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RESEARCH

Effect of TiO_2 and CuO Based Nanolubricants on the Static Thermal Performance of Circular Journal Bearings

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A B S T R A C T

Circular journal bearings constitute the heart of power generation equipments due to their simple design, ease in manufacturing and high load-carrying capacity. Hydrodynamic action in these bearings is manifested by the presence of a single oil film, which increases the oil film temperature at higher loads. However, in order to improve the performance of bearings and for overall optimization of equipment efficiency, lubricants with specialized additives are being explored. With the advancements in nanotechnology, lubricants have been blended with nanoparticles for enhancing tribological characteristics. The existing studies are more focused on examining the influence of nanolubricants for boundary lubrication regime. Hence, this article investigates the effect of CuO and TiO_2 based nanolubricants in hydrodynamic lubrication regime by studying the static thermal characteristics of circular bearing. Comparative performance analysis has been conducted with both the nanoparticles based lubricants at different concentrations by mixing them in two mineral-based oils having different viscosities. The modified Krieger Dougherty method has been applied for determining the viscosity of nanolubricants. Hydrodynamic pressure and oil film temperature calculations have been performed by developing a model in MATLAB®R2020a software by considering the effect of variation in the thermophysical properties of nanolubricants. There has been a significant increase in load capacity with just small rise in oil film temperature with the use of nanolubricants. Load capacity with 2wt% addition of TiO_2 increased by 14.23% at a journal speed of 2000rpm and eccentricity ratio of 0.6, while there was an increase of 9.23% in the case of CuO at the same parameters. The proposed methodology has an enormous potential for assessing the performance enhancement of turbomachinery used for power generation.

1. INTRODUCTION

Circular journal bearings have been extensively used in numerous applications because of their simple design and high load-carrying capacity. These bearings operate at different journal speeds and result in heat generation due to higher shear rates of the lubricant film. Lubricants play a pivotal role for the dissipation of heat and also help in developing and preserving the oil film. There is a further rise in oil temperature at the higher journal speeds, which in turn decreases the lubricant viscosity or even can cause oil oxidation [1]. Uninterrupted oil film at lower temperatures is the manifestation of journal bearing performance. Once the oil film is ruptured, the hydrodynamic action of bearing ceases and there happens to be physical contact of bearing and journal, resulting in complete failure of bearing. With the advancements in material technology, a wide variety of materials have been specifically explored for journal bearings and in certain instances, bearings are reinforced with liners of specialized materials [2]. However, role of lubricant assumes significant importance when the newly designed and manufactured bearing is put in critical machinery for rigorous applications. Efforts are focused on use of variety of lubricants and further improvement in oil characteristics by the inclusion of certain additives which enhance the antifriction and antiwear characteristics [3]–[5]. In certain applications, there is a need to improve the extreme pressure characteristics or anti-corrosion performance of bearings. No single additive is capable of enhancing all the oil characteristics and the simultaneous addition of different additives may improve one characteristic and degrade others. Static thermal performance of journal bearing need to be studied in detail in light of different types of lubricants and specialized additives at different operating conditions. Synergy of these additives with different type of lubricating oils needs further exploration.

With the advent of nanotechnology, nanoparticles have been explored as lubricant additives for overcoming the drawbacks of traditional antifriction/ antiwear additives [6]–[8]. Due to their small size, nanoparticles are capable of filling the small spaces between matching surfaces and impart special characteristics to the surfaces. Numerous researchers have analyzed the use of nanolubricants and proposed various lubrication mechanisms, which facilitate in improving the tribological characteristics [9]–[11]. Nanoparticles act as small-sized balls between contact surfaces which reduce sliding friction and initiate the mechanism of rolling friction in contact surfaces. Experimental investigations have provided evidence of the formation of tribofilms on mating surfaces which reduces the asperities and results in friction and wear reduction [12], [13]. Another important mechanism, which comes into play with nanolubricants, is; mending effect. Nanoparticles get deposited on worn surfaces and penetrate into the scars and irregularities, which improve the surface quality, and hence this is also termed as a self-repairing effect. Furthermore, the polishing mechanism deals with the smoothing of surfaces with nanoparticles. Nanoparticles of different types and sizes have been used by researchers by blending with different oils for improving the tribological characteristics which can be classified as; carbon-based, metallic elements, oxides, sulphides, rare earth compounds, nano-composites, etc. [6], [14], [15]. Recently, boron nitride [16] has also been used as additive in cutting fluids for improvement in surface finish. Nanoparticles of different sizes and morphologies are added in very small concentrations which reduce the area of contact. However, optimum concentrations of nanoparticles are a function of operating conditions, characteristics of contact surfaces and base lubricant. It has been found that the use of nano fluids is instrumental in improving the heat transfer characteristics when flowing through cavities of different shapes [17]–[19]. Recently, Hatami et al. [20] discussed the use of various nanolubricants for IC engine applications and came out with the conclusion that TiO_2 and Al_2O_3 nanoparticles blended with SAE 40 oil had excellent tribological properties.

Success of using nanolubricants in different test conditions, boundary layer lubrication regime and real-life applications prompted researchers to use nanolubricants in the hydrodynamic lubrication regime for journal bearing applications. Nair et al. [21], [22] studied the Thermohydrodynamic (THD) analysis of journal bearing by separately blending CuO, CeO_2 and Al_2O_3 in 15W40 engine oil. Load capacity and friction force were found to be increased with an increase of nanoparticles concentration. Kalakada et al. [23] also used these three nanoparticles in different concentrations to study the effect of variation in viscosity with change in temperature for monitoring bearing performance. There was an improvement in static

and dynamic performance and CuO based nanolubricant was found to be the best performing lubricant. Nicoletti [24] conducted THD analysis of circular journal bearings by separately adding Cu, Al, Si and their oxides namely, CuO, Al_2O_3 and SiO_2 separately in base lubricants. It had been observed that the volumetric heat capacity of nanoparticles played a major role in bearing performance enhancement and CuO based nanolubricant delivered best results due to maximum volumetric heat capacity. Shenoy et al. [25] theoretically simulated the performance of externally adjustable bearing by mixing three types of nanoparticles, namely diamond, CuO and TiO_2 in engine oil. It was predicted that bearing with negative tilt and radial adjustment, operated with nanolubricant had exhibited higher load capacity as compared to base oil and the best results were obtained with TiO_2 based nanolubricant. Solghar [26] assessed the THD performance for a single grooved journal bearing with pure oil and Al_2O_3 based nanolubricant. The results were presented by mixing 5% volume fraction of nanoparticles, which depicted an improvement in load capacity by about 18% at eccentricity ratio of 0.9. It was further noticed that maximum flow rate reduced by 7.5% as compared to base oil. Suryawanshi and Pattiwar [27] used TiO_2 nanoparticles with 0.5wt.% concentration in three different oils. A comparative performance analysis was done for circular and elliptical bearings using an analytical approach. There was a decrease in temperature for elliptical bearing as compared to circular bearings. Pressure distribution and load capacity increased with the addition of nanoparticles, while there was a reduction in side leakage and oil flow rate. Later, they [28] experimentally investigated the performance of circular and elliptical bearings with three types of base oils and also with 0.5wt% of TiO_2 nanoparticles blended in these oils. There was a maximum improvement in the static and dynamic performance of elliptical bearing operated with nanolubricant. Khan et al. [8] studied the journal bearing performance by using nanolubricants and adopting power law model at different power index values and also considering the influence of liner deformation. Load capacity and friction force were found to be increased with the increase in nanoparticle concentration for rigid as well as flexible bearings. Nallusamy [29] performed characterization of punga oil and plastic oil by adding 0.1wt% of graphite in these oils for exploring their use in heavy load ball bearing. Punga based nanolubricant had lower value of friction coefficient and 3.3% higher value of load capacity as compared to base oil (punga) and 38% more load capacity was found in comparison to VG32 mineral oil. Dhanola and Garg [7] analysed the journal bearing performance by the addition of TiO_2 nanoparticles in bio-based lubricants and considering various power law index values. Bio-based nanolubricants resulted in higher values of pressure, load capacity and maximum temperature and moreover the increase was more pronounced at higher power law index and increased concentration of nanoparticles. Baskar et al. [30] performed a comparative experimental investigation with synthetic oil and nanoparticles based bio-lubricants for examining the tribological characteristics of journal bearing. Three nanoparticles i.e. TiO_2, CuO and WS_2 in 0.5wt% concentrations were separately mixed in Chemically Modified Rapeseed Oil (CMRO) to study their effect on bearing journal bearings by applying load of 10 kN and at journal speed of 3000 rpm. CuO based CMRO nanolubricant was instrumental in reducing wear and friction in bearings. Later, Baskar et al. [31] developed a fuzzy logic-based model for comparing the oil film pressure with same set of synthetic and nanoparticles based biolubricants. CuO based biolubricants were found to be a suitable replacement for synthetic lubricant at lower values of loads and speeds.

In the sum of above analysis, it is quite clear that very limited literature is available for investigating the static thermal performance of circular journal bearings operated with nanoparticles-based lubricants. Hence, this article develops a model for examining the static thermal performance of circular journal bearing by blending different concentrations of CuO and TiO_2 nanoparticles in mineral-based oils. The effect of change in journal speed and eccentricity ratio has been studied in a theoretical model for evaluating load capacity, oil temperature and hydrodynamic pressure distribution along with power losses for both the nanoparticles-based lubricants. The remainder of present paper has been organized as follows. Equations related to performance characteristics of circular journal bearing have been given in section 2, while section 3 represents the different relationships used for determining thermo physical properties of nanolubricants. Section 4 explains the methodology used in modeling,

while results obtained by solution of bearing characteristics equations have been given in section 5. Concluding remarks of the present study have been given in section 6.

2. BEARING PERFORMANCE CHARACTERISTIC EQUATIONS

There is formation of thin oil film in clearance space between bearing and journal and after rotation of journal for some time; the developed hydrodynamic film gets stabilized. As a result, the pressure generated inside the bearing under state steady conditions for incompressible fluid is governed by Reynolds equation as expressed in Eq. (1) [5], [23], [32]:

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

where, μ represents the dynamic viscosity of lubricant (Pa.s), p is the oil film pressure (Pa), U is journal translational velocity (m/sec).

The solution of Reynolds equation is carried out by assuming following conditions:

- Negative pressure (cavitation pressure) has been assumed as zero pressure.
- The pressure at the edges of the bearing i.e. at $x=0$ and $x=l$ has been assumed to be atmospheric pressure (l is the bearing length).
- Journal bearing is supplied with constant pressure lubricating oil.
- Pressure distribution has been assumed to be symmetric about the central plane of bearing.

Oil film thickness for circular bearing h_c can be expressed as Eq. (2) [5]

$$h_c = C\left(1+\varepsilon\cos\theta\right) \tag{2}$$

Where; e is eccentricity, C is clearance of circular bearing, θ is circumferential angle and ε is eccentricity ratio which is given by $\varepsilon = e/C$.

The bearing and lubricating oil parameters for the present study are given in Table 1.

Table 1. Bearing and lubricating oil parameters

Parameter	Value	
Bearing		
Bearing length (m)	0.1	
Journal diameter (m)	0.1	
Clearance (m)	200×10^{-6}	
l/d ratio	1	
Journal speed (rpm)	2000, 3000, 4000, 5000	
Lubricating Oils [33]		
Parameter	LO1	LO2
Lubricant type	AW68	AW100
Oil viscosity @40°C (cSt)	68.3	98.3
Flash point (°C)	218	230
Pour point (°C)	-9	-9
Oil density (kg/m^3)	881	886
Viscosity Index of oil	99	97

Energy equation under steady state conditions for incompressible flow can be expressed as Eq. (3) [32]:

$$\rho S\left(u\frac{\partial T}{\partial x}+w\frac{\partial T}{\partial z}\right)=\frac{\partial}{\partial y}\left(K\frac{\partial T}{\partial y}\right)+\mu\left[\left(\frac{\partial u}{\partial y}\right)^2+\left(\frac{\partial w}{\partial y}\right)^2\right] \tag{3}$$

Where ρ denotes the density (kg/m^3), S is the specific heat (J/kg-K), while u and w are velocity components along x and z directions respectively. K represents the thermal conductivity (W/m-K). Left side term of Eq. (3) is the energy transfer due to convection, while the first and second terms on the right side are energy transfers due to conduction and dissipation respectively.

Variation of temperature across the film thickness in energy equation has been approximated by parabolic temperature profile approximation. The double integration of momentum equation (Eq. 4) has been carried out to calculate the velocity components u and w by taking the boundary conditions (Eqs. 5 and 6) into consideration [17], [18] and temperature profile across film thickness is represented by second order polynomial.

$$\frac{\partial p}{\partial x}=\frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right);\ \frac{\partial p}{\partial z}=\frac{\partial}{\partial y}\left(\mu\frac{\partial w}{\partial y}\right) \tag{4}$$

$$u = u_{LB} \text{ ; at } y = 0 \text{ and } w = 0 \quad (5)$$

$$u = u_{UB} \text{ ; at } y = h_c \text{ and } w = 0 \quad (6)$$

u_{LB} and u_{UB} represent the velocities of lower and upper bonding surfaces respectively.

Oil film temperature has been computed by applying following assumptions and boundary conditions on energy equations [34]:

- Boundary conditions at interface of lubricating oil- bearing, lubricating oil-journal and inlet of the bearing are:

$$\begin{aligned} &\text{When } y = h_c,\ T = T_{UB} \text{ for } 0 \le x \le l; \\ &\text{When } y = 0,\ T = T_{LB} \text{ for } 0 \le x \le l; \\ &\text{When } x = 0,\ T = T_0 \text{ for } 0 \le y \le h_c; \\ &T(0,y) = T_0;\ T(x,0) = T_0; \\ &K_{lo}\left\{\frac{\partial T}{\partial y}\right\}_{UB} = K_b\left\{\frac{\partial T_b}{\partial y_b}\right\}_{y_b=0} \end{aligned} \quad (7)$$

Here T_0 represents oil inlet temperature, K_{lo} and K_b are thermal conductivities of lubricating oil and bearing material respectively, subscripts *LB* and *UB* express lower and upper bonding surface of bearing.

- Oil density has been assumed constant throughout the oil film
- Heat conduction and dissipation have been assumed negligible in x and z directions

3. NANOLUBRICANTS

For the purpose of modeling, nanolubricants are prepared by mixing 0.5wt%, 1.0wt% and 2.0wt% concentrations of nanoparticles, namely CuO and TiO_2 in LO1 and LO2 based mineral oils. Such nanoparticles impart special characteristics to lubricants, which cannot be achieved by the addition of traditional additives. The characteristics of both the nanoparticles have been summarized in Table 2.

Nanoparticle weight concentration has been converted to volume fraction of nanoparticles using the relationship given by Azmi et al. [36] as given in Eq. (8).

$$\varphi_{np} = \frac{\left(\dfrac{W_{np}}{\rho_{np}}\right)}{\left(\dfrac{W_{np}}{\rho_{np}} + \dfrac{W_{bl}}{\rho_{bl}}\right)} \quad (8)$$

where, φ_{np} = volume fraction of nanoparticles in base lubricant; W_{np} = weight of nanoparticles mixed in base lubricant; ρ_{np} = nanoparticle density, W_{bl} = base lubricant weight; ρ_{bl} = base lubricant density

Table 2. Nanoparticles properties [35]

Characteristic	Titanium oxide (TiO_2)	Copper oxide (CuO)
Density (kg/m³)	4230	6310
Thermal conductivity (W/m-K)	8.3	20
Specific heat (J/kg-K)	690	565
Boiling point (°C)	2972	2000
Average particle size (nm)	40	40
Molecular weight (g/mol)	79.866	79.54
Physical appearance	Spherical shape/ white colour	Spherical shape/ black colour

Density and specific heat of nanolubricants have been calculated with relationship given by Buongiorno [37]

$$\rho_{nl} = (1 - \varphi_{np})\rho_{bl} + \varphi_{np}\rho_{np} \quad (9)$$

where, ρ_{bl} = base lubricant density; ρ_{nl} = nanolubricant density

$$S_{nl} = \frac{(1 - \varphi_{np})\rho_{bl}S_{bl} + \varphi_{np}\rho_{np}S_{np}}{\rho_{nl}} \quad (10)$$

where, S_{np}, S_{bl} and S_{nl} are the specific heat of nanoparticle, base lubricant and nanolubricant respectively.

The density and specific heat obtained using Eq. (9) and (10) have been used in energy equation (Eq. 3).

Modified Krieger Dougherty method has been deployed for computation of nanolubricants viscosity at different concentrations of nanoparticles. The relationship used for determining the viscosity of oil that containing dispersed particles is as follows [38]:

$$\mu_r = \frac{\mu_{nl}}{\mu_{bl}} = \left(1 - \frac{\varphi_{np}}{\varphi_{mf}}\right)^{-\eta\varphi_{mf}} \tag{11}$$

μ_{bl} is base lubricant viscosity; μ_{bl} is nanolubricant viscosity at particular volume fraction of nanoparticles; φ_{mf} depicts the maximum particle packing fraction; η is the intrinsic viscosity. φ_{mf} and η have been taken as 0.605 and 2.5 (for spherical shaped suspensions) respectively in this study [39]. Equation (11) was modified by considering the mechanisms of primary and aggregate particles [40] and can be rewritten as :

$$\mu_r = \frac{\mu_{nl}}{\mu_{bl}} = \left(1 - \frac{\varphi_{ap}}{\varphi_{mf}}\right)^{-2.5\varphi_{mf}} \tag{12}$$

and $$\varphi_{ap} = \varphi_{np}\left(\frac{a_{rg}}{a_{rp}}\right)^{3-F} \tag{13}$$

where, a_{rp} and a_{rg} denotes the radius of primary and aggregate particles respectively. The value of a_{rg}/a_{rp} has been assumed as 7.77 and F is fractal index and its value is reported as 1.8 for nanofluids [41].

The thermal conductivities of different nanolubricants have been measured using the Maxwell model [42] and are expressed as Eq. 14:

$$\frac{K_{nl}}{K_{bl}} = 1 + \frac{3(\beta - 1)\varphi_{np}}{(\beta + 2) - (\beta - 1)\varphi_{np}} \tag{14}$$

K_{bl} and K_{nl} are thermal conductivities of base lubricant and nanolubricant respectively; $\beta = K_{nl}/K_{bl}$.

4. METHODOLOGY

The static thermal performance of circular bearing has been determined by generating the model in MATLAB®R2020a software for solving the Reynolds equation by applying Finite Difference Method (FDM). The viscosity of nanolubricants has been calculated by modified Krieger Dougherty method, whereas the thermal conductivity was predicted using Maxwell method. Both density and specific heat of nanolubricants have been calculated using the relationships given by Buongiorno. The process of meshing for bearing was done by performing 100 divisions along circumference of bearing and 50 divisions along length of bearing. Values of bearing dimensions, journal speed and eccentricity ratio were put in code at initial assumed values of pressure, temperature and viscosity. The clearance zone between the bearing and shaft has been divided into finite nodal elements. The gap between the bearing and journal filled by lubricants is calculated at different nodal points using film thickness equation. The isothermal pressure was then computed by keeping the viscosity constant and assuming over relaxation for error convergence. The criterion used for estimating the error convergence for pressure is given in Eq. (15). Negative pressure obtained during the simulation was considered to belong to the cavitation zone and was set equal to zero. The energy equation was solved with FDM by applying Parabolic Temperature Profile Approximation (PTPA) technique. Temperature values were calculated using under relaxation for error convergence as given in Eq. (16).

$$\frac{\left|\left(\sum P_{j,k}\right)_{i-1} - \left(\sum P_{j,k}\right)_i\right|}{\left|\left(\sum P_{j,k}\right)_i\right|} \le 0.0001 \tag{15}$$

$$\frac{\left|\left(\sum T_{j,k}\right)_{i-1} - \left(\sum T_{j,k}\right)_i\right|}{\left|\left(\sum T_{j,k}\right)_i\right|} \le 0.0001 \tag{16}$$

Where 'i' is the number of iterations

The real challenge lies in adopted methodology is that the pressure and temperature values obtained in Eq. 15 and 16 need to be iterated till the convergence criteria is met. Then, the viscosity obtained at new temperature was calculated on the basis of viscosity temperature relationship [43] given in Eq. (17).

$$\mu = \mu_i e^{-\gamma(T_p - T_i)} \tag{17}$$

where μ_i, denotes lubricant's dynamic viscosity (Pa.s) at oil inlet temperature. T_i and T_p depict the temperatures at the preceding nodes for including the effect of viscosity variation in Reynolds equation and γ is temperature viscosity coefficient of oil. The methodology adopted for studying the influence of nanolubricants on the performance of journal bearing in this study is given in Fig. 1

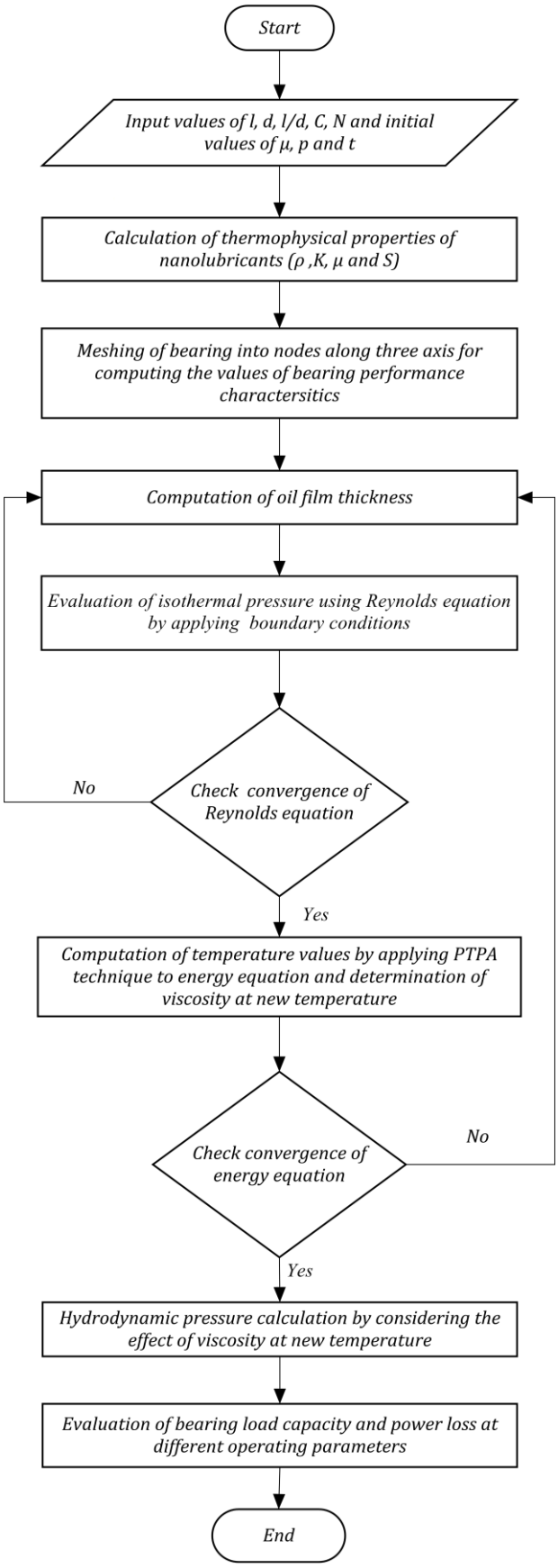


Fig.1. Flowchart of methodology proposed for evaluating the static thermal performance of circular bearing

The new viscosity values were used to calculate the hydrodynamic pressure obtained at all the nodal points. The developed model was validated by comparing the values of pressure obtained with the results of Sehgal et al. [44]. Further, the load capacity of bearing is calculated by integration of pressure distribution along circumference and width of journal bearing and Simpson's rule is applied for this. The power loss value has been computed from frictional force and which can be expressed as in Eq. (18) [5].

$$F = \int_0^{2\pi} \int_0^{l} \tau dz d\theta \tag{18}$$

Where τ and F are the shear stress and frictional force respectively.

5. RESULTS AND DISCUSSION

This section presents the results obtained by modeling of circular journal bearing to study the effect of nanoparticles-based lubricants on the static thermal performance of bearing. Pressure and temperature distribution were evaluated at different journal speeds and eccentricity ratios for both the base lubricants and nanolubricants. Figure 2 depicts the oil film pressure distribution obtained at 0.6 eccentricity ratio and at journal speed of 2000 rpm for LO2. It shows the presence of one peak due to the presence of a single oil film, while negative pressures have been shown as zero pressure.

Figure 3 depicts the variation in maximum pressure attained at different journal speeds and for nanolubricants for an eccentricity ratio of 0.6. Value of maximum pressure has been found to be increased with an increase in journal speed while LO2 based nanolubricants have a higher value of pressure than LO1 based nanolubricants due to the high viscosity of LO2. The maximum values of hydrodynamic pressure obtained at different operating parameters and nanolubricants have been summarized in Table 3. It has been observed that the pressure rises with an increase in eccentricity ratio and the addition of nanoparticles in lubricants further increased the values of hydrodynamic pressure. It may be due to the fact that the viscosity of nanolubricant improves with increase in nanoparticles concentration, which causes an

increase in hydrodynamic pressure. An increase of 14.23% and 9.23% in maximum oil pressure has been found with 2wt% addition of TiO_2 and CuO respectively in LO1 for eccentricity ratio of 0.6 at journal speed of 2000 rpm. A similar trend was also observed with the nanoparticles mixed in LO2. It can be noticed that TiO_2 based nanolubricants have a pronounced effect on maximum pressure in comparison to copper oxide, which was also reported in [25].

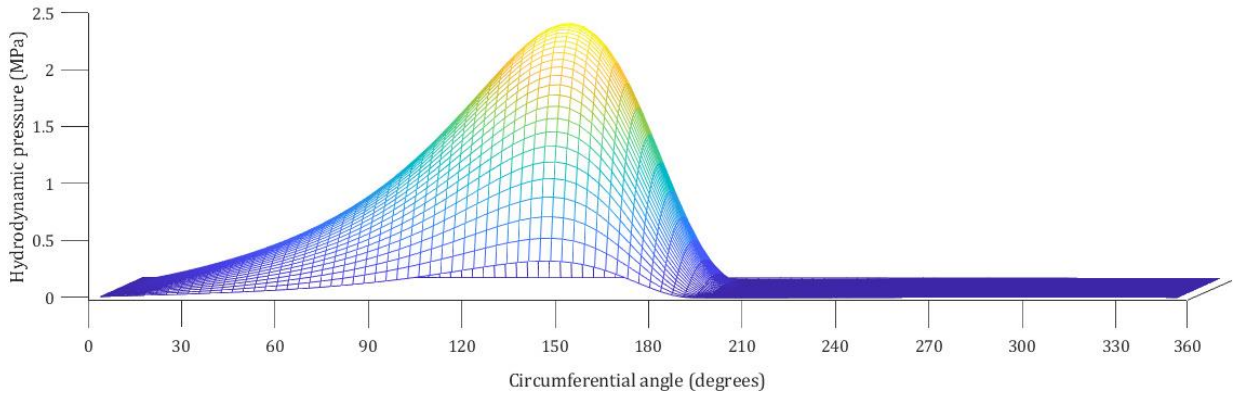
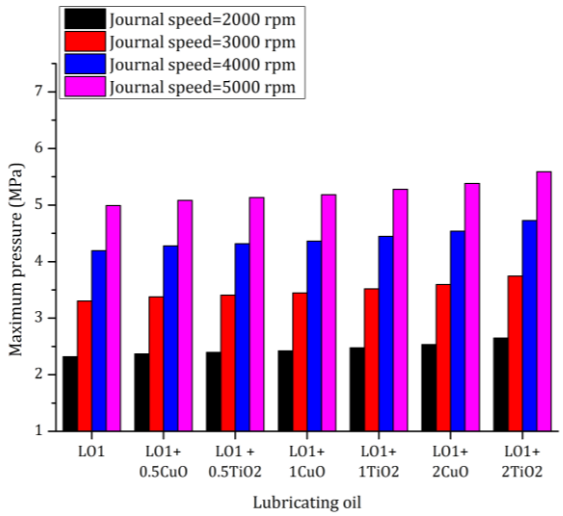
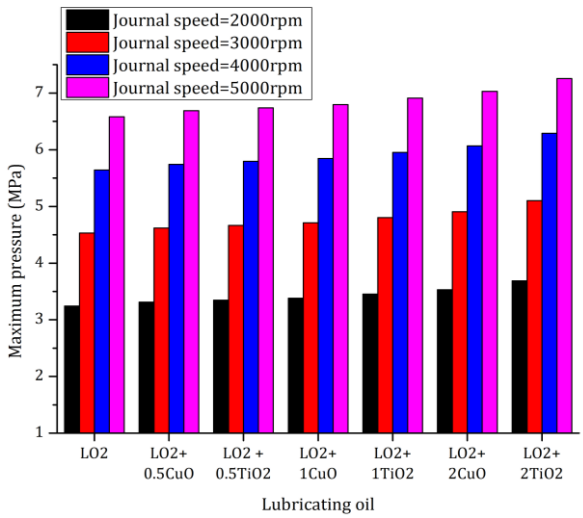


Fig. 2. Pressure distribution for LO2 at ε= 0.6, N=2000 rpm

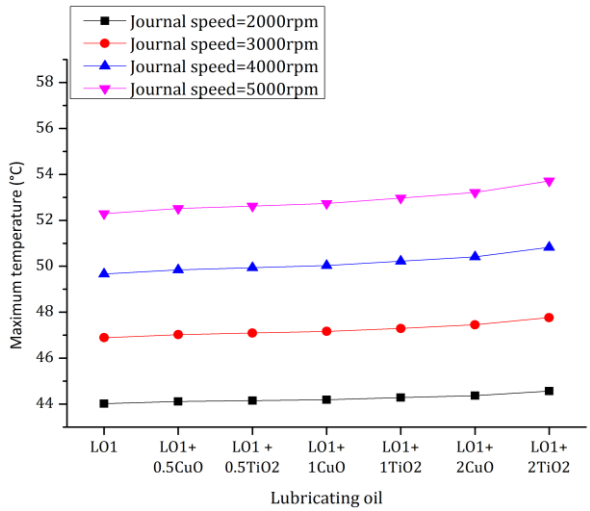


(a) LO1 with different concentrations of CuO and TiO_2

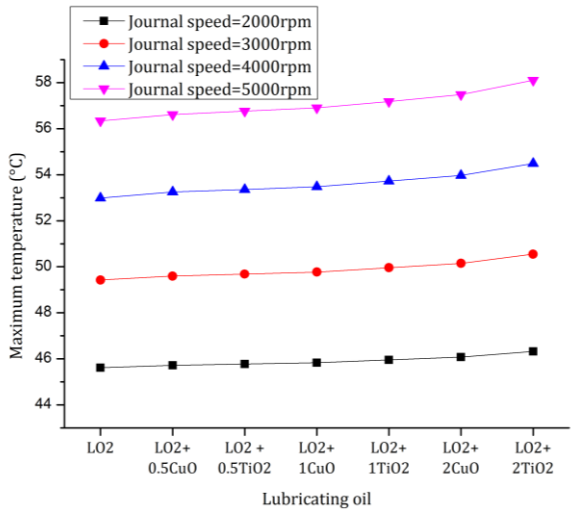


(b) LO2 with different concentrations of CuO and TiO_2

Fig. 3. Maximum pressure values obtained at different journal speeds and at eccentricity ratio of 0.6



(a) LO1 with different concentrations of CuO and TiO_2



(b) LO2 with different concentrations of CuO and TiO_2

Fig. 4. Maximum temperature values obtained at different journal speeds and at eccentricity ratio of 0.6

Table 3. Hydrodynamic pressure values at different operating parameters with nanolubricants

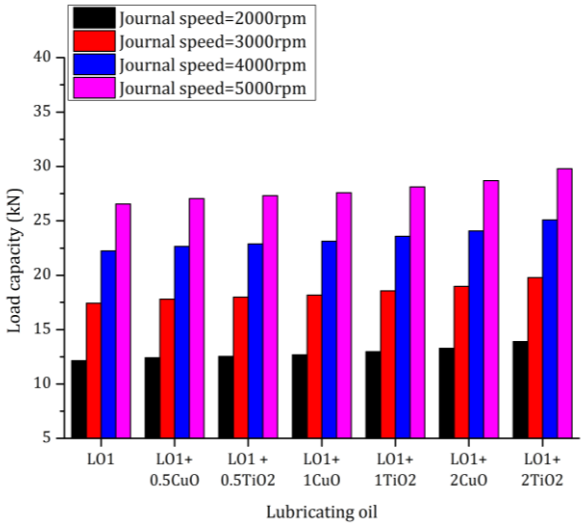
Maximum hydrodynamic pressure (in MPa)										
Lubricating oil	Nanoparticle concentration (wt %)	Eccentricity ratio	**CuO nanoparticles**				**TiO_2 nanoparticles**			
			Journal Speed (rpm)				Journal Speed (rpm)			
			2000	3000	4000	5000	2000	3000	4000	5000
LO1	0.5	0.4	0.973	1.410	1.820	2.203	0.983	1.426	1.838	2.225
		0.5	1.501	2.161	2.770	3.331	1.517	2.185	2.799	3.365
		0.6	2.369	3.375	4.277	5.085	2.395	3.409	4.318	5.133
		0.7	3.994	5.564	6.881	7.977	4.035	5.616	6.943	8.040
LO1	1	0.4	0.995	1.442	1.859	2.248	1.017	1.473	1.897	2.294
		0.5	1.534	2.209	2.828	3.398	1.568	2.255	2.885	3.464
		0.6	2.422	3.445	4.362	5.183	2.476	3.516	4.447	5.279
		0.7	4.079	5.675	7.008	8.110	4.167	5.789	7.133	8.238
LO1	2	0.4	1.041	1.507	1.939	2.343	1.090	1.575	2.024	2.442
		0.5	1.605	2.307	2.947	3.537	1.680	2.408	3.071	3.680
		0.6	2.533	3.594	4.539	5.384	2.649	3.748	4.727	5.588
		0.7	4.261	5.908	7.270	8.384	4.452	6.148	7.535	8.663
LO2	0.5	0.4	1.372	1.964	2.503	2.996	1.387	1.985	2.529	3.026
		0.5	2.110	2.991	3.775	4.480	2.133	3.020	3.811	4.518
		0.6	3.310	4.621	5.743	6.689	3.344	4.666	5.794	6.737
		0.7	5.524	7.466	8.957	10.312	5.578	7.532	9.027	10.406
LO2	1	0.4	1.403	2.006	2.554	3.056	1.433	2.049	2.606	3.112
		0.5	2.156	3.052	3.850	4.560	2.202	3.113	3.926	4.642
		0.6	3.381	4.713	5.846	6.797	3.453	4.804	5.952	6.909
		0.7	5.638	7.603	9.101	10.469	5.751	7.739	9.241	10.615
LO2	2	0.4	1.466	2.094	2.662	3.175	1.534	2.184	2.770	3.298
		0.5	2.253	3.180	4.006	4.728	2.355	3.314	4.163	4.906
		0.6	3.530	4.905	6.066	7.028	3.686	5.104	6.290	7.255
		0.7	5.876	7.886	9.400	10.776	6.122	8.176	9.697	11.064

Maximum temperature values have been computed by using PTPA technique for different journal speeds and lubricants at an eccentricity ratio of 0.6 as shown in Fig. 4. The oil inlet temperature was assumed to be 40°C [24] . It has been found that there is a rise in maximum temperature with an increase in journal speed which may be attributed to the increase of shear forces at higher speeds. Maximum value of the oil film temperature is obtained with the oil having highest viscosity as also reported in [4].

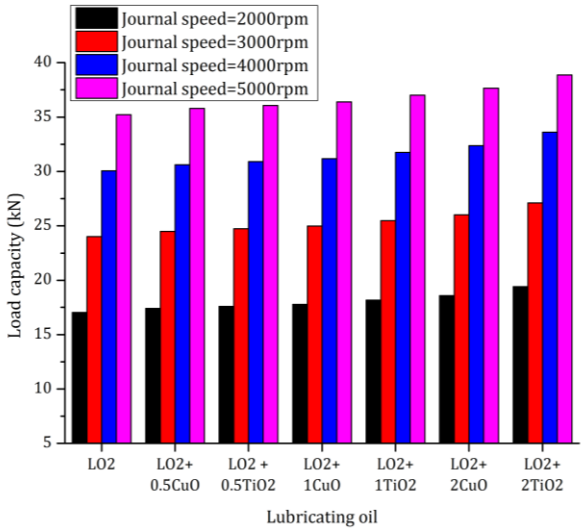
Although there is a slight increase in oil temperature on the addition of nanoparticles in mineral oils, a higher increase has been observed with an increase in journal speed. The oil temperature for LO1 with 1wt% CuO was found to 44.19°C and 52.73°C at 2000 and 5000 rpm respectively, while for the same parameters temperature was found to be 45.83°C and 56.9°C for LO2 oil with CuO. There was an increase in oil film temperature from 44.02°C to 44.56°C on the addition of 2wt% TiO_2 in LO1 at a journal speed of 2000 rpm at eccentricity ratio of 0.6 and it increased from 52.29°C to 53.71°C at journal speed of 5000 rpm. There were slightly higher values of temperature for LO2 as well as for LO2 based nanolubricants.

Figure 5(a) and (b) shows the load capacity values, which have been obtained from the pressure distribution across oil film for both the oils with varying concentrations of CuO and TiO_2 nanoparticles at different operating

conditions. The trend shows that load capacity is improved with an increase in journal speed by keeping all other operating parameters constant.



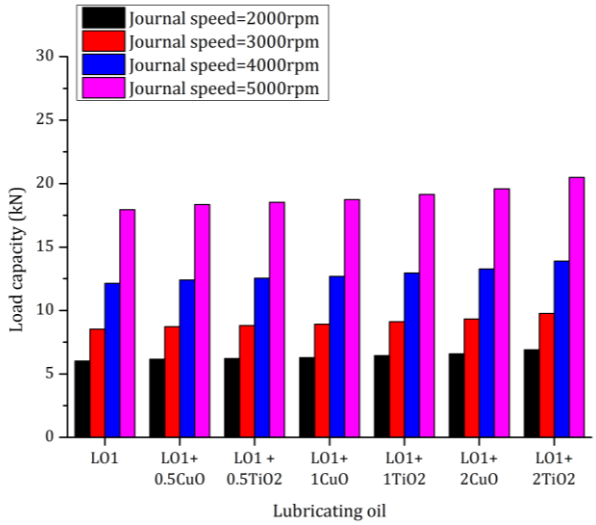
(a) LO1 with different concentrations of CuO and TiO_2



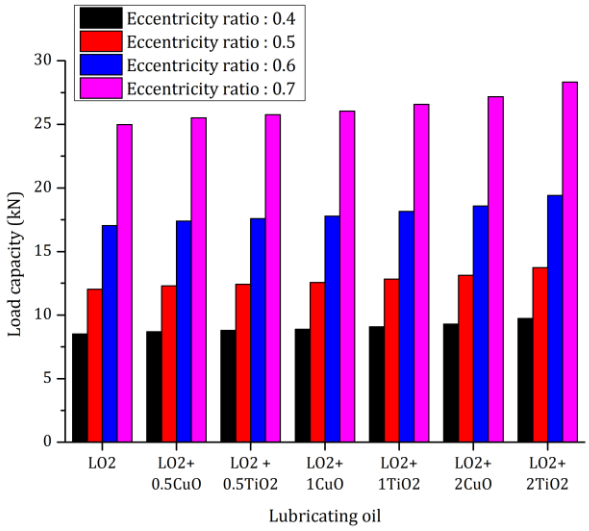
(b) LO2with different concentrations of CuO and TiO_2

Fig. 5. Load capacity at different journal speeds at eccentricity ratio of 0.6

A significant positive effect on load capacity with an increase in the concentration of nanoparticles has also been observed which was in accordance with the results presented by Nair et al. [22]. This happens because of the fact that the addition of nanoparticles causes an increase in maximum pressure values, thereby increasing the bearing load capacity. Moreover, TiO_2 nanoparticles-based lubricants achieved higher load capacity as compared to copper oxide-based nanolubricants. Load capacity has been observed to be enhanced with the increase in journal speed and oil viscosity at same eccentricity ratio. The effect of change in eccentricity ratio on load capacity has also been studied at different lubricating conditions and constant journal speed as shown in Figure 6 (a) and (b). It has been clearly noticed that the load capacity improves at higher eccentricity ratio which can be attributed to a decrease in oil film thickness. Also, the addition of nanoparticles causes a further increase in load capacity at all the eccentricity ratios.



(a) LO1 with different concentrations of CuO and TiO_2

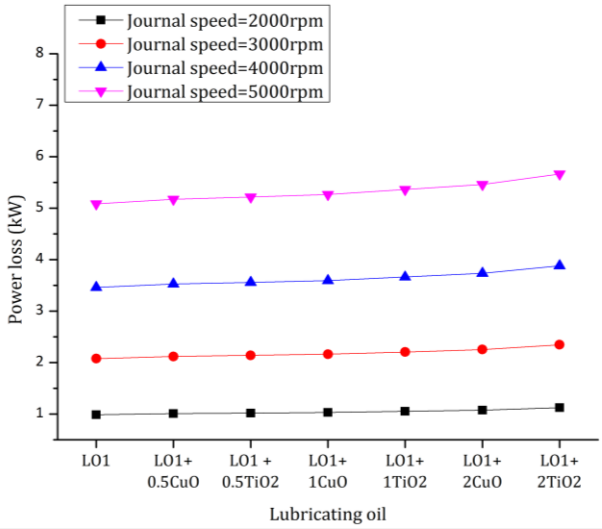


(b) LO2 with different concentrations of CuO and TiO_2

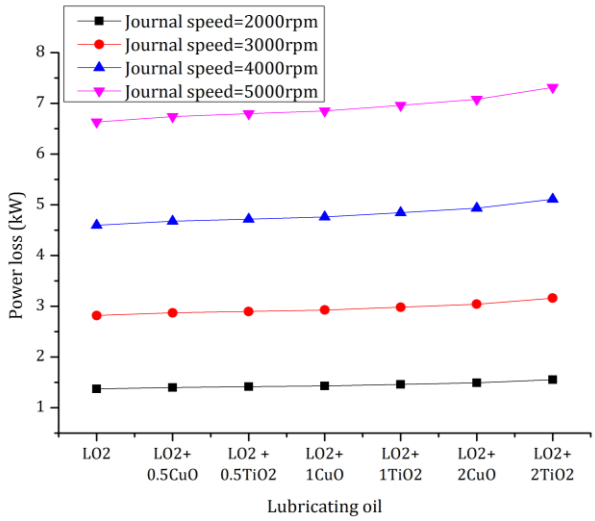
Fig. 6. Load capacity at different eccentricity ratios for journal speed of 2000 rpm

The percentage increase in load capacity and maximum value of oil temperature for LO1

with different concentrations of nanoparticles at 2000 rpm has been summarized in Table 4. An increase of 14.49% and 9.39% in load capacity has been observed at ε=0.5 with 2wt% addition of CuO and TiO2 respectively with a corresponding increase of just 1.04% and 0.67% in maximum oil temperature. The trend reveals that with an increase in nanoparticle concentration, load capacity improves significantly with a slight increase in oil temperature which outlines the importance of nanoparticles-based lubricants for hydrodynamic journal bearings. This fact is in accordance with the findings of Nicoletti [24] which demonstrated that an increase in dynamic viscosity contributes to enhancement in hydro-dynamic journal bearing load capacity, and further increases in load capacity could be due to a slight rise in oil temperature due to improvement of volumetric heat capacity of nanolubricants. The trend of power loss with the variation in journal speed and nanoparticles concentrations at an eccentricity ratio of 0.6 has been shown in Figure 7. It has been found that the power losses increase with an increase in oil viscosity and also with increased journal speed. An increase in the nanoparticles also leads to the slight rise in power loss which can be attributed to the fact that with the increase in nanoparticle concentration, there is greater resistance due to viscous force, which further contributes to an increase in frictional force and hence power loss as observed in [23].



(a) LO1 with different concentrations of CuO and TiO_2



(b) LO2 with different concentrations of CuO and TiO_2

Fig. 7. Power loss at different journal speeds and at eccentricity ratio of 0.6

Table 4. Percentage variation in load capacity and maximum temperature for LO1 with different concentrations of nanoparticles at 2000 rpm

Nanolubricant description	Percentage Increase in load capacity at eccentricity ratios		Percentage Increase in maximum oil temperature at eccentricity ratios	
	ε=0.5	ε=0.6	ε=0.5	ε=0.6
LO1 with 0.5wt% CuO	2.25	2.17	0.16	0.20
LO1 with 0.5wt% TiO_2	3.38	3.28	0.23	0.29
LO1 with 1.0wt% CuO	4.55	4.44	0.32	0.39
LO1 with 1.0wt% TiO_2	6.88	6.75	0.46	0.59
LO1 with 2.0wt% CuO	9.39	9.23	0.67	0.89
LO1 with 2.0wt% TiO_2	14.49	14.23	1.04	1.23

6. CONCLUSIONS

Static thermal performance of circular journal bearing has been evaluated by generating the model in MATLAB®R2020a software. Comparative performance analysis has been carried out by mixing different concentrations of CuO and TiO_2 nanoparticles in two different viscosity grade oils at different operating conditions. The major findings obtained from this work are as follows:

a) The maximum pressure values and hence load capacity of circular bearing was found to be increased with the addition of nanoparticles in base lubricating oils.

b) An increase in nanoparticle concentration contributes to a significant improvement in load capacity with a slight increase in oil temperature, which outlines the importance of nanoparticles-based lubricants for hydrodynamic journal bearings.

c) There was an increase in power loss with the increase in nanoparticles concentration due to an increase in frictional fo0rces which is manifested by an increase in lubricant viscosity. Higher journal speed further increases the frictional forces resulting in an increase of power loss.

d) The static thermal performance obtained using TiO_2 nanoparticles-based lubricants provide superior results in comparison to CuO based nanolubricants

e) An increase in eccentricity ratio leads to an increase in maximum pressure values and load capacity of journal bearings for both base lubricants as well as nanolubricants

Future work may focus on experimental investigation of nanolubricants on the static and dynamic characteristics of journal bearings, which can be implemented in turbomachinery for performance optimization. Hybrid nanoparticles can be explored for their synergistic effect in hydrodynamic lubrication regime. The deployment of graphene-based lubricants is also an interesting area for evaluating the hydrodynamic performance of journal bearings. The application of nanolubricants can be considered for performance evaluation of worn-out journal bearings and bearings having higher surface roughness.

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Nomenclature

C	Clearance of circular bearing (m)
D	Bearing diameter (m)
e	Eccentricity (m)
h_c	Film thickness of circular bearing (m)
i	Number of iterations
K	Thermal conductivity (W/m-K)
l	Bearing length (m)
LO1	Lubricating oil -1 (AW68)
LO2	Lubricating oil -2 (AW100)
LOm + nCuO	LOm + nwt% CuO; (m=1,2 and n=0.5, 1 and 2)
LOm + nTiO_2	LOm + nwt% TiO_2; (m=1,2 and n=0.5, 1 and 2)
N	Journal rotational speed (rpm)
p	Film pressure (Pa)
S	Specific heat (J/kg-K)
T_0	Oil inlet temperature (°C)
U	Journal velocity (m/sec)
u_{LB}	Velocity of lower bonding surface (m/sec)
u_{UB}	Velocity of upper bonding surface (m/sec)
W_{np}	Weight of nanoparticles (kg)
W_{bl}	Weight of base lubricant (kg)
ϕ	Attitude angle
θ	Circumferential angle
ρ	Density (kg/m^3)
μ	Dynamic viscosity of lubricant (Pa.s)
ε	Eccentricity ratio
τ	Shear stress, (N/m^2)
φ_{np}	Volume fraction of nanoparticles
μ_r	μ_{nl} / μ_{bl}
β	K_{nl} / K_{bl}
Al_2O_3	Aluminum oxide
CuO	Copper oxide
CeO_2	Cerium oxide
CMRO	Chemically Modified Rapeseed oil
FDM	Finite Difference Method
FEM	Finite Element Method
PTPA	Parabolic Temperature Profile Approximation
TiO_2	Titanium dioxide
THD	Thermohydrodynamic

Subscripts:

b	Bearing material
bl	Base lubricant
lo	Lubricating oil
LB	Lower bonding surface of bearing
np	Nanoparticles
nl	Nano lubricant
UB	Upper bonding surface of bearing