K. Shafi and Charoo, "An overall review on the tribological, thermal and rheological properties of nanolubricants," *Tribology - Materials, Surfaces & Interfaces*, vol. 15, no. 1, pp. 20–54, Jun. 2020, doi: 10.1080/17515831.2020.1785233.

[5] S. Yu. Tarasov, A. V. Kolubaev, S. A. Belyaev, M. I. Lerner and F. Tepper, "Study of friction reduction by nanocopper additives to motor oil," *Wear*, vol. 252, no. 1–2, pp. 63–69, Jan. 2002, doi: 10.1016/s0043-1648(01)00860-2.

[6] Y. Wu, W. C. Tsui, and T. C. Liu, "Experimental analysis of tribological properties of lubricating oils with nanoparticle additives," *Wear*, vol. 262, no. 7–8, pp. 819–825, Mar. 2007, doi: 10.1016/j.wear.2006.08.021.

[7] S. Ingole, A. Charanpahari, A. Kakade, S. S. Umare, D. V. Bhatt, and J. Menghani, "Tribological behavior of nano TiO_2 as an additive in base oil," *Wear*, vol. 301, no. 1–2, pp. 776–785, Apr. 2013, doi: 10.1016/j.wear.2013.01.037.

[8] K. R. Yathish, K. G. Binu, B. Shenoy, D. S. Rao, and R. Pai, "Study of TIO_2 nanoparticles as lubricant additive in Two-Axial Groove Journal Bearing," *Zenodo (CERN European Organization for Nuclear Research)*, Oct. 2014, doi: 10.5281/zenodo.1096705.

[9] S. R. Suryawanshi and J. T. Pattiwar, "Effect of TiO_2 Nanoparticles Blended with Lubricating Oil on the Tribological Performance of the Journal Bearing," *Tribology in Industry*, vol. 40, no. 3, pp. 370–391, Sep. 2018, doi: 10.24874/ti.2018.40.03.04.

[10] A. A. Solghar, "Investigation of nanoparticle additive impacts on thermohydrodynamic characteristics of journal bearings," *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, vol. 229, no. 10, pp. 1176–1186, Mar. 2015, doi: 10.1177/1350650115574734.

[11] M. A. Mahdi, A. Hussain, and H. H. Hadwan, "Investigation of cavitation in a finite journal bearing considering the journal speed and couple stress fluids effects," *Tribology in Industry*, vol. 40, no. 4, pp. 670–680, Dec. 2018, doi: 10.24874/ti.2018.40.04.13.

[12] H. Saberi and A. S. Nezhad, "Nanofluids for Performance Improvement of Heavy Machinery Journal Bearings: A Simulation study," *Nanomaterials*, vol. 10, no. 11, p. 2120, Oct. 2020, doi: 10.3390/nano10112120.

[13] R. K. Dang, A. Chauhan, and S. S. Dhami, "Static thermal performance evaluation of elliptical journal bearings with nanolubricants," *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, vol. 235, no. 8, pp. 1627–1640, Nov. 2020, doi: 10.1177/1350650120970742.

[14] V. K. Stokes, *Theories of fluids with microstructure: An introduction*, Springer Science & Business Media, Berlin/Heidelberg, Germany, 2012.

[15] D. Dowson, "A generalized Reynolds equation for fluid-film lubrication," *International Journal of Mechanical Sciences*, vol. 4, no. 2, pp. 159–170, Mar. 1962, doi: 10.1016/s0020-7403(62)80038-1.

[16] G. B. DuBois and F. W. Ocvirk, "The short bearing approximation for plain journal bearings," *Transactions of the American Society of Mechanical Engineers*, vol. 77, no. 8, pp. 1173–1178, Nov. 1955, doi: 10.1115/1.4014634.

[17] M. Mahbubul, R. Saidur, and A. M. Afifi, "Latest developments on the viscosity of nanofluids," *International Journal of Heat and Mass Transfer*, vol. 55, no. 4, pp. 874–885, Jan. 2012, doi: 10.1016/j.ijheatmasstransfer.2011.10.021.

[18] H. Chen, Y. Ding, and C. Tan, "Rheological behaviour of nanofluids," *New Journal of Physics*, vol. 9, no. 10, p. 367, Oct. 2007, doi: 10.1088/1367-2630/9/10/367.

[19] K. G. Binu, B. Shenoy, D. S. Rao, and R. Pai, "Static characteristics of a fluid film bearing with TiO_2 based nanolubricant using the modified Krieger–Dougherty viscosity model and couple stress model," *Tribology International*, vol. 75, pp. 69–79, Jul. 2014, doi: 10.1016/j.triboint.2014.03.013.

[20] R. D. Selvakumar and S. Dhinakaran, "Effective viscosity of nanofluids — A modified Krieger–Dougherty model based on particle size distribution (PSD) analysis," *Journal of Molecular Liquids*, vol. 225, pp. 20–27, Jan. 2017, doi: 10.1016/j.molliq.2016.10.137.

[21] X. Wang, K. Zhu, and S. Wen, "Thermohydrodynamic analysis of journal bearings lubricated with couple stress fluids," *Tribology International*, vol. 34, no. 5, pp. 335–343, May 2001, doi: 10.1016/s0301-679x(01)00022-6.

[22] M. Kole and T. K. Dey, "Enhanced thermophysical properties of copper nanoparticles dispersed in gear oil," *Applied Thermal Engineering*, vol. 56, no. 1–2, pp. 45–53, Jul. 2013, doi: 10.1016/j.applthermaleng.2013.03.022.

[23] M. Kole and T. K. Dey, "Effect of aggregation on the viscosity of copper oxide–gear oil nanofluids," *International Journal of Thermal Sciences*, vol. 50, no. 9, pp. 1741–1747, Sep. 2011, doi: 10.1016/j.ijthermalsci.2011.03.027.

[24] K. S. S. Babu, K. P. Nair, and P. K. Rajendrakumar, "Computational analysis of journal bearing operating under lubricant containing Al_2O_3 and ZnO nanoparticles," *International Journal of Engineering, Science and Technology &B. Lagos*, vol. 6, no. 1, pp. 34–42, Feb. 2014, doi: 10.4314/ijest.v6i1.4.

[25] M. K. Ghosh, B. C. Majumdar, and M. Sarangi, *Fundamentals of fluid film lubrication*, McGraw Hill Professional, New York, United States, 2014.

[26] R. Ishii, R. Miwa, N. Miyanaga, and J. Tomioka, "Effect of Cross-Sectional profile of circular dimples on hydrodynamic lubrication characteristics of thrust bearings," *Journal of Materials and Engineering*, vol. 1, no. 1, pp. 21–23, Jan. 2023, doi: 10.61552/jme.2023.01.003.

[27] J. Lin, "Effects of couple stresses on the lubrication of finite journal bearings," *Wear*, vol. 206, no. 1–2, pp. 171–178, May 1997, doi: 10.1016/s0043-1648(96)07357-7.

[28] D. G. Christopherson, "A new mathematical method for the solution of film lubrication problems," *Proceedings - Institution of Mechanical Engineers*, vol. 146, no. 1, pp. 126–135, Jun. 1941, doi: 10.1243/pime_proc_1941_146_027_02.

NOMENCLATURE

x, y, z	Bearing coordinates in circumferential, radial and axial directions (m)
$\bar{x}, \bar{y}, \bar{z}$	Non dimensional bearing coordinates
μ_n	Viscosity of nanolubricant (Pa-S)
μ_b	Viscosity of base lubricant (Pa-S)
$\bar{\mu}$	Relative viscosity = (μ_n/μ_b)
Ø	Nanoparticle volume fraction
a_a	Average size of aggregates (nm)
a	Size of nanoparticles (nm)
R	Radius of journal (m)
L	Length of bearing (m)
λ	Aspect ratio = (L/2R)
C	Radial clearance (m)
h	Film thickness (m)
$\bar{h}$	Dimensionless film thickness = (h/C)
U	Velocity of journal (m/s)
$\bar{f}$	Dimensionless Friction parameter (R/C)f
d	Characteristic dimension of additive (m) = $\sqrt{\eta_0/\mu_b}$
η_0	Material constant for couple stress property (N-s)
$\bar{l}$	Dimensionless couple stress parameter = (d/C)
e	Eccentricity (m)
ε	Eccentricity ratio = (e/C)
p	Film pressure (N/m^2)
$\bar{p}$	Dimensionless film pressure
θ	Cylindrical coordinate (rad)
θ_c	Angle at which cavitation starts (rad)
$\bar{W}_R$	Dimensionless radial component of load carrying capacity
$\bar{W}_T$	Dimensionless tangential component of load carrying capacity
$\bar{W}$	Dimensionless load carrying capacity
ϕ	Attitude angle (rad)
$\bar{F}_f$	Dimensionless Friction force