Thermohydrodynamic Characteristics of Worn Journal Bearing Lubricated With Oil Containing Nanoparticles Additive

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Abstract:

In the present work the worn journal bearing is simulated to discuss the effect of adding TiO_2 nanoparticles to the base oil on its thermal performance. An extensive numerical investigation is carried out to study the effect of different parameters affecting thermal performance of worn journal bearing such as the eccentricity ratio (ε), the wear depth parameter (δ), and the nanoparticle concentration (Φ). The computational approach is provided by using finite difference method for solving the governing equations, namely, the modified Reynolds equation, energy and heat conduction equations with suitable equation to include the variation of the oil film thickness due to the bearing wear in order to estimate the benefits of using nanolubricant in worn journal bearings. Oil viscosity dependence on nanoparticle concentrations is considered by using Krieger Dougherty model. The mathematical model as well as the computer program prepared to solve the governing equations were validated by comparing the oil film pressure distribution obtained in the present work for a worn journal bearing with that obtained numerically by Hashimoto et.al [2](1986) with 3% maximum deviation between the results. The maximum oil film pressure obtained in this work was compared with that obtained experimentally by Roy [12] (2009) for intact journal bearing with 3% as a maximum error between the results. The results obtained show that the nanoparticles addition by 0.5% and 1% to the base oil increases the load carrying capacity of the worn journal bearing by 20% and 40% respectively while decreases the oil side leakage by 5% and 10% and friction coefficient by 2.75% and 5.7% as compared to that lubricated with pure oil. This is happen with the expense of power losses. Calculations also shows that adding a higher percentage of nanoparticles (2%) has a harmful effect on the performance of a worn journal bearing since the power losses is highly increased.

Keywords: Thermo-hydrodynamic lubrication, Worn journal bearing, Nanoparticle

1 Introduction:

Bearing is a machine element which supports another moving machine element (known as journal).It permits a relative motion between the contact surfaces of the members, while carrying the load .In order to reduce frictional resistance , wear

and in some cases carry away the heat generated, a layer of lubricant provided to separate the journal and the bearing. A thin film lubricated journal bearing was used to develop positive Pressure due to the relative motion of the two surfaces **Dufran et.al [1] (1983)** conducted experimental work on the wear of steam turbine generator journal bearing**.** It has been pointed out through this work that the wear pattern is uniform in width along the bearing length and is located almost exactly symmetrically at the bottom of the bearing. No work had been recorded in this field before this work. **Hashimoto et.al, [2], (1986)** investigated the effects of geometric change due to wear on the hydrodynamic lubrication of journal bearings in both laminar and turbulent regimes theoretically and experimentally. It was found that the geometric change due to wear has significant effects on the steady- state characteristics in both laminar and turbulent regimes. The static characteristics of worn non circular journal bearings in laminar and turbulent regimes were discussed by **Vaidyanathan and Keith, [3], (1991)** and **Kumar and Mishra, [4], (1996)** .

Wear measurement for radial journal bearings was carried out by **Ligterink and Gee, [5], (1996)** .

Rozeanu and Kennedy, [6], (2001) shows that the wear in journal bearings occurs in three different locations. It has been shown that the study of the wear problem requires different experimental model and remedy for each of the three different wear modes. Many research works were carried out to investigate the thermal performance of worn and intact bearings. **Nikolakopoulos and Papadopoulos [7], (2008)** presented an analytical model in order to find the relationship among the friction force, the misalignment angles and wear depth in journal bearing. **Fillon and Bouyer, [8], (2004)** presented a thermo-dynamic analysis of a worn plain journal bearing considering cavitation effect. It has been shown that the defects caused by wear of up to 20% have little influence on bearing performance whereas above this value (30 to 50%) it displays an interesting advantage due to the significant fall in oil film temperature. **Awasthi et.al, [9], (2007)** presented an analytical study in attempt to replicate the performance of a worn non-recessed (hole entry) capillary-compensated hybrid journal-bearing system. The computed results further indicate that the influence of wear defects on journal bearing performance may be minimized if the designer selects a suitable bearing configuration. In parallel many workers **[10-21]**deal with the effect of adding solid nanoparticles to the base oil on the performance of intact journal bearings. **Elsharkawy[10], (2005)** used a rheolo-gical model to discuss theoretically the influence of lubricant additives on the steady state performance of hydro dynamically lubricated finite journal bearings. The results showed that lubricant additives significantly increase the load carrying capacity and reduce both the coefficient of friction and the side leakage as compared to the Newtonian lubricants. **Wu et.al, [11], (2007)** examined the tribological properties of two lubricating oils, API-SF engine oil and a base oil, with CuO, TiO_2, and Diamond nanoparticles used as additives. Eexperimental results show that nanoparticles, especially CuO, added to the base oils exhibit good friction-reduction and anti-wear properties. The addition of CuO nanoparticles in the API-SF engine oil and the Base oil decreased the friction coefficient by 18.4 and 5.8%, respectively, and reduced the worn scar depth by 16.7 and 78.8%, respectively, as compared to the standard oils without CuO nanoparticles. . **Lee, et.al, [12], (2009)** used graphite nanoparticles with nano lubricants to enhance tribological properties and lubrication characteristics. The base lubricant used was industrial gear oil, which has a kinematic viscosity of 220 CST at 40ºC. To investigate the physical and tribological properties of nano lubricants, friction coefficients and temperatures were measured by a disk-on-disk tribotester. Results obtained indicated that graphite nanoparticle additives improve the lubrication properties of base lubricants.

Murshed et.al, [13], (2008) introduced a combined experimental and theoretical study on the effective thermal conductivity and viscosity of nanofluids. It has been found that both viscosity and thermal conductivity of the nanolfluid increase with particle volume fraction in comparison with that of the base fluid. The static characteristics of intact bearings lubricated with nanolubricants of different types of nanoparticles were studied by many research works. **Chandrasekar et.al, [14], (2010)**, presented the static and dynamic performance characteristics of four-lobe bearing operating with couple stress lubricant. It was found that the values of the couple stress parameter increases the load carrying capacity, decreases the friction coefficient and make this type of journal bearing more stable. The computed results show that the presence of couple stresses improves the performance characteristics of a four-lobe bearing compared to that lubricated with Newtonian fluids. **Nair et.al, [15], (2009)** show an increase in load carrying capacity of (40%) and a decrease in side leakage when the bearing lubricated with TiO_2 nanolubricant than that of pure oil.

Shenoy et.al, [16], (2012) studied the effect of CuO, TiO_2 and Diamond nanoparticles additives in API-SF engine oil, on static characteristics of an externally adjustable fluid-film bearing . It has been shown that a bearing having negative radial and negative tilt adjustments, and operating with API-SF engine oil blended particularly withTiO_2 nanoparticles, results in better load capacity with reduced end leakage and increased friction force, as compared to the base oil without nanoparticle additives. **Binu et.al , [17] and [18], (2014)** studied the static characteristics of a journal bearing operating with nanolubricant containing TiO_2 using variable viscosity approach. Results obtained show significant improvement in load carrying capacity of journal bearing operating with oil containing TiO_2 nanoparticles as compared to base oil. **Babu et al., [19], (2014)** developed a relationship between viscosity and temperature for the lubricant SAE 15W40 multi grade engine oil with Al_2O_3 and ZnO nanoparticles. It has been found that the addition of nanoparticles on commercially available lubricant considerably enhances the viscosity of lubricant and in turn changes the performance characteristics. **Gunnuang et.al, [20], (2015)** investigated the influence of Al_2O_3 nanoparticle additives on the performance characteristics of a journal bearing. Non-Newtonian fluid based on Carreau viscosity model was represented for SAE10W50 oil blended with Al_2O_3 nanoparticles. The results show that the addition of Al_2O_3 nanoparticles improve the load-carrying capacity of the journal but almost no change in film temperature due to good thermal property of Al_2O_3 nanoparticles. **Solghar [21], (2015)** studied the performance of the journal bearing lubricated with two types of lubricant: pure oil and base oil blended with Al_2O_3 nanoparticles.. It was found that the lubricant with nano additives increases the load-carrying capacity by (17.7%) at eccentricity ratio (0.9) and decreases the friction coefficient in comparison to that of the pure oil.

2 Bearing Geometry and Governing Equations

The schematic diagram of a worn journal bearing considered in this work is given in fig.1.

The shaft rotates at a constant angular velocity about its axis. The shaft radius (R_s) and the bearing radius (R_b) are practically identical.

To generalize the study, the following non-dimensional coordinates and parameters can be defined:

$$P=\bar{P}\mu_o\omega\left(\frac{R_sR_b}{c^2}\right),\quad \bar{h}=\frac{h}{c},\quad \bar{c}=\frac{c}{R_{bi}},\theta=\frac{x}{R_{bi}},$$

$$\bar{y}=\frac{y}{h}$$

$$\bar{Z}=\frac{z}{L},\bar{Z}=\frac{z}{L},\qquad \bar{\mu}_{nf}=\frac{\mu_{nf}}{\mu_o},$$

$$u,v,w=\omega R_s(\bar{u},\bar{v},\bar{w})$$

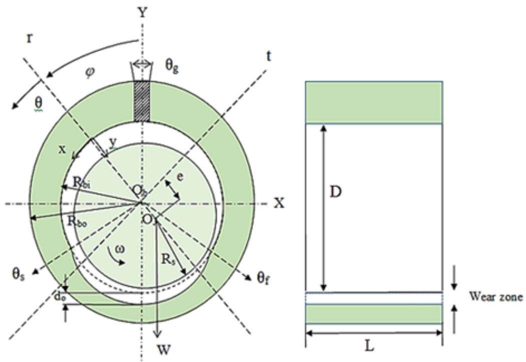


Figure (1). Geometrical Configuration of worn Journal Bearing.

3 Reynolds equation

The modified Reynolds equation for plain journal bearing considering thermal effect, steady state incompressible fluid flow is expressed as [22]

$$\frac{\partial}{\partial\theta}\left(\bar{G}\bar{h}^3\frac{\partial\bar{P}}{\partial\theta}\right)+\left(\frac{1}{4\alpha^2}\right)\frac{\partial}{\partial\bar{z}}\left(\bar{G}\bar{h}^3\frac{\partial\bar{P}}{\partial\bar{z}}\right)=\frac{\partial}{\partial x}\left(\bar{F}\bar{h}\right) \qquad (1)$$

Where

$$\alpha=\frac{L}{D} \qquad (2)$$

$$\bar{G}=\frac{\int_0^1\left(\bar{I}.\bar{J}-\bar{I}.\bar{I}_1\right)d\bar{y}}{\bar{I}_1} \qquad (3)$$

$$\bar{I}=\int_0^1\frac{\bar{y}}{\bar{\mu}_{nf}}d\bar{y},\quad \bar{J}=\int_0^{\bar{y}}\frac{\bar{y}}{\bar{\mu}_{nf}}d\bar{y},$$

$$\bar{I}_1=\int_0^1\frac{1}{\bar{\mu}_{nf}}d\bar{y} \qquad (4)$$

4 Oil film thickness:

The geometry of worn hydrodynamic journal bearing can be shown in figure (1). The journal and the bearing centers are assumed to be parallel and the oil film thickness becomes a function of (θ) only. The oil film thickness for the conventional journal bearing can be expressed in dimensional form as [1]:

$$h=c+e\cos(\theta) \qquad (5)$$

Equation (5) can be written in dimensionless form as:

$$\bar{h}=\frac{h}{c}=1+\varepsilon\cos(\theta) \qquad (6)$$

The oil film thickness is modified according to Dufran's model [1] to include the effect of geometry change due to wear:-

$$h=d_o+e\cos(\theta)-c\cos(\theta+\varphi) \qquad (7)$$

where:

d_o = Defect depth

Equation (7) can be rewritten in dimensionless form as:

$$\bar{h}=\delta_o+\varepsilon\cos(\theta)-\cos(\theta+\varphi) \qquad (8)$$

Where:

$$\delta_o=\text{ Dimensionless defect depth }=\frac{d_o}{c}$$

5 Viscosity –Temperature relationship

The Reynolds and energy equations are coupled by the oil viscosity, which was assumed to be function of temperature. The viscosity temperature relation-ship used by Ferron et.al [22] was adopted in the present work.

$$\bar{\mu}=\frac{\mu}{\mu_o}=k_o-k_1\bar{T}+k_2\bar{T}^2 \qquad (9)$$

Where:

k_o, k_1, k_2 are viscosity coefficients.

$\bar{T}$ = dimensionless oil film temperature.

μ_o = viscosity of the lubricant at the oil inlet temperature.

6 Thermo- physical properties of Nano lubricant:

a) Viscosity

The modified Krieger–Dougherty viscosity model [18] was adopted in the present work to evaluate the effect of nanoparticles on the oil viscosity as:

$$\mu_{nf}=\mu\left(1-\frac{\Phi_a}{\Phi_m}\right)^{-\eta\Phi_m} \qquad (10)$$

$$\Phi_a = \Phi\left(\frac{a_a}{a}\right)^{3-D} \quad (11)$$

where, Φ_a is the effective volume fraction, Φ is nanoparticle volume fraction, a_a and a are the radii of aggregates and primary particles respectively. D is the fractal index having a typical value of 1.8 for nanofluids. Φ_m is the maximum particle packing fraction ,which is approximately 0.605 at high shear rates and η is the intrinsic viscosity ,whose typical value for mono disperse suspensions of hard spheres is 2.5. The modified Krieger–Dougherty equation can be written as:

$$\mu_{nf} = \mu_o e^{-\beta_o(T-T_o)}\left(1-\frac{\Phi}{0.605}\left(\frac{a_a}{a}\right)^{1.2}\right)^{-1.5} \quad (12)$$

According to Binu et al. (2014) the aggregate to primary particle size ratio (a_a/a) for TiO_2 nanoparticles is 7.77.

b) Density

The density variation of nanofluid was calculated as [21]:

$$\rho_{nf} = \rho(1-\Phi)+\Phi.\rho_{np} \quad (13)$$

c) Specific heat

The specific heat variation of nanofluid can be calculated as [21]:

$$c_{pnf} = \frac{((1-\Phi.\rho.c_p)+(\Phi.\rho.c_{pnp})}{\rho_{nf}} \quad (14)$$

d) Thermal conductivity

The thermal conductivity variation of nanofluid can be calculated as [21]:

$$K_{nf} = \frac{K_{np}+2K_f+2(K_{np}-K_f)\Phi}{K_{np}+2K_f-(K_{np}-K_f)\Phi}K_f \quad (15)$$

7 Energy equation

The temperature distribution in circumferential and a cross fluid – film directions for Newtonian incompressible steady fluid flow can be expressed as [22]:

$$P_e\left[\bar{u}\frac{\partial\bar{T}}{\partial\theta}+\left(\frac{\bar{v}}{\bar{c}.\bar{h}}-\bar{u}\frac{\bar{y}}{\bar{h}}\frac{\partial\bar{h}}{\partial\theta}\right)\frac{\partial\bar{T}}{\partial\bar{y}}\right]=\frac{\partial^2\bar{T}}{\bar{h}^2\partial\bar{y}^2}+ \frac{\bar{\mu}_{nf}}{\bar{h}^2}N_d\left[\left(\frac{\partial\bar{u}}{\partial\bar{y}}\right)^2+\left(\frac{\partial\bar{w}}{\partial\bar{y}}\right)^2\right] \quad (16)$$

Where,

$$P_e = \frac{\rho C_{pnf}Uc^2}{K_{nf}R_{bi}} \;,\; N_d = \frac{\mu_o U^2}{K_{nf}T_o}$$

8 Heat conduction equation

The temperature distribution through the bearing material can be evaluated by solving the heat conduction equation. The following non dimensional steady state heat conduction equation was adopted in the present work [22]:

$$\frac{\partial^2\bar{T}_b}{\partial\bar{r}^2}+\frac{1}{\bar{r}}\frac{\partial\bar{T}_b}{\partial\bar{r}}+\frac{1}{\bar{r}^2}\frac{\partial^2\bar{T}_b}{\partial\theta^2}=0 \quad (17)$$

9 Fluid-Film Velocity Components

The velocity components for the lubricant flow in clearance gap in circumferential (u), axial (w) and across the oil film (v) directions can be expressed in dimensionless form as [24]:

$$\bar{u}=\frac{u}{U}=\bar{h}^2\frac{\partial\bar{p}}{\partial\theta}\left\{\int_0^{\bar{y}}\frac{\bar{y}}{\bar{\mu}_{nf}}d\bar{y}-\frac{\int_0^1\frac{\bar{y}}{\bar{\mu}_{nf}}d\bar{y}.\int_0^{\bar{y}}\frac{1}{\bar{\mu}_{nf}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}_{nf}}d\bar{y}}\right\} + \frac{\int_0^{\bar{y}}\frac{1}{\bar{\mu}_{nf}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}_{nf}}d\bar{y}} \quad (18)$$

$$\bar{w}=\frac{w}{U}=\bar{h}^2\frac{\partial\bar{p}}{\partial\bar{Z}}\left(\frac{1}{2\alpha}\right)\left\{\int_0^{\bar{y}}\frac{\bar{y}}{\bar{\mu}_{nf}}d\bar{y}-\frac{\int_0^1\frac{\bar{y}}{\bar{\mu}_{nf}}d\bar{y}.\int_0^{\bar{y}}\frac{1}{\bar{\mu}_{nf}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}_{nf}}d\bar{y}}\right\} \quad (19)$$

$$\bar{v}=\frac{v}{U}\left(\frac{R_{bi}}{c}\right)=-\bar{h}\int_0^{\bar{y}}\left\{\frac{\partial\bar{u}}{\partial\theta}+\left(\frac{1}{2\alpha}\right)\frac{\partial\bar{w}}{\partial\bar{z}}-\frac{\bar{y}}{\bar{h}}.\frac{\partial\bar{h}}{\partial\theta}\frac{\partial\bar{u}}{\partial\bar{y}}\right\}d\bar{y} \quad (20)$$

10 Boundary Conditions

The following boundary conditions are used together with the governing equations to analyze the problem of isoviscouse and thermoviscouse performance of worn journal bearing lubricated with nanolubricant:

(A). Lubricant flow field

1. At the oil supply groove $\theta = 2\pi-\phi$ $\rightarrow$ $\bar{P}=\bar{P}_s$

2. At the journal bearing edges $\bar{Z}=0$ and $\bar{Z}=1$ $\rightarrow \bar{P}=\bar{P}_{atm}=0.0$

3. At the Cavitation zone $\frac{\partial \overline{P}}{\partial \theta} = 0.0$ and $\overline{p} = 0.0$ (Reynolds boundary conditions).

(B). Thermal field

The temperature distribution through the oil film can be determined by solving the energy equation subjected to the following boundary conditions:

1. The oil film temperature in the supply groove is the mixing temperature (T_{mix}). It is assumed to be constant and can be estimated as described by [24]:

$$T_{mix} = \frac{Q_{rec}\overline{T}_r + Q_{in}\overline{T}_{in}}{Q_{rec} + Q_{in}} \quad (21)$$

Where:

$\overline{T}_r$ = Recirculation temperature

$\overline{T}_{in}$ = Inlet oil temperature

Q_{in} = Supply oil flow rate m3/sec

Q_{rec} = Recirculation flow rate (m3/sec) and is expressed as:

$$Q_{rec} = L\int_0^h u.dy$$

Which can be rearranged as follows:

$$Q_{rec} = LUc\int_0^1 \overline{u}\overline{h}\, d\overline{y} \quad (22)$$

2. The heat flux continuity on the surface between the bush and the oil film interface which yield to the followed as described by [24]:

$$\left.\frac{\partial \overline{T}_b}{\partial \overline{R}_b}\right|_{\overline{r}=1} = -\frac{K_{nf}}{K_b}.\frac{R_{bi}}{c}.\frac{1}{\overline{h}}.\left.\frac{\partial \overline{T}}{\partial \overline{y}}\right|_{\overline{y}=0} \quad (23)$$

3. The heat losses by free convection can be expressed as [22]:

$$\left.\frac{\partial \overline{T}_b}{\partial \overline{r}}\right|_{\overline{R}_b=\frac{R_{bo}}{R_{bi}}} = -\frac{h_{conv}}{K_b}R_{bi}\left(\overline{T}_{bo} - \overline{T}_a\right) \quad (24)$$

11 Bearing Parameters

The following are the most important bearing parameters which were studied through this work.

The bearing load carrying capacity is defined as the reaction force of the lubricant film. It can be expressed in non-dimensional form as [24]:

$$\overline{W}_r = \frac{c^2 W_r}{\mu_o U R_{bi}^2 L} = \int_0^1\int_0^{2\pi} \overline{P}.\cos(\theta).d\theta.d\overline{z} \quad (25)$$

$$\overline{W}_t = \frac{c^2 W_t}{\mu_o U R_{bi}^2 L} = \int_0^1\int_0^{2\pi} \overline{P}.\sin(\theta).d\theta.d\overline{z} \quad (26)$$

The total load carrying capacity of journal bearing can be evaluated as:

$$\overline{W} = \frac{c^2 W}{\mu_o U R_{bi}^2 L} = \sqrt{\overline{W}_r^{\,2} + \overline{W}_t^{\,2}} \quad (27)$$

The attitude angle between the load line and the line of centers can be expressed as [24]:

$$\phi = \tan^{-1}\left(-\frac{\overline{W}_t}{\overline{W}_r}\right) \quad (28)$$

The mean viscous friction force of worn journal bearing in non-dimensional form is described by the following equation as [24]:

$$\overline{f}r = \frac{fr}{\mu_o UL\left(\frac{R_{bi}}{c}\right)} = \int_0^1\int_0^{2\pi}\left(\frac{\overline{\mu}_{nf}}{\overline{h}} + \frac{\overline{h}}{2}\frac{\partial \overline{P}}{\partial \theta}\right)d\theta.d\overline{z} \quad (29)$$

The non-dimensional side leakage flow is determined by using the following relation [24].

$$\overline{Q}_s = \frac{Q_s L}{c\omega R_s^2} = \int_0^{2\pi}\int_0^1 \overline{h}\overline{w}\Big|_{z=0,1} d\overline{y}d\theta \quad (30)$$

The power consumed due to friction force (power loss) can be evaluated as:

$$\text{Power losses} = T_r.\omega \quad (31)$$

Where:

$$T_r = f_r.R_s \quad (32)$$

The Sommerfeld number can evaluated as:

$$S_O = \frac{\mu_o \omega L R_s}{\pi W}\left(\frac{R_{bi}}{c}\right)^2 = \frac{2\mu_o N L R_s}{60W}\left(\frac{R_{bi}}{c}\right)^2 = \frac{1}{\pi \overline{W}} \quad (33)$$

12 Results and Discussion:

The results obtained through this work have been computed for a worn journal bearing with the geometric and operating parameters presented in table (1) lubricated with a lubricant containing TiO_2 nanoparticles blended with the base oil. The mathematical model as well as the computer program written to solve the governing equations were validated by comparing the results of pressure distribution for a worn journal bearing and the maximum pressure for intact bearing obtained in the present work with that obtained numerically by Hashimoto et al. [2] (1986) and the experimental results obtained by Roy [24] as presented in figures (2)and (3). The maximum error is calculated and found to be 3%. It appears from these figures that the results are in a good agreement.

Table 1: Technical data used for the numerical simulation:

Bearing length	L = 0.08 m
Radial clearance	c = 0.0000152 m
Shaft radius	$R_s = 0.05$ m
External bearing radius	R_{bo} = 0.1 m
Inlet lubricant pressure	Pin=70000 pa
Rotation speed	N=2000 rpm
Lubricant density at inlet temperature	ρ=860 Kg/m^3
Lubricant viscosity at inlet temperature	μo=0.0277 pa.sec
Temperature - viscosity coefficient	β=0.34
Lubricant specific heat	C_o=2000 J/Kg.˚C
Lubricant thermal conductivity	K_{oil}=0.13 W/m .˚C
Bush convection heat transfer coefficient	h_{conv}=80 W/m^2.˚C
Bush thermal conductivity	K_b=250 W/m .˚C
Groove angle	18 deg
Initial lubricant temperature	T_o = 40 ^{o}C
Nanoparticle density	$\rho_{np} = 3790$ kg/m^3
Nanoparticle thermal conductivity	$K_{np} = 40$ W/m .˚C
Nanoparticle specific heat	$C_{pnp} = 765$ J/kg.^{o}C
Volume fraction of nanoparticle	$\Phi = (0.5, 1, 2)\%$

The variation of dimensionless pressure distribution in circumferential direction for intact ($\delta = 0$) and worn journal bearing with different wear depth parameters works at different eccentricity ratios(ε=0.3 and 0.7) lubricated with oil containing TiO_2 nanoparticles with particle concentration (Φ=1%) is shown in figures (4and 5). It is clear from these figures that the wear depth parameter affecting the oil film pressure in varying proportions. Figure (4) shows that the oil film pressure increases for the bearing that has higher wear depth parameter when it works at lower eccentricity ratio ($\varepsilon = 0.3$), while the situation is reversed when the bearing works at higher eccentricity ratios as can be shown from figure (5) when the bearing works at eccentricity ratios 0.7 . This is agreed with finding of Fillon et. al (2004). This is can be attributed to the increase in the active zone extension in the case of the bearing that has higher wear depth parameter and works at lower eccentricity ratio. It is clear from figure(4) that the maximum pressure increases by 16% for the worn bearing with δ=0.2 works at ε=0.3 while it becomes 49% for the one with δ=0.4 works at the same ε. From figure (5) it can be seen that the maximum pressure increases by 5% for the worn bearing with δ=0.2 works at ε=0.7 while it decreases by 20% for the one with δ=0.4 works at the same ε. The influence of wear depth parameter (δ=0,0.2 and 0.4) on the oil film temperature distribution in circumferential direction for a worn journal bearing lubricated with nanolubricant containing TiO_2 nanoparticles with particle concentration (Φ=1%) works at eccentricity ratio (0.7) can be shown in figure (6). A slightly decrease in oil film temperature can be observed for the worn bearing that has a wear defect with higher values of wear depth parameter. A percentage decrease of 2% can be observed for a worn bearing with δ=0.4 in comparison with the intact one (δ=0). This is can be attributed to the decrease of oil film pressure and its derivatives. Figure(7) shows a slight increase in oil film temperature when the worn journal bearing lubricated with nanolubricant containing different particle concentrations of TiO_2 nanoparticles. A maximum percentage increase was calculated and found to be 7.7 for a worn journal bearing with δ=0.2 when lubricated with oil containing TiO_2 nanoparticles with particle concentration (Φ=2%). This is can be attributed to the increase in oil viscosity and hence the shear rate of the nanolubricant. The variation of dimensionless load carrying capacity with the eccentricity ratio (ε) for intact and worn journal bearing lubricated with a lubricant containing TiO_2 nanoparticles with different particle concentrations (Φ) is shown in figure (8-a, b, c and d). It can be deduced from this figure that the worn journal bearing with higher value of dimensionless wear parameter (δ) supporting lower load in comparison with that which has lower dimensionless wear depth parameter. This is can be attributed to the decrease of the oil film pressure in this case due to the increase in oil film thickness for the worn journal bearing that has higher wear depth parameter. The decrease percentages in load carrying capacity were calculated and presented in figure (8-E). This figure also shows that the load carrying capacity was enhanced when the worn bearing lubricated with nanolubricant that has higher particle concentration (Φ) works at the same value of (ε). This is can be explained by knowing that the oil viscosity is highly affected by the nanoparticles added to the base oil which leads to higher oil film pressure. The increase percentages in load carrying capacity of a worn bearing due to the addition of TiO_2 nanoparticles to the base oil are calculated and presented in figure (8-F) . Figure (9-a, b, c and d) explains the variation of Sommerfeld number with eccentricity ratio for intact and worn journal that has various values of (δ) lubricated with nanolubricant that contained different particle concentrations (Φ) of Tio2 nanoparticles. This figure shows that Sommerfeld number increases for the worn journal bearing that has higher wear depth parameter .This is due to the decrease in load carrying capacity of the bearing in this case. The increase percentages in Sommerfeld number for worn journal bearing that has higher wear depth parameter have been calculated and presented in figure (9-E). It is also obviously shown from this figure that the Sommerfeld number decreases when the bearing works at higher eccentricity ratios and lubricated with nanolubricant that contains higher particle concentration of TiO_2 nanoparticles. This is due to the effect of these parameters (ε and Φ) in increasing the load carrying capacity of the bearing as discussed previously. The decrease percentages in Sommerfeld number in this case have been calculated and presented in figure (9-F). The variation of non-dimensional oil side leakage with eccentricity ratio for a worn journal bearing that has different values of wear depth parameter (δ) lubricated with nanolub-ricant containing different TiO_2 nanoparticle concen-trations (Φ) is illustrated in figure (10-a, b, c and d). This figure shows obviously that the oil side leakage increases for the bearing that has higher value of (δ). This is can be attributed to the increase in oil film thickness in this case. The oil side leakage is strongly dependent on the oil film thickness. The percentage increase of non-dimensional side leakage under wear effect has been calculated and presented in figure (10-E). It can also be seen from this figure that the higher the particle concentration (Φ) of TiO_2 nanoparticles added to the base oil the lower is the non-dimensional side leakage of the oil. This is can be attributed to the higher oil viscosity in this case which causes a decrease in axial oil velocity . Figure (10-F) shows that the maximum Percentage decrease in side leakage for a worn journal bearing that has a wear depth parameter (δ=0.1) lubricated with nanolu-bricant containing different particle concentrations (Φ) of TiO_2 nanoparticles is 17%. The variation in dimensionless friction force with eccentricity ratio for a worn bearing that has different values of wear depth parameter (δ) when lubricated with lubricant containing different (Φ) of Tio_2 nanoparticles can be shown in figure (11-a, b, c and d). It is obviously shown from this figure that the dimensionless friction force decreases for the worn bearing that has higher wear depth parameter. This is can be attributed to the decrease in oil shearing rate due to the increase in oil film thickness. The maximum percentage decrease in friction force under the wear effect has been calculated and found to be 21% as can be shown in figure (11-E) . Also, it was found that the increase in particle concentration (Φ) of the nanoparticles added to the base oil causes a rise in dimensionless friction force due to the increase in oil viscosity and hence the oil shearing rate. A worn journal bearing lubricated with nano-lubricant containing TiO_2 nanoparticles with (Φ=1%) is considered in this case. Figure (11-F) shows the increase percentage in dimensionless friction force for a worn journal bearing with (δ=0.1) lubricated with nanolubricant containing different particle concentrations of (TiO_2) nanoparticles. This figure shows that the maximum increase in friction force became about 20% to 40% when TiO_2 nanoparticles added to the base oil by 0.5% to 1% respectively. It is clear from this figure that adding nanoparticles with higher particle concentrations (Φ=2%) causes higher values of friction force which is undesirable from the power loss point of view as may discussed later. This is can be attributed to the increase in oil viscosity and hence the film shear stress in this case. Figure (12-a, b, c and d) shows the variation of friction coefficient with eccentricity ratio for a worn journal bearing that has various values of wear depth parameters lubricated with lubricant containing TiO_2 nanoparticles with different (Φ). This figure shows that the coefficient of friction increases for the worn journal bearing that has higher wear depth parameter as a result of the decrease in load carrying capacity in this case. It is slightly decreases when the bearing lubricated with nanolubricant that has higher particle concentrations of the nanoparticles. This can attributed to the increase in the load carried by the bearing when it is lubricated with nanolubricant with higher particle concentration of nanoparticles. The percentage of increase in coefficient of friction for the worn journal bearing that has different wear depth parameters has been calculated and presented in figure (12-E).The percentage increase in friction coefficient becomes higher as the worn journal bearing works at higher eccentricity ratio. The figure shows that the percentages increase in coefficient of friction for a worn journal bearing that has δ=0.1,0.2 and 0.4 works at eccentricity ratio of 0.3 are between 2%,4% and 7% respectively, while it becomes 7%, 13% and 45% respectively when it works at eccentricity ratio of 0.8. Figure (12-F) shows the Percentage decrease in friction coefficient for a worn journal bearing that has δ=0.1 lubricated with nanolubricant containing different particle concentrations of nanoparticles. It is obvious from this figure that percentage decrease in coefficient of friction becomes lower when the bearing works at higher eccentricity ratios. This is due concomitant rise in load and friction force in this case. A maximum decrease in the coefficient of friction is 11%. Figure (13-a, b, c and d) explains the change of attitude angle with eccentricity ratio for a worn journal bearing with various values of (δ) lubricated with nanolubricant containing different particle concentrations (Φ) of Tio_2 nanoparticles. It can be seen from this figure that the worn bearing with higher values of bearing depth parameter (δ) has smaller attitude angle. Also, it can be shown from this figure that adding the nanoparticles to the base oil causes a decrease in attitude angle. The decrease percentage in attitude angle when lubricating the bearing with nanolubricant that has higher particle concentration has been calculated and presented in figure (13-F).

Figure (14-a, b, c and d) shows the alteration of power loss with the eccentricity ratio for a worn bearing with various values of (δ) lubricated with lubricant contained TiO_2 nanoparticles with different (Φ). It is clear from this figure that the power loss of the worn bearing decreases for the bearings with higher values of (δ). A maximum percentage decrease reaches 22% as shown in figure (14-E). This is can be attributed to the decrease in friction force induced in this case. Also, it was found that the increase in particle concentration of TiO_2 nanoparticles (Φ) added to the base oil causes an increase in power losses due to the increase in oil viscosity and hence the friction force. The increase percentage in power losses for a worn journal bearing with (δ=0.1) lubricated with nanolubricant containing different particle concentrations of nanoparticles can be shown in figure (14-F).

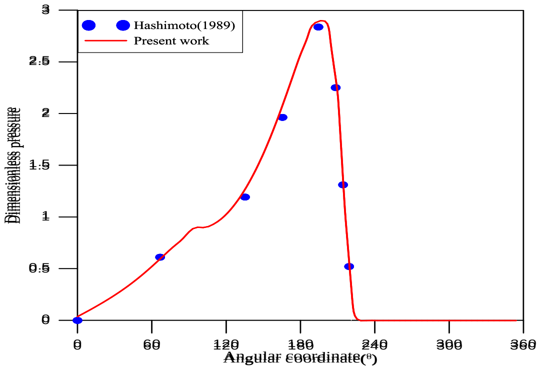


Figure (2): Comparison between the pressure distributior obtained in the present work with that obtained by Hashimc [2].

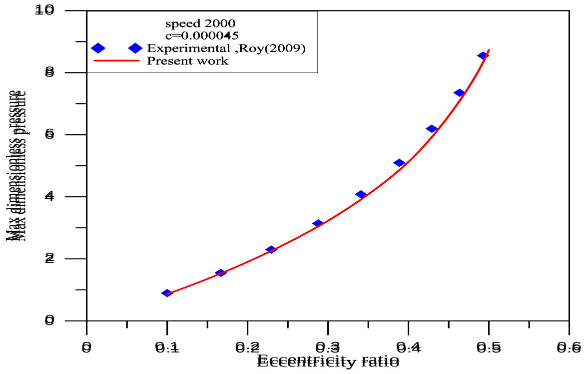


Figure (3): Comparison between the maximum pressures with eccentricity ratio obtained in the present work with that obtained by Roy [24].

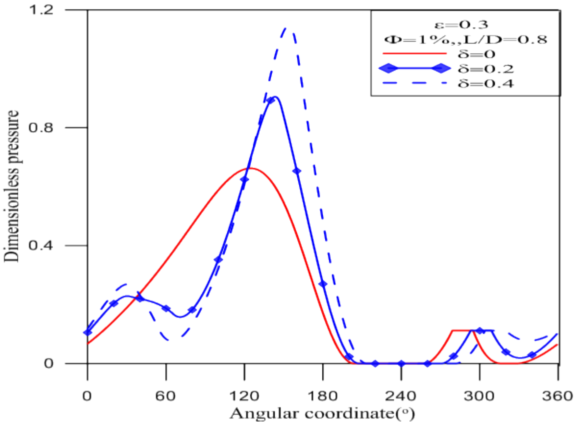


Figure (4): Dimensionless pressure distribution in circumferential direction for a worn journal bearing with nanolubricant works at ε=0.3.

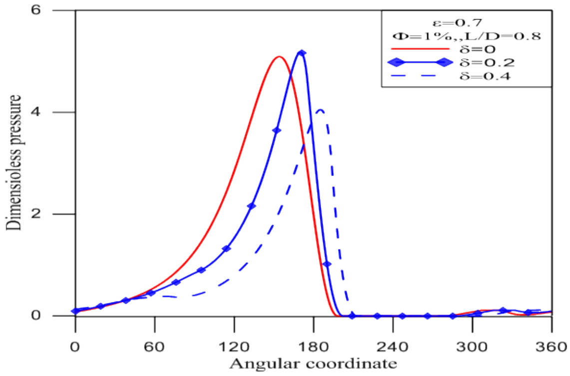


Figure (5): Dimensionless pressure distribution in circumferential direction for a worn journal bearing with nanolubricant works at ε=0.7.

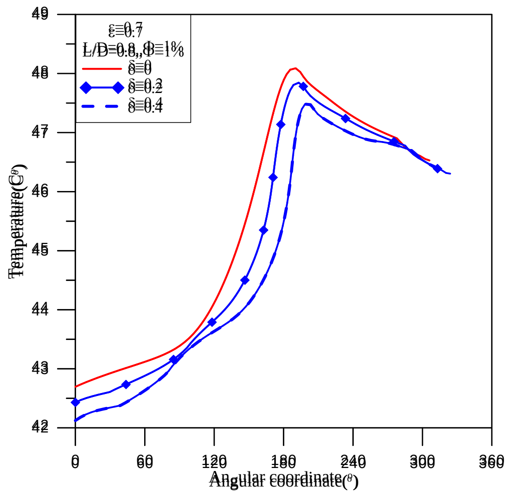


Figure (6): Temperature distribution in circumferential direction for a worn journal bearing with nanolubricant including (Φ=1%) particle concentration of Tio2 nanoparticle

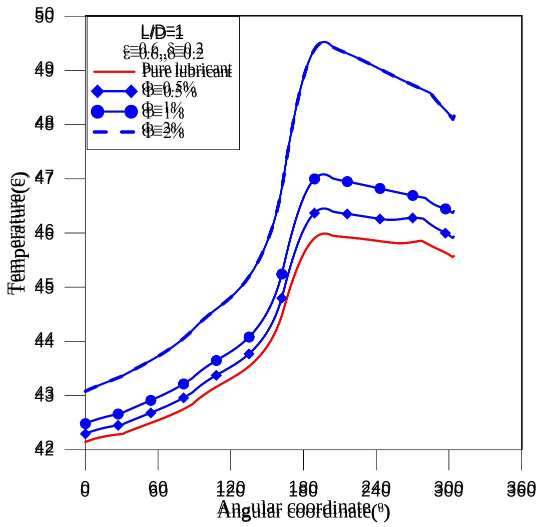
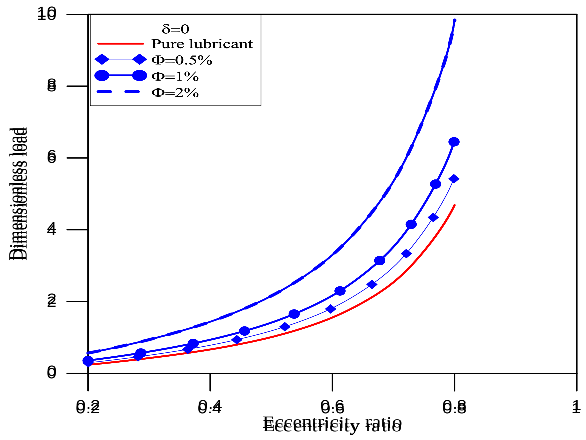
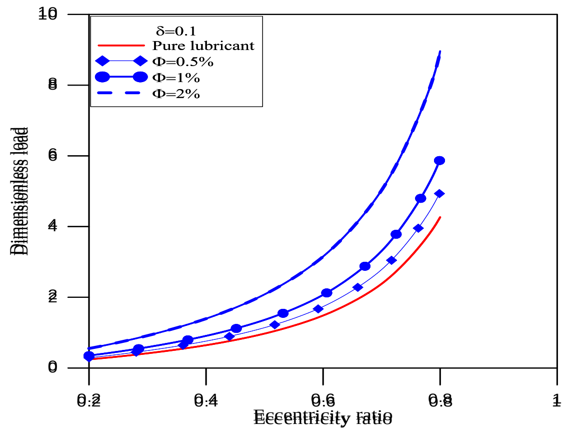


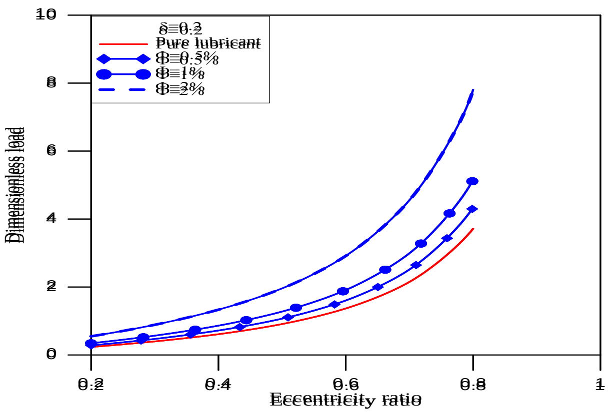
Figure (7): Temperature distribution in circumferential direction for a worn journal bearing lubricated with oil containing different particle concentrate-ions of nanoparticles.



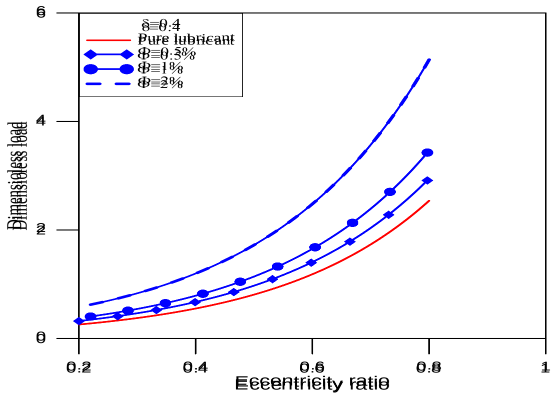
A



B



C



D

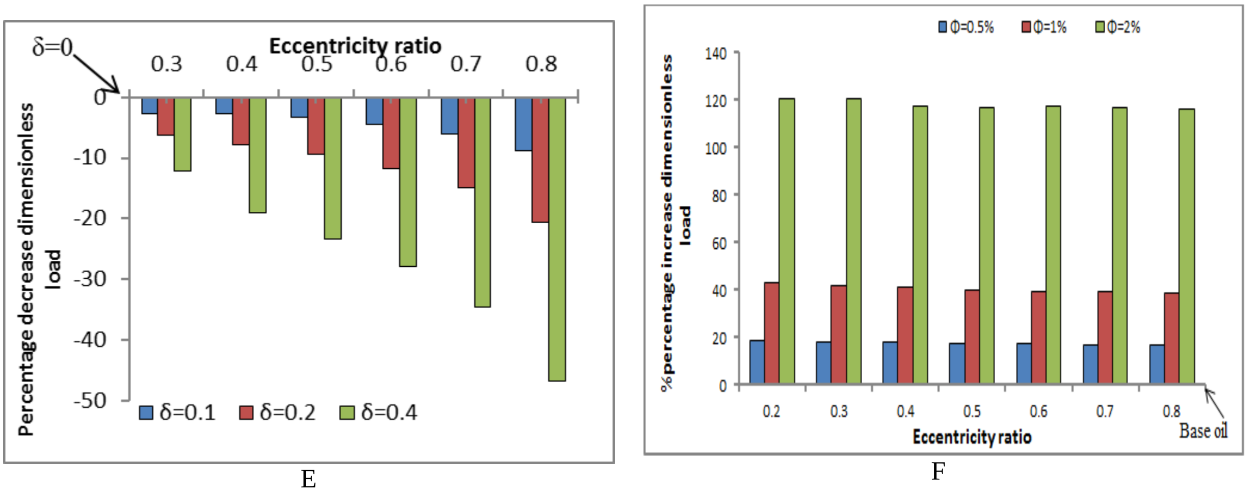
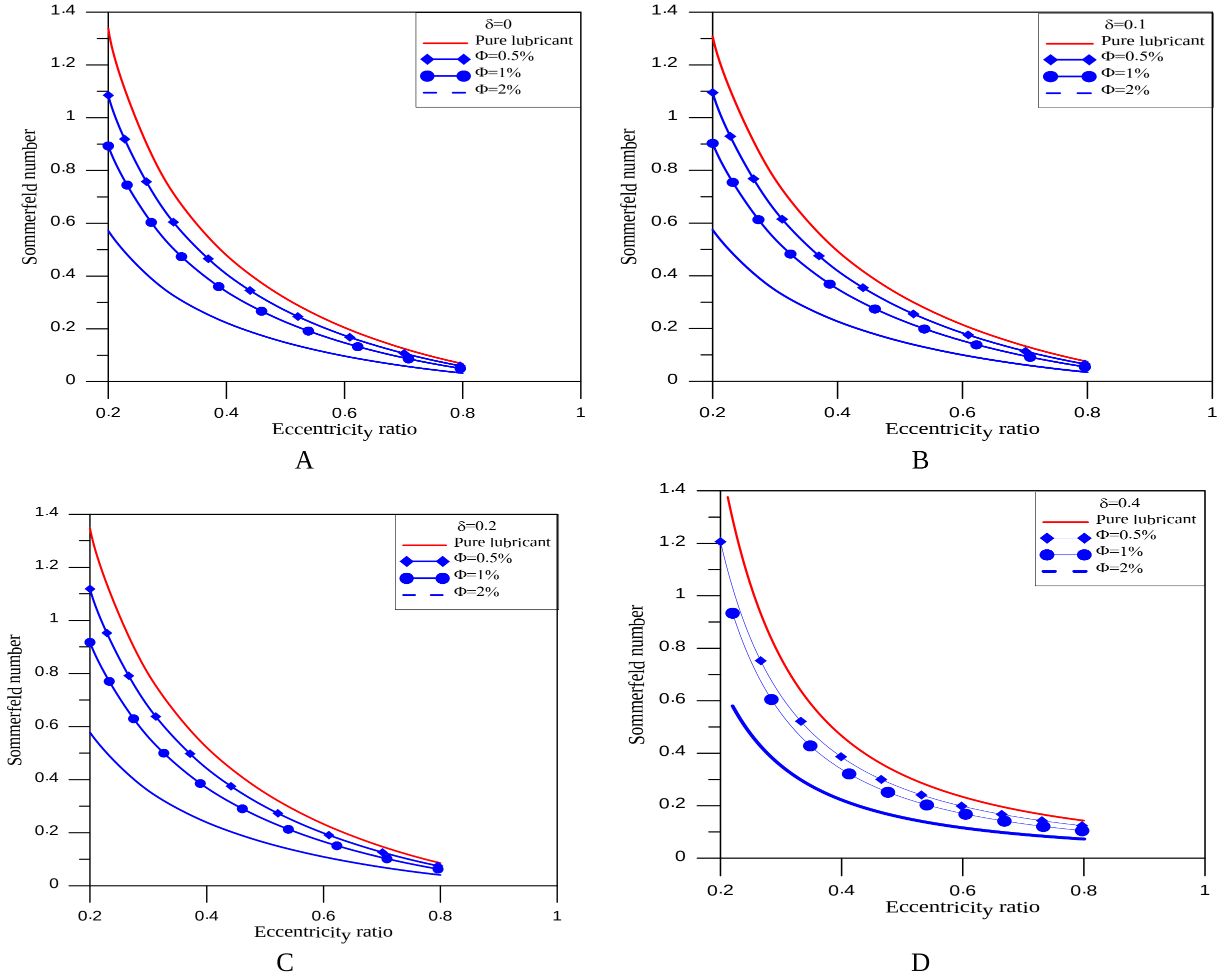


Figure (8): Dimensionless Load verses eccentricity ratio for a worn journal bearing lubricated with nanolubricant including different particle concentration of TiO_2 nano particles**.**



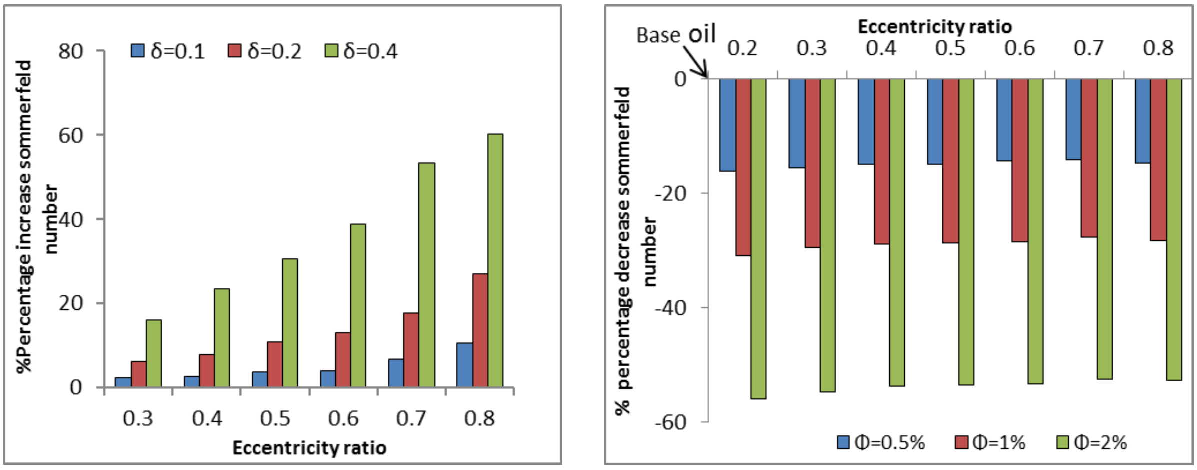
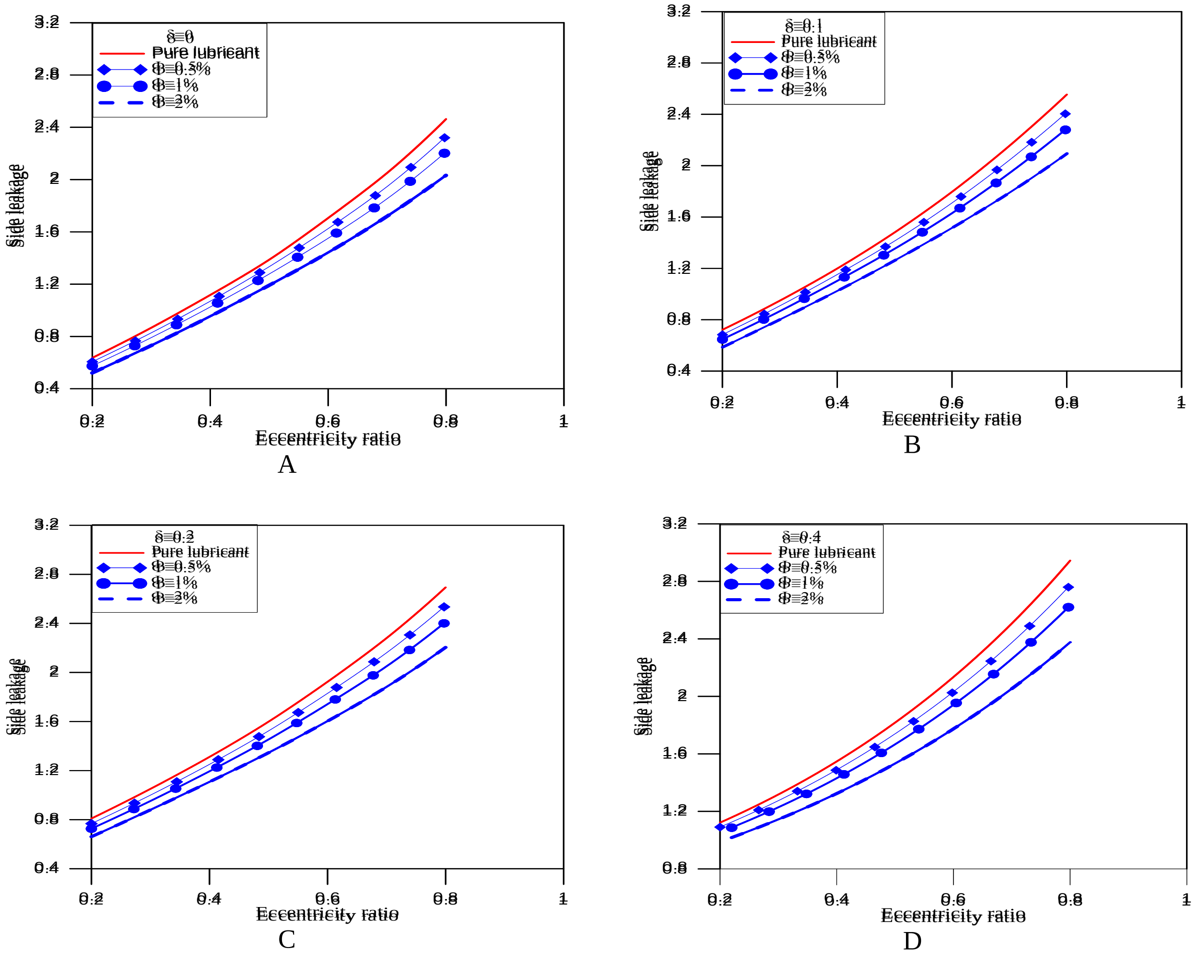
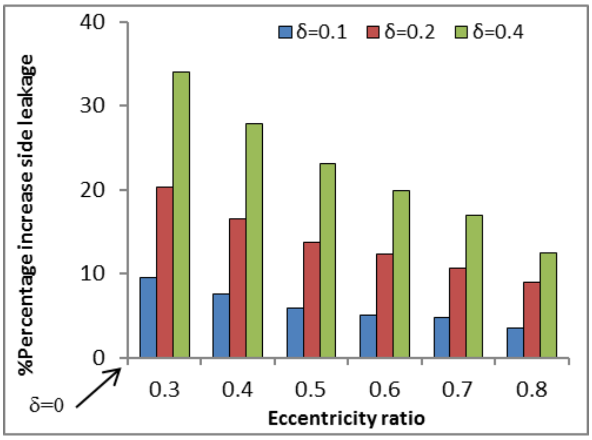


Figure (9): Sommerfeld number versus eccentricity ratio for a worn journal bearing lubricated with nano lubricant including different particle concentration of TiO_2 nano particles.





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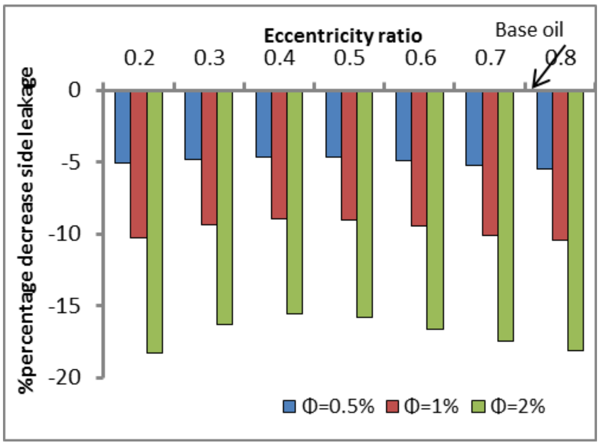
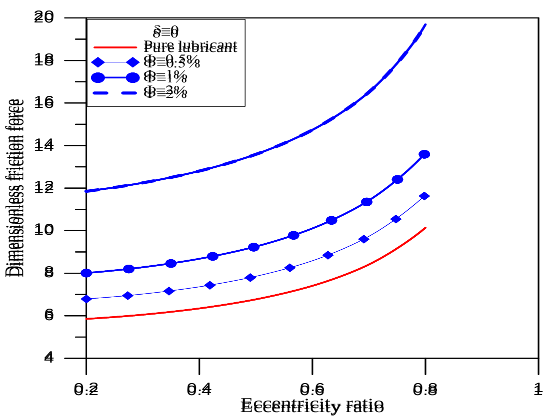


F

Figure (10): Side leakage versus eccentricity ratio for a worn journal bearing lubricated with nano lubricant including different particle concentration of Tio_2 nano particles.



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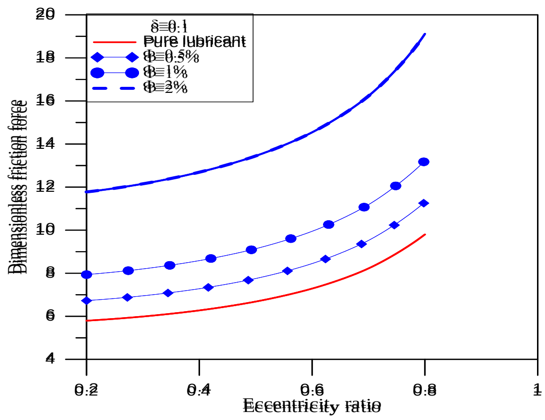

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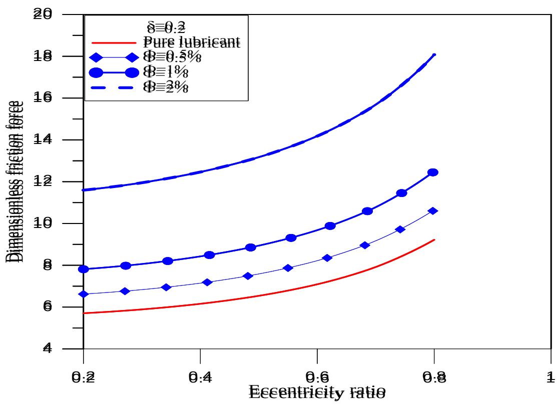

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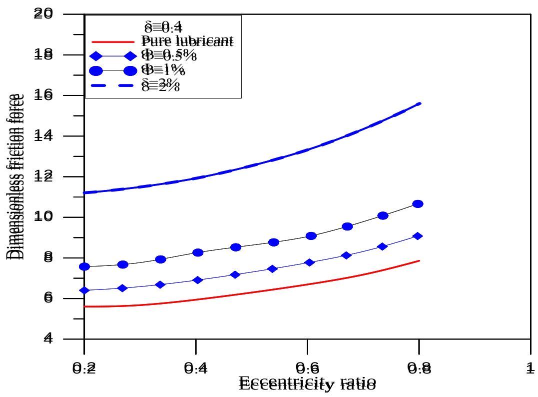

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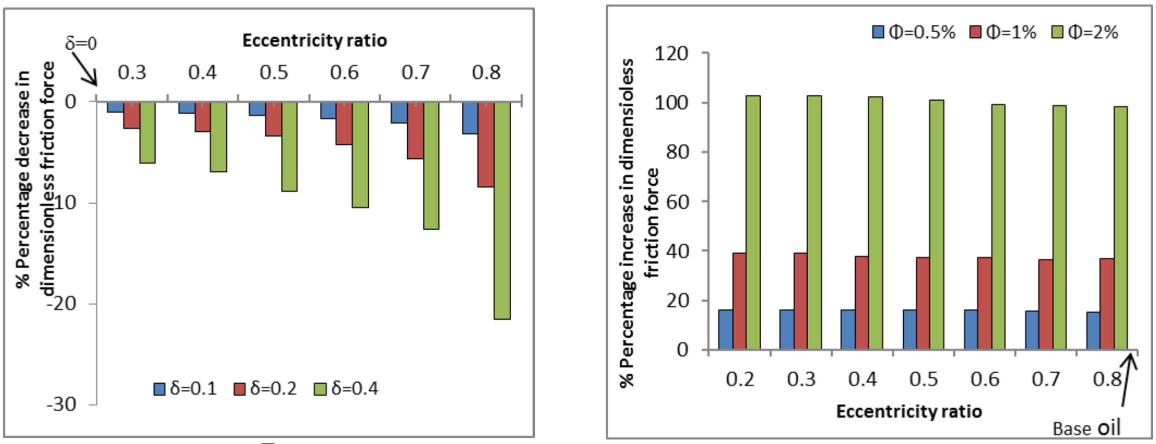
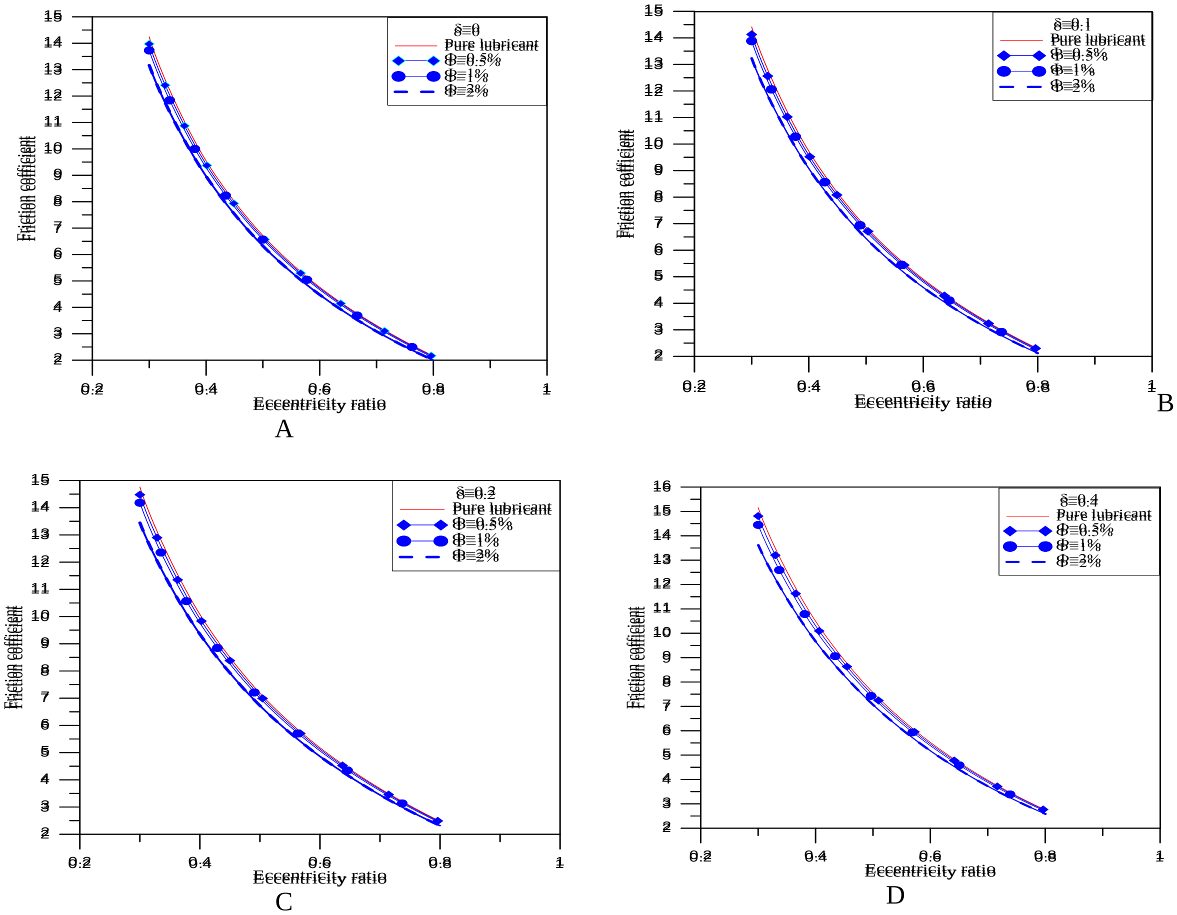
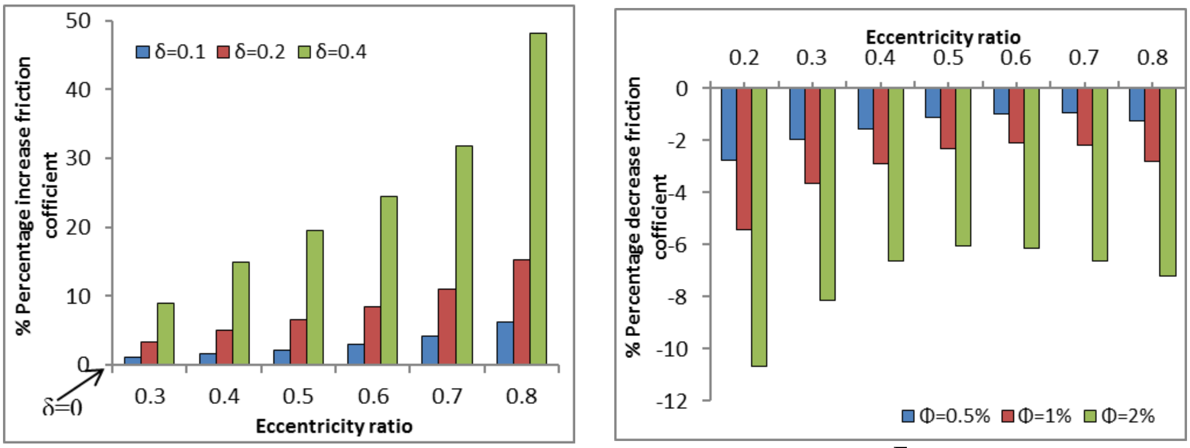


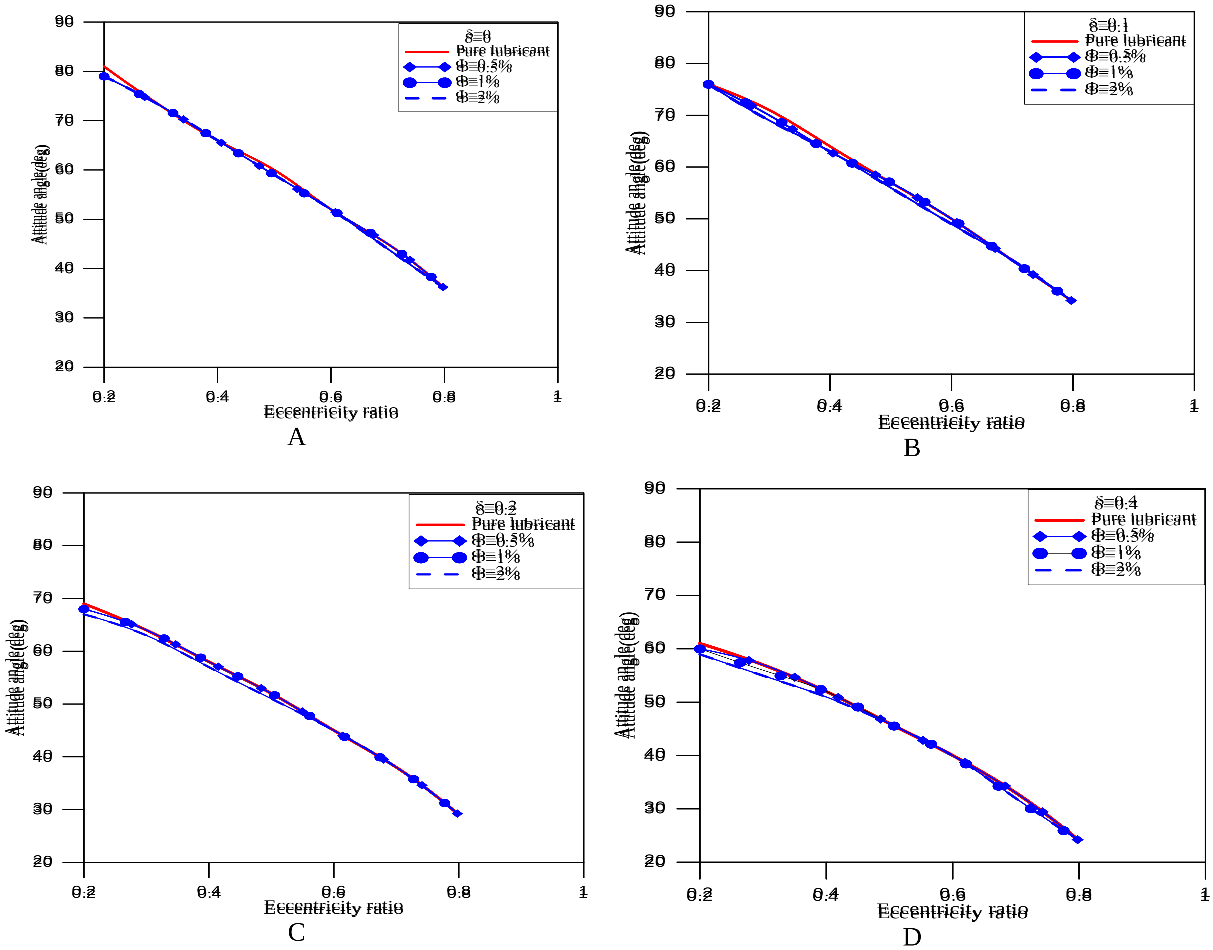
Figure (11): Dimensionless friction force versus eccentricity ratio for a worn journal bearing lubricated with nano lubricant including different particle concentration of Tio_2 nano particles.

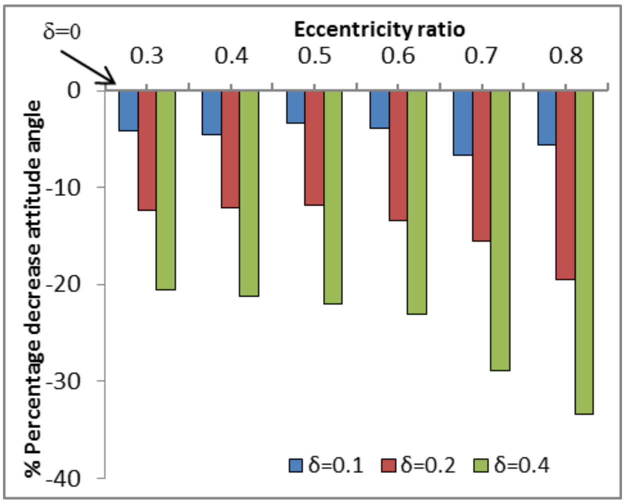




E F

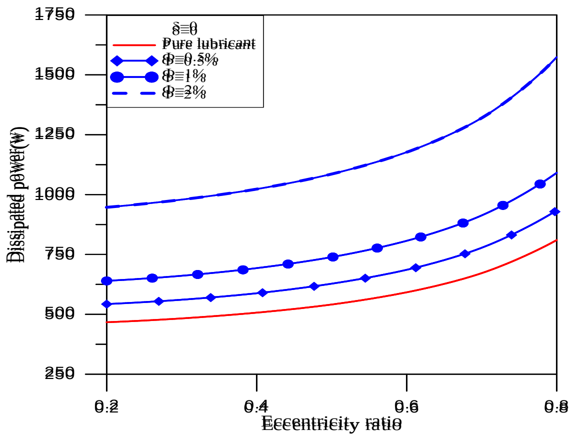
Figure (12): Friction coefficient versus eccentricity ratio for a worn journal bearing lubricated with nano lubricant including different particle concentration of Tio_2 nanoparticles.



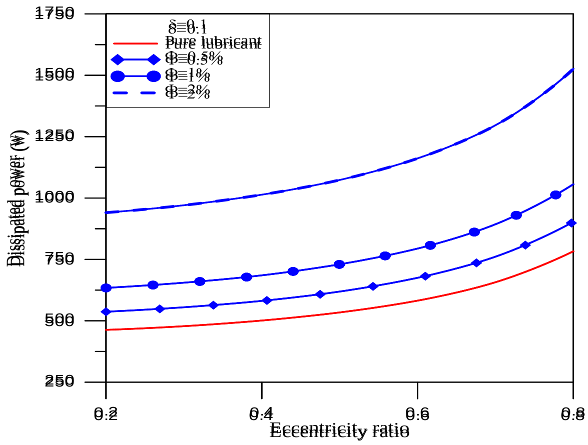


E

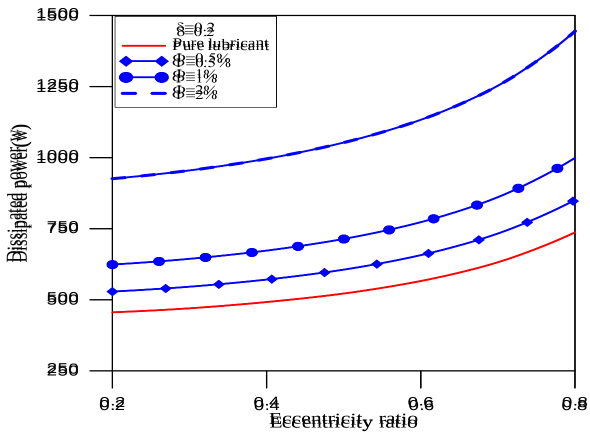
Figure (13): Attitude angle versus eccentricity ratio for a worn journal bearing lubricated with nano lubricant including different particle concentration of Tio_2 nano particles.



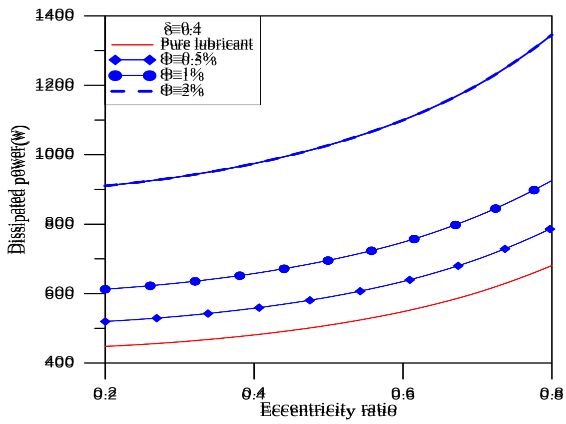
(a)



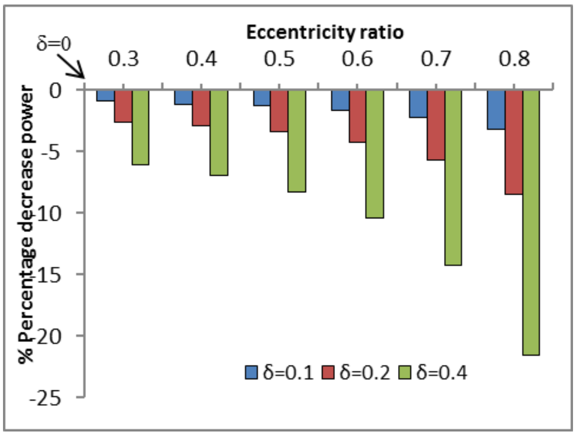
(b)



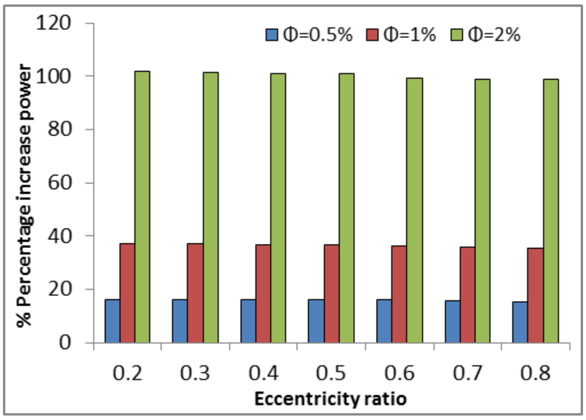
(c)



(d)



E



F

Figure (14): Power versus eccentricity ratio for a worn journal bearing lubricated with nano lubricant including different particle concentration of Tio_2 nano particles.

13 Conclusions

According to the discussion of the results developed previously the following conclusions can be drawn.

1. Effect of wear depth:-

a. The load carrying capacity decreased and the oil side leakage was increased for the worn bearing that has higher wear depth parameter. For a worn journal bearing that has δ=0.4, ε=0.8 and Φ= 1% the load carrying capacity decreased by (47%), while the side leakage increase by 13% in comparison with intact bearing.

b. The power losses decreases for a worn bearing that has a higher value of (δ). A decrease of 21% in power losses has been obtained for a worn journal bearing work at the same above conditions.

2. Effect of Nanoparticles :-

For the effect of the nanolubricant on the performance of worn journal bearing it was found that:-

a. The load carrying capacity increased and the oil side leakage was decreased when a higher particle concentration of nanoparticles added to the base oil. For a worn journal bearing that has δ=0.1, ε=0.8 and Φ= 1% the load carrying capacity increased by (39%), while the side leakage decreases by (10.5%) in comparison with that of worn journal bearing lubricated with pure oil.

b. Slight increase in oil film temperature by 1%, 2.4% and 7% when a worn journal bearing that has δ=0.4,ε=0.6 lubricated with nanolubricant containing nanoparticles with concentrations 0.5%, 1% and 2% respectively in comparison with that lubricated with pure oil.

c. Side leakage flow decreases for a worn journal bearing that has higher values of TiO_2 nanoparticles concentration. The maximum decreases in side leakage for bearing with δ=0.1, works at ε=0.8 lubricated with oil containing 1% TiO_2 nanoparticles was found to be 10%.

d. The coefficient of friction decreases for the bearing lubricated with nanolubricant containing higher particle concentration of the TiO_2. For the bearing works at eccentricity ratio less than 0.5 the maximum decrease is 10% .

e. The addition of TiO_2 nanoparticle increases the power losses of a worn journal bearing work at any eccentricity ratio. The maximum increases in power losses for a worn journal bearing with δ=0.1, works at ε=0.8 lubricated with oil containing TiO_2 nanoparticles with Φ=1% was found to be 36%.

f. Adding a higher particle concentration of TiO_2 nanoparticles to the base oil has undesirable effect on the performance of the bearing from the power loss point of view.

14 Nomenclature

a	Radius of primary nanoparticles (nm)
a_a	Radius of aggregate nanoparticles (nm)
c	Radial clearance (m)
C_p	Specific heat of pure lubricant (J/kg. ˚C)
C_{pnf}	Specific heat of nanolubricant (J/kg. ˚C)
C_{pnp}	Specific heat of nanoparticle (J/kg. ˚C)
d	Wear depth
D	Fractal index
$\overline{fr}$	Dimensionless friction force
$\overline{h}$	Non-dimensional oil film thickness
K_b	Thermal conductivity of the bush (W/m.˚C)

K_f Thermal conductivity of the lubricant (W/m.˚C)

K_{nf} Thermal conductivity of the nanolubricant (W/m.˚C)

L Bearing length (m)

N Rotation speed (rpm)

N_d Dissipation number

$\bar{P}$ Non-dimensional oil film pressure

P_e Peclet number

$\bar{Q}_s$ Non-dimensional Side leakage flow

R_s Journal Radius (m)

R_b Bearing (bush) radius (m)

R_{bi} Bush inner radius (m)

R_{bo} Bush outer radius (m)

$\bar{T}$ Dimensionless oil film temperature

T_0 Inlet temperature ($^{\circ}C$)

$\bar{W}$ Dimensionless load carrying capacity of the worn journal bearing

$\bar{W}_r$ Dimensionless radial load component

$\bar{W}_t$ Dimensionless tangential load component

x, y, z Coordinate system (m)

15 Greek symbols

μ Pure lubricant viscosity (Pa.sec.)

$\bar{\mu}$ Dimensionless pure lubricant viscosity = $\frac{\mu}{\mu_o}$

μ_{nf} Nanolubricant viscosity (Pa.sec.)

$\bar{\mu}_{nf}$ Dimensionless nanolubricant viscosity= $\frac{\mu_{nf}}{\mu_o}$

μ_0 Inlet Lubricant viscosity (Pa.sec)

β Thermo-viscous coefficient (1/˚C)

ε Eccentricity ratio

ρ Pure lubricant density (kg / m^3)

ρ_{nf} Nanolubricant density, kg / m^3

ρ_{np} Nanoparticle density, kg / m^3

ϕ Attitude angle, Deg.

θ Angular coordinate, Deg.

Φ Concentration of nanoparticle, %

δ Dimensionless wear depth

ω Journal rotational speed, rad / sec

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الخصائص الهيدروديناميكية الحرارية للمساند المقعدية المتاكلة المزيتة بزيوت حاوية على دقائق نانوية

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الخلاصة

تم في هذا البحث تمثيل المسند المقعدي المتاكل لمناقشة تاثير اضافة دقائق نانوية من مادة ثاني اوكسيد التيتانيوم (TiO_2) الى الزيت الاساس على الاداء الحراري لتلك المساند.تم اجراء دراسة موسعة لتاثير العوامل المؤثرة على الاداء الحراري للمساند المقعدية المتاكلة مثل نسبة اللاتمركز(ε)و معامل التاكل (δ) ونسبة المواد النانوية المضافة الى الزيت الاساس (Φ) . تم اجراء التحليل العددي للمسالة باعتماد طريقة الفروقات المحددة لحل المعادلات الحاكمة للمسالة والمتمثلة بمعادلة رينولدز المعدلة ومعادلة الطاقة ومعادلة انتقال الحرارة بالتوصيل الحراري بالاضافة الى معادلة سمك طبقة الزيت المحورة لتاخذ بنظر الاعتبار تاثير تاكل المسند عليها وذلك لدراسة الفوائد من استعمال الدقائق النانوية الى الزيت الاساس على اداء تلك المساند. تم اعتماد النموذج الرياضي العائد للباحث (Krieger-Dougherty) لتوصيف اعتماد لزوجة الزيت على نسبة المواد النانوية المضافة. تم التحقق من صحة النموذج الرياضي علاوة على البرنامج الحاسوبي الذي اعد لغرض حل المعادلات الحاكمة عن طريق مقارنة بعض النتائج المستحصلة في هذا البحث مع النتائج العددية المنشورة من قبل الباحث Hashimoto(1986) بالنسبة للمساند المتاكلة والنتائج العملية المنشورة من قبل الباحث Roy(2009) . تم احتساب الانحراف في النتائج الكمستحصلة في البحث الخالي وتلك المنشورة من قبل الباحثين المذكورين اعلاه ووجدت لاتتجاوز 3%. اظهرت النتائج المستحصلة في هذا البحث ان اضافة مانسبته 0.5%و 1% من المادة النانوية الى الزيت الاساس ادت الى زيادة قابلية المسند لتخمل الاخمال بنسبة 20% و 40% على التوالي بينما انخفضت كمية التسرب الجانبي للزيت بنسبة 2.75 و 5.7% بالمقارنة مع المساند المقعدية المزيتة بزيوت غير حاوية على اضافات . اظهرت النتائج ايضا بان اضافة المادة النانوية بنسب قد تصل الى 2% الى الزيت الاساس له تاثير ضار على اداء المساند المقعدية المتاكلة حيث ازدادت قيمة الطاقة المستهلكة بشكل كبير.