



EFFECT OF LUBRICANTS CONTAINING DIFFERENT NANO-PARTICLES ON THE PERFORMANCE OF WORN JOURNAL BEARINGS

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ABSTRACT:

The present work deals with the isothermal analysis of worn journal bearing lubricated with Nano-lubricants. The bearing wear affect the oil film thickness and causes a decrease in load carrying capacity of the journal bearing. On the other hand, many works show that the addition of Nano- particles to the oil enhances its viscosity. The main goal of the present work is to investigate the ability of the Nano-lubricant to correct the wear effect on the performance of such bearings. The modified Reynolds equation was solved using finite difference technique to obtain the pressure distribution in the clearance gap of the worn journal bearing. Dufran's model was used to include the effect of wear on the thickness of the oil film. The effect of adding three different types of Nano-particles namely, Dimond, CuO and TiO₂ to the base oil has been investigated. Bearing static characteristics in terms of load capacity, friction force, end leakage and power loss for different eccentricity ratios have been investigated. The mathematical model has been validated for the pressure results obtained in the present work with that obtained by other workers and the results found to be in a good agreement. The results obtained in the present work show that the load carrying capacity enhances by about 40%, 9% and 3% for the bearing with wear depth parameter of 0.4, eccentricity ratio of 0.7 lubricated with oil containing TiO₂, CuO and Dimond Nano-particles respectively in comparison with pure oil lubricated bearing.

KEYWORDS Nano-lubricant, worn bearing, static performance

تأثير الزيوت الحاوية على دقائق نانوية على تصحيح تأثير البلى في المساند المقعدية الاسطوانية

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الخلاصة:

تتضمن هذه الدراسة تحليلاً للمساند المقعدية التي تعمل بزيوت حاوية على دقائق نانوية. من المعروف جيداً ان قابلية تحمل المساند المقعدية تقل بسبب بليان سطحه. من ناحية اخرى اثبتت دراسات متعددة ان اضافة مواد نانوية الى الزيت الاساس تساعد على رفع لزوجته. الهدف الرئيسي لهذا العمل هو مناقشة قابلية الدقائق النانوية المضافة على قابلية تصحيح تأثير البلى في المساند المقعدية الاسطوانية عن طريق دراسة تأثيرها على اداء تلك المساند. تم حل معادلة رينولدز المعدلة للاخذ بنظر الاعتبار تأثير البلى على شكله الهندسي باستخدام طريقة الفروقات المحددة وذلك للحصول على توزيع ضغط الزيت داخل فجوة المسند. تم استخدام النموذج المقترح عن طريق (Dufran) الخصائص السكونية للاخذ بنظر الاعتبار تأثير البلى على سمك طبقة الزيت. تم دراسة تأثير ثلاثة انواع من المواد النانوية وهي الالمامس وثاني اوكسيد التيتانيوم واوكسيد النحاس. تم دراسة الخصائص السكونية للمسند بدلالة قابلية تحميله للحمل وقوة الاحتكاك والتسرب الجانبي للزيت وخسائر الطاقة عندما يعمل المسند في قيم مختلفة من نسب اللاتمرکز. اوضحت النتائج المستحصلة في هذه الدراسة بان قابلية المسند المقعدية المتأكل قد زادت بنسب تراوحت 50% و 23% و 3% عند اضافة دقائق نانوية من ثاني اوكسيد التيتانيوم واوكسيد النحاس والالمامس على التعاقب بالمقارنة مع تلك المستحصلة عند تزيت المسند بزيت لا يحتوي على دقائق نانوية.

INTRODUCTION

Hydrodynamic journal bearings used in a high speed turbomachinery, such as turbine generators, are generally used for long period of times. The bearing wears progressively due to the higher friction force induced during start stop periods. The local change in the bearing geometry due to wear strongly affect the bearing performance. It is well known that the addition of Nano-particles to the base oil lubricated the journal bearing enhances the oil viscosity and the load carried by the bearing. Wear phenomenon, was a real problem for bearing users which must be taken within the industry. **Duckworth and Forrester (1957), Forrester (1960) , Katzenmeier (1972) . Dufrane et al. (1983)** analyzed a power plant worn journal bearing. Suitable geometrical model to investigate the worn region of a bearing has been proposed. It has been shown that the wear occurs symmetrically at the bearing bottom. **Hashimoto et al. (1986)** discussed the effect of the bearing wear on its pressure field. Stability of the bearing was damaged due to the bearing wear and it is less sensitive to defect at the weak L/D ratio. **Vaidyanathan and Keith,(1991)** studied the effect of wear on the bearing parameters such as friction, pressure and the Sommerfeld number which were presented against the bearing eccentricity ratios. **Kumar and Mishra (1996)** studied the effect of different wear depth on the steady state behavior of worn non-circular hydrodynamic bearings using Constantinescu's turbulent theory. It was concluded that bearing wear has a significant effect on its steady state characteristics. **Rozeanu and Kennedy, (2001)** show three different wear zones with different wear mechanism for each zone. An experimental model for each wear mode is required and each wear mode requires different remedy to achieve wear reduction. **Fillon and Bouyer(2004)** analyzed a worn journal bearing thermally considering cavitation effect. **Papadopoulos et.al (2008)** predict the bearing radial clearance theoretically using response measurements of the rotor at the middle point of the rotor. Wear was measured at two different speeds and wear defects. The enhancement of lubricant characteristics due to the addition of various additives has been studied by many workers. Many workers show that the suspended solid particles added to the pure oil affects the rendering characteristics of the bearings. **Narayanan et al.(1995)** discuss the statically and dynamically circular hydrodynamic bearings with un-deformed housing operating with micro polar fluids. **Nair (2004)** investigated the elasto-hydrodynamic lubrication of plain journal-bearing operating with micropolar lubricant. **Elsharkawy (2005) studied** the finite journal bearings lubricated by oil dispersed with solid additives using suitable rheological model. **Wu et al.(2007)** studied the tribological performance of oils with nano particle additives. Engine oil (SAE30 LB51153) viscosity has been measured when Nano- particles of different materials such as CuO, TiO₂ and diamond have been added to the base oil. It has been shown that the oil viscosity increases by 40% for the Nano-lubricant with TiO₂ Nano-particles. **Nair, et al. (2009)** discussed the statically operating characteristics of a journal bearing lubricated with nano lubricant in terms of load carrying capacity ,attitude angle and end leakage. **Binu.et al. (2014)** studied the influence of addition TiO₂ nanoparticles lubricant on the load carried by a journal bearing. The lubricant viscosity was modeled using a modified Krieger-Dougherty viscosity model. **Gunnuang.et al. (2014)** investigated the influence of adding Al₂O₃ nanoparticles to oil lubricating the journal bearing on its performance. Carreau viscosity model was used to represent the viscosity for SAE10W50 oil blended with Al₂O₃ nanoparticles. **Solghar (2015)** discussed the performance of purely oil lubricated journal bearing and that lubricated by oil blended with Al₂O₃ nanoparticles. It is clear from the above survey of literatures that the problem of using nanolubricant to correct the effect of wear in journal bearing is not studied which is the main goal of the present work.

Geometry of the bearing

Figure (1) shows the geometrical model of a worn journal bearing considered in the present work. The shaft with radius (R_s) rotates at a constant angular velocity about its center. The bearing has a radius of (R_b).

Mathematical Consideration

The following governing equations are used to represent the mathematical model used in this work.

Reynolds equation

The following non-dimensional Reynolds equation for steady, laminar, and isoviscouse flow was adopted to obtain the oil film pressure in the clearance gap of worn journal bearing **Hashimoto et al.(1986)**.

$$\frac{\partial}{\partial \theta} \left(\bar{h}^3 \frac{\partial \bar{P}}{\partial \theta} \right) + \left(\frac{D}{L} \right)^2 \frac{\partial}{\partial \bar{z}} \left(\bar{h}^3 \frac{\partial \bar{P}}{\partial \bar{z}} \right) = \frac{\partial \bar{h}}{2 \partial \theta} \quad (1)$$

Thickness of the Oil film:

Thickness of the oil film for the non-worn journal bearing can be expressed as.

$$\bar{h} = 1 + \varepsilon \cos(\theta) \quad (2)$$

Dufran's model for the oil film thickness was adopted to take the effect of the change in bearing geometry on the thickness of the oil film into consideration. **Hashimoto et al.(1986)**.

$$\bar{h} = \delta_o + \varepsilon \cos(\theta) - \cos(\theta + \varphi) \quad (3)$$

Where:

$$\bar{h} = \frac{h}{c}, \quad \varepsilon = \frac{e}{c}, \quad \delta_o = \frac{d}{c}$$

The starting and ending points of the worn region θ_s and θ_f can be determined following **Hashimoto et al.(1986)**.

$$\cos(\theta + \varphi) = \delta_o - 1 \quad (4)$$

Boundary Conditions

The solution of the governing equations require the use of the following conditions:

1. The pressure equals to the supply pressure at the oil supply groove i.e at $\theta = 2\pi - \phi \rightarrow \bar{P} = \bar{P}_s$
2. The pressure equals to the atmospheric pressure at the bearing edges i.e.at $\bar{Z} = 0$ and $\bar{Z} = 1 \rightarrow \bar{P} = \bar{P}_{atm} = 0.0$
3. Reynolds boundary conditions is adopted i.e.at the cavitation zone $\frac{\partial \bar{P}}{\partial \theta} = 0.0$ and $\bar{P} = 0.0$.

Bearing Parameters

The components of the load carrying capacity along and perpendicular to the line of centers are calculated as the following non-dimensional parameters:

$$\bar{W}_r = \frac{c^2 \bar{W}_r}{\mu_o U R_{bi}^2 L} = \int_0^1 \int_0^{2\pi} \bar{P} \cdot \cos(\theta) d\theta d\bar{Z} \quad (5)$$

$$\bar{W}_t = \frac{c^2 \bar{W}_t}{\mu_o U R_{bi}^2 L} = \int_0^1 \int_0^{2\pi} \bar{P} \cdot \sin(\theta) d\theta d\bar{Z} \quad (6)$$

The total load carried by the bearing can be evaluated as:

$$\bar{W} = \frac{c^2 \bar{W}}{\mu_o U R_{bi}^2 L} = \sqrt{\bar{W}_r^2 + \bar{W}_t^2} \quad (7)$$

While the attitude angle can be evaluated as:

$$\phi = \tan^{-1} \left(-\frac{\bar{W}_t}{\bar{W}_r} \right) \quad (8)$$

The friction force at the journal and the bearing surfaces of the worn journal bearing in non-dimensional can be expressed as **Hashimoto et al.(1986)**.

$$\bar{f}_r = \frac{fr}{\mu_o UL \left(\frac{R_{bi}}{c} \right)} = \int_0^1 \int_0^{2\pi} \left(\frac{\mu_{ef}}{h} + \frac{\bar{h}}{2} \frac{\partial \bar{P}}{\partial \theta} \right) d\theta d\bar{Z} \quad (9)$$

The velocity component of the oil film along the axial direction is the key factor in the oil flow leakage. The dimensionless leakage flow can be determined using the following relation: **Hashimoto et al.(1986)**.

$$\bar{Q}_s = \frac{Q_s L}{c \omega R_s^2} = \int_0^{2\pi} \int_0^1 \bar{h} \bar{w} \Big|_{z=0,1} d\bar{y} d\theta \quad (10)$$

Numerical Procedure:

Isothermal analysis of a worn journal bearing system required the simultaneous solution of governing equations with appropriate boundary conditions. These equations are numerically solved using the finite difference approach. Iterative procedure with successive under relaxation has been used to obtain the pressure distributions in the oil film. A suitable computer program has been written in FORTRAN90 to solve the governing equations of the problem of the present study.

DISCUSSION OF THE OBTAINED RESULTS

The static performance characteristics of worn journal bearing, with the operating characteristics summarized in table (1), are discussed here in terms of load carrying capacity, end leakage, attitude angle, power losses and friction force. Worn bearing with different wear depth parameters and eccentricity ratios lubricated with lubricants containing different types of nanoparticles additives have been discussed.

Table 1: Data used for numerical simulation Wu et al.(2007):

Length of the Bearing	$L = 0.08 \text{ m}$
Bearing clearance	$c = 0.000152 \text{ m}$
Radius of the Shaft	$R_s = 0.05 \text{ m}$
Outer radius of the bearing	$R_{bo} = 0.1 \text{ m}$
Rotation speed	$N = 2000 \text{ rpm}$
Lubricant density	$\rho = 8600 \text{ kg/m}^3$
Engine oil (Base oil)	0.1036 Pa s
oil with nano CuO	0.1108 Pa s
oil with nanoTiO ₂	0.1425 Pa s
oil with nano diamond	0.1052 Pa s

To validate the mathematical model used in this work the worn bearing pressure distribution obtained in the present study was compared to that obtained by Hashimoto et al.(1986) as presented in figure (2). The results seem to match good with maximum deviation of 4%. It represents a good validation to the solution procedure adopted to solve the governing equations of the present work. The effect of wear depth parameter on the dimensionless pressure distribution for non-worn and worn journal bearings with different wear depth parameters works at eccentricity ratio of 0.8 lubricated with pure oil is shown in figure (3). It is clearly shown that the higher the wear depth parameter of the bearing the lower is the load carried. Figure (4) shows the pressure distribution in when the bearing works at an eccentricity ratio of 0.8 lubricated with oil containing nanoparticles of copper oxide, Diamond and titanium dioxide. This figure clearly shows that the oil film pressure increases when the bearing lubricated with oil containing nanoparticles. It is also clear that the pressure becomes higher when the bearing lubricated with oil dispersed with TiO₂ nanoparticles. This is due to higher viscosity of the lubricant in this case. The variation of dimensionless load capacity with the eccentricity ratio for non-worn and worn journal bearing lubricated with the pure oil is shown in figure (5).

This figure shows that the higher the value of the dimensionless wear parameter the lower is the load supported by the bearing. This is due to the lower of the oil film pressure generated in the clearance gap in this case. The decrease percentage in dimensionless load carrying capacity for a defected journal bearing that has different wear depth parameters ($\delta=0.2$ and 0.4) works at a wide range of eccentricity ratios has been calculated and reported in figure(6). The effect of adding different types of nano-particles to the oil on the worn bearing load carrying capacity is presented in Figure (7). This figure shows that the load carrying capacity increases with the increase in eccentricity ratio. Also it can be seen from this figure that the load carrying capacity of the bearing is greatly affected by the addition of the nanoparticles to the base oil. The value of load carrying capacity obtained for the bearing lubricated with oil containing Titanium dioxide is approximately 40% higher than that obtained when it was lubricated with pure oil. The percentages of increase in bearing load carrying capacity have been reported in figure (8).

The dimensionless friction force variation with the eccentricity ratio for different worn bearing lubricated with the pure oil is presented in figure (9). It is clearly depicted that the friction force decreases for the worn bearing with higher wear depth parameter. The percentage decrease can be shown in figure (10). This can be attributed to the decrease in oil shear rate due to the increase in oil film thickness. The effect of lubricating the worn bearing with oil containing different types of nanoparticles can be shown in figure (11). This figure clearly shows that the induced friction force in the oil film increases with percentages shown in figure (12) when the bearing lubricated with Nano-lubricant. Figure (13) shows that the coefficient of friction increases with the percentages shown in figure (14) for the worn journal bearing with higher wear depth parameter. This can be explained due to the decrease in load carrying capacity in this case. Figures (15) and (16) show that the type of nanoparticles dispersed in the base oil has a little effect on friction coefficient and attitude angle of the bearing since the load of the bearing and the friction force induced varies with different amounts in this case. The variation of the attitude angle with eccentricity ratio for different worn journal bearings lubricated with base oil can be noticed from figure (17). This figure depicts that lower attitude angle can be obtained for the worn bearing with higher values of bearing wear depth parameter. This is attributed to the decrease of the load components due to the higher wear depth parameter. The percentage decrease in attitude angle due to wear effect has been reported in figure (18). The worn bearing side leakage seems to increase with the bearing eccentricity ratio for different worn journal bearings as illustrated in figure (19). The side leakage increases with percentages shown in figure (20) for the worn bearing with higher value of (δ) due to the increase in oil film thickness. Figure (21) shows the effect of using CuO Diamond and titanium dioxide on the oil leakage of the bearing when it is presented against the bearing eccentricity ratios. It can be seen from this figure that the side leakage decreases with percentages shown in figure (22) in this case. It is clear that the side leakage has the lower values when the bearing lubricated with oil containing TiO_2 nanoparticles. This is due to the increase in oil viscosity with the addition of nano particles. Figure (23) shows that the power loss of the worn bearing decreases with percentages shown in figure (24) for the bearings with higher values of (δ). This is due to the decrease in friction force induced in this case. Figure (25) gives the relation between power loss and eccentricity ratio for various lubricants (Diamond, Copper oxide and Titanium dioxide). This figure shows that the power loss increases with percentages shown in figure (26) when the bearing lubricated with Nano-lubricant.

CONCLUSIONS

From the previous discussion the following conclusions can be drawn .

1. The effect of wear is to decrease the load carrying capacity, attitude angle and power losses, and to increase the friction coefficient and flow rate of journal bearings.
2. For the lubricant with TiO_2 Nano-particles the load carrying capacity, friction force and power losses for the worn bearing works at any eccentricity ratio increased by approximately 40% .
3. The side leakage decreases by about 25% for the worn bearing lubricated with oil containing Nano-lubricant when it was compared with that lubricated with pure oil.
4. The attitude angle slightly affected when the bearing lubricated with oil containing different types of Nano-particles.

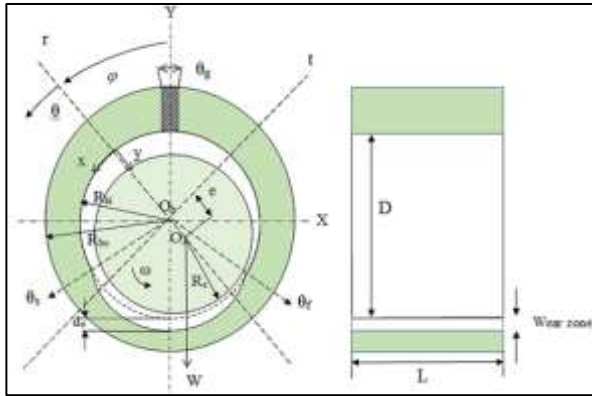


Fig. (1) Geometrical Configuration of worn Journal Bearing

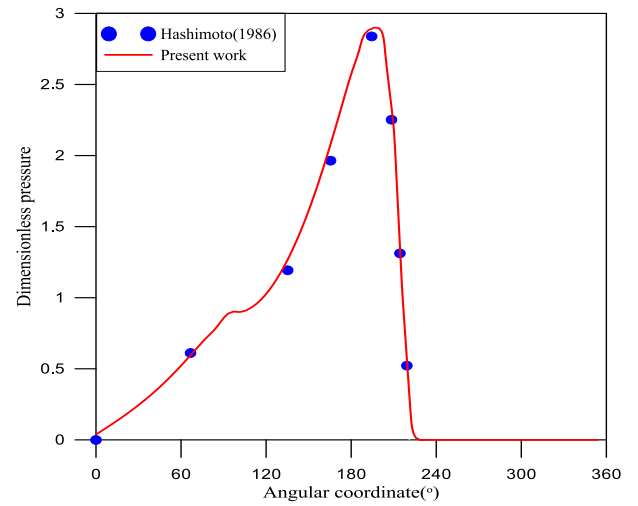


Figure (2): Validation for the pressure distributions Obtained in the present work

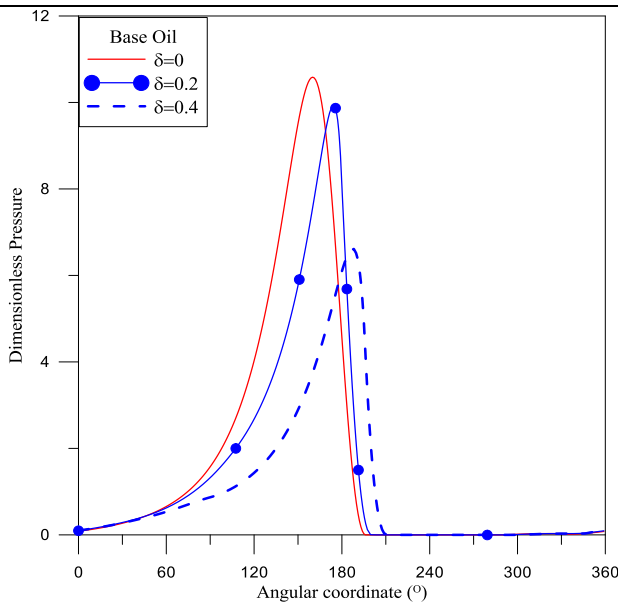


Fig (3): Dimensionless pressure distribution for a worn journal bearing with pure oil t

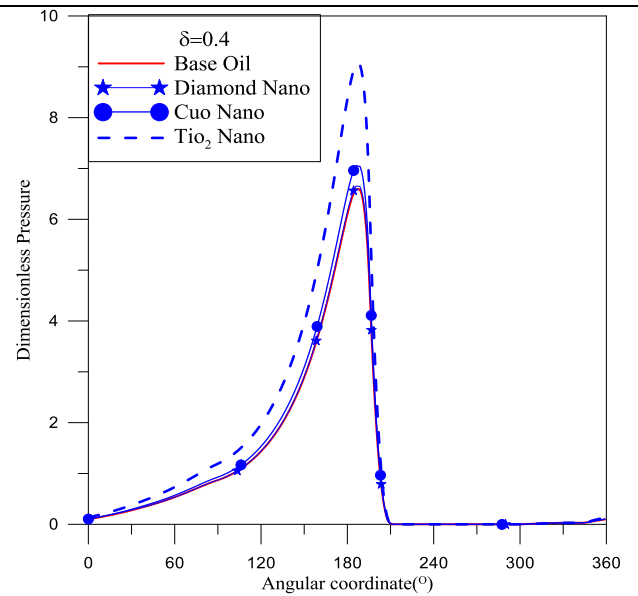


Fig (4): Dimensionless pressure distribution for a worn journal bearing lubricated with and different Nano-lubricant

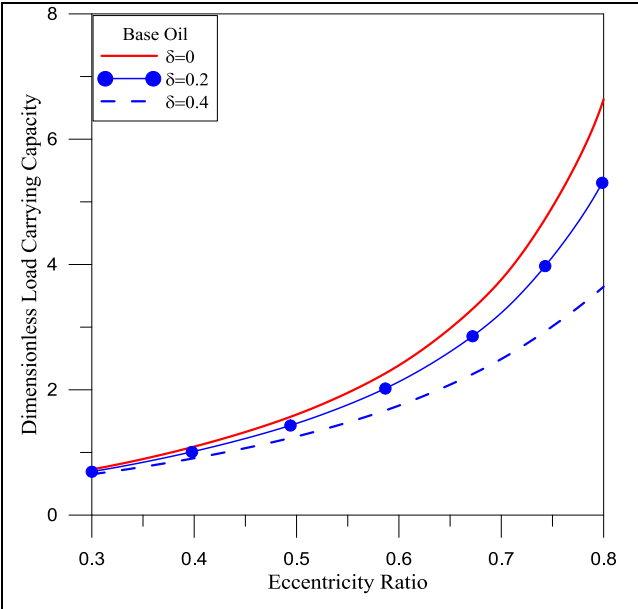


Fig (5): Load carrying capacity verses eccentricity ratio for a worn journal bearing lubricated with base oil lubricant.

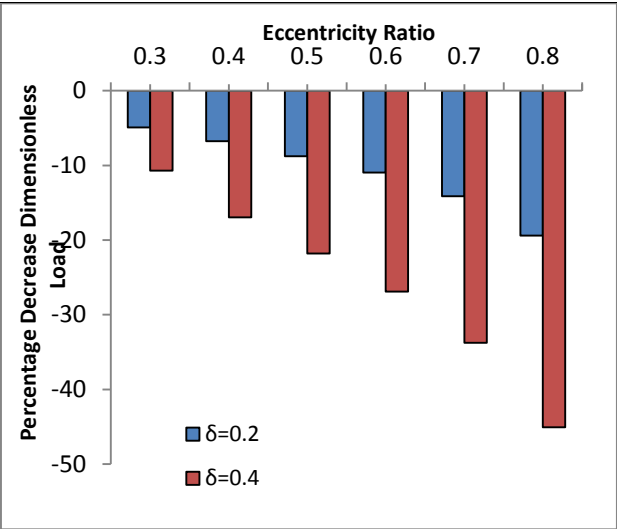


Fig (6): Percentage decrease in load carrying capacity.

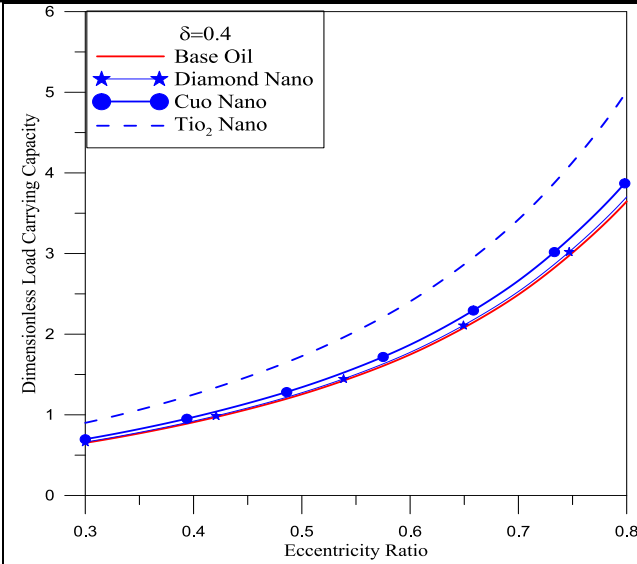


Fig (7): Dimensionless Load versus eccentricity ratio for a worn journal bearing ($\delta=0.4$) lubricated with base oil lubricant and different nanolubricants

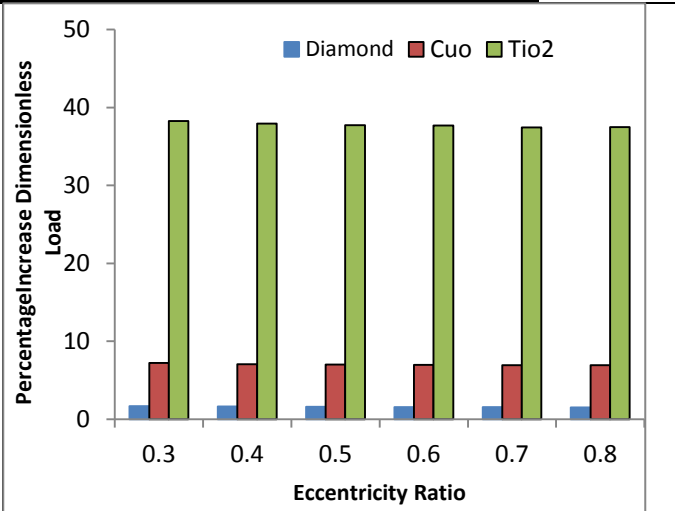


Fig (8): Percentage increase in load carrying capacity for a worn journal bearing ($\delta=0.4$) lubricated base oil and different nanolubricants

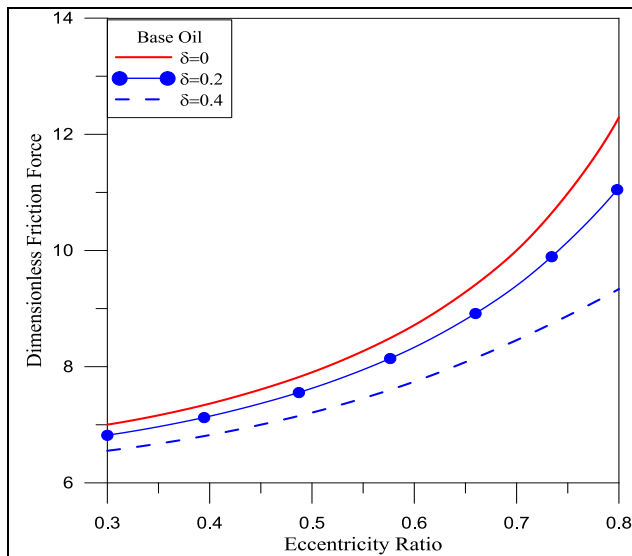


Fig (9): Friction force versus eccentricity ratio for a worn journal bearing lubricated with base oil lubricant.

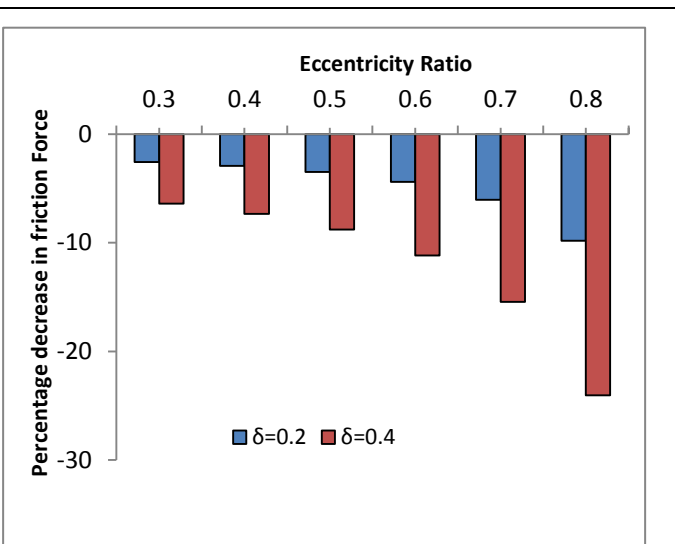
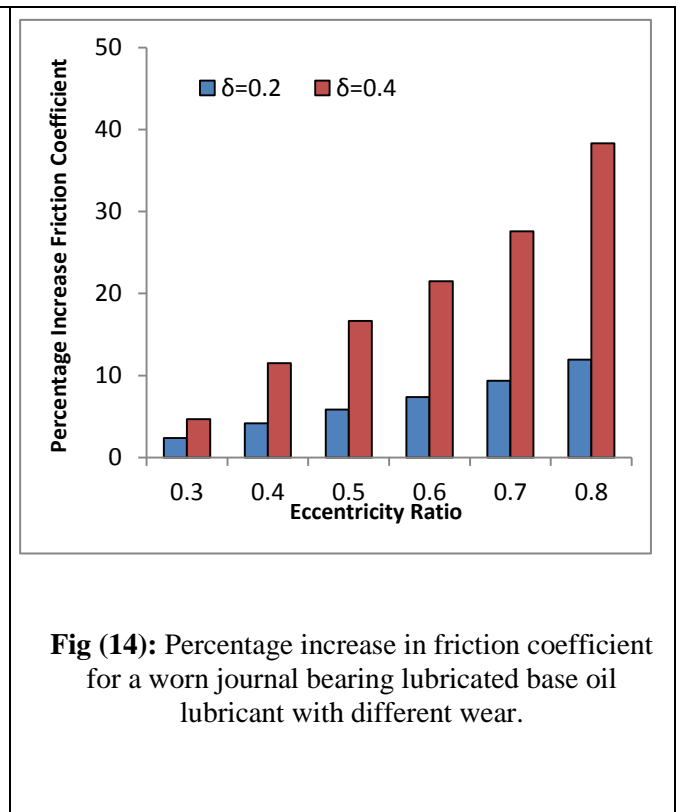
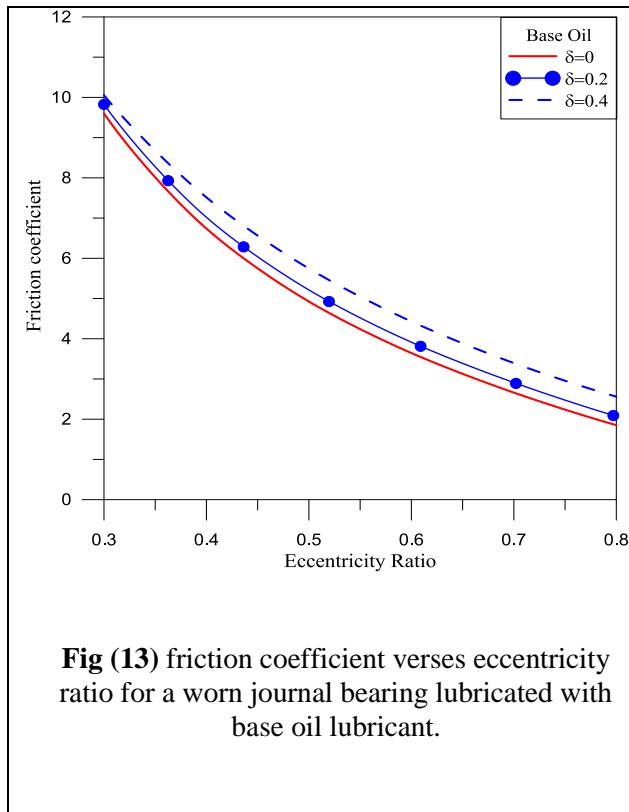
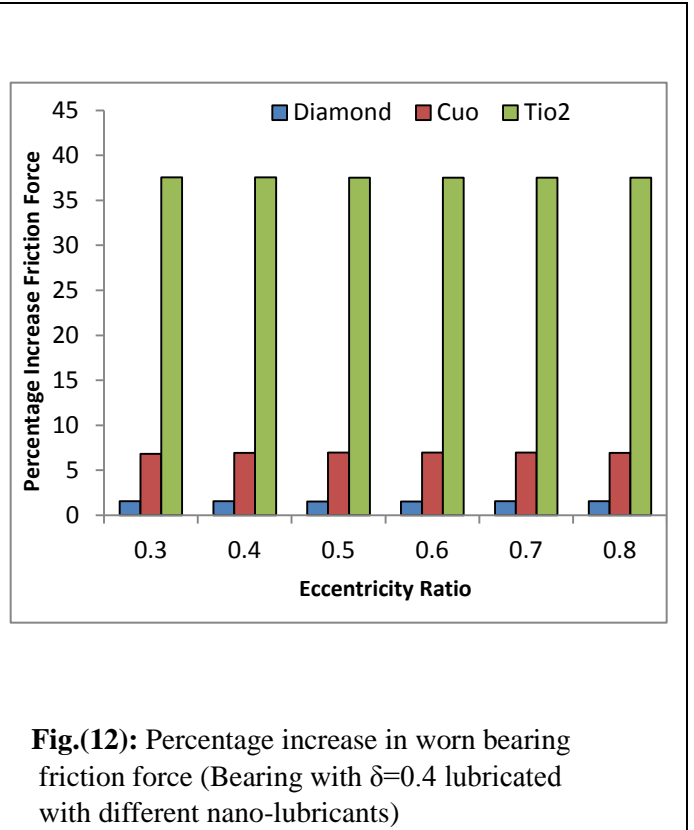
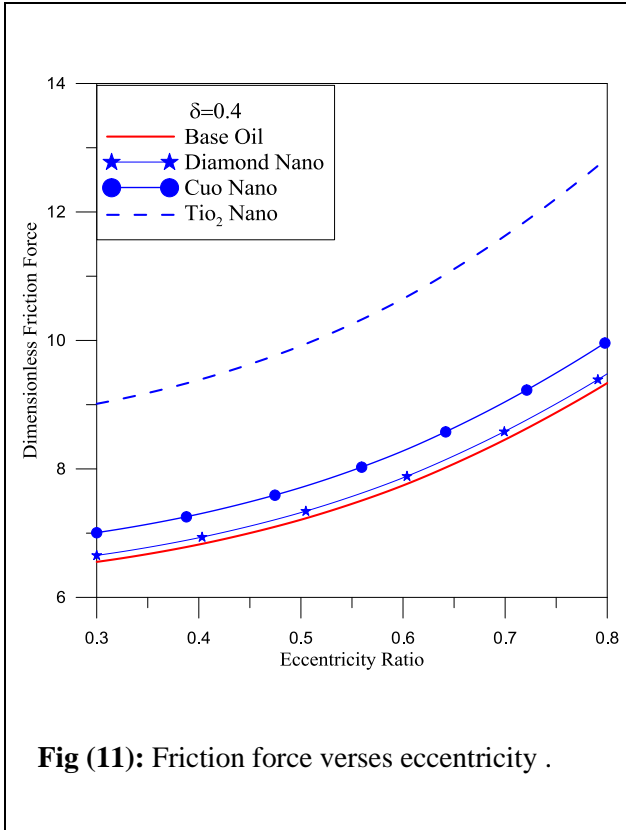


Fig (10): Percentage decrease in friction force .



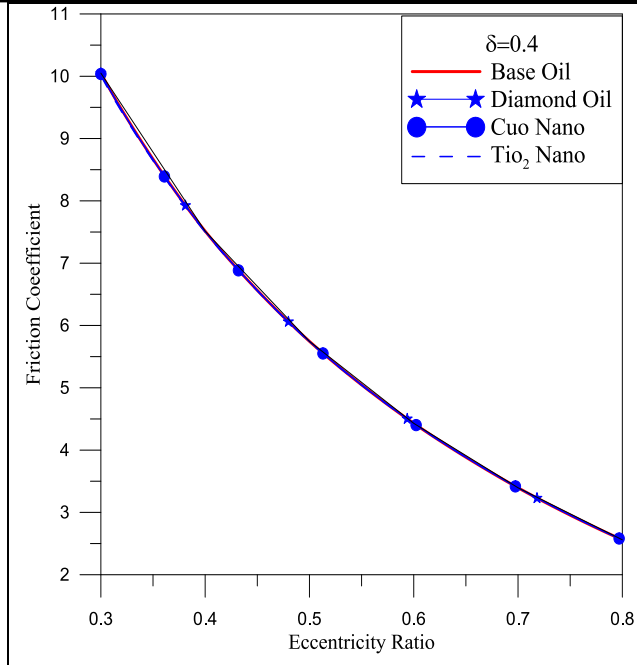


Fig (15): friction coefficient verses eccentricity ratio for a worn journal bearing ($\delta=0.4$) lubricated with base oil lubricant and different nanolubricants

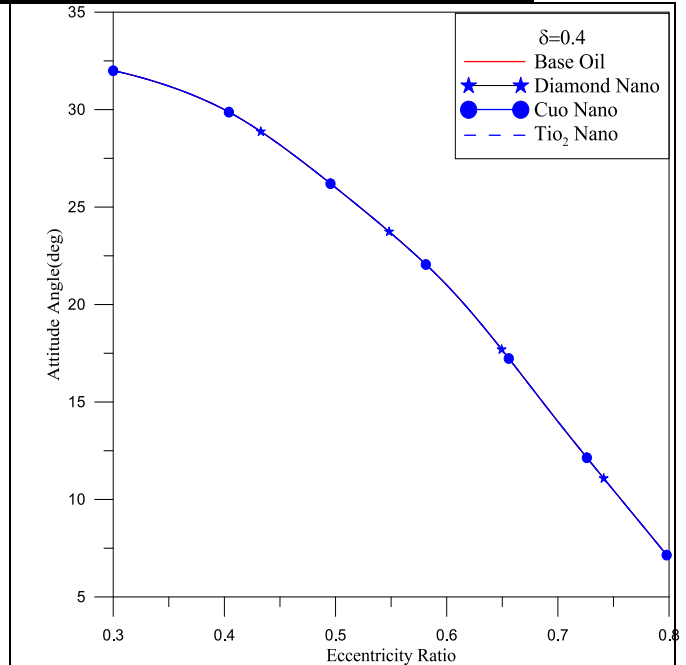


Fig (16): Attitude angle verses eccentricity ratio for a worn journal bearing ($\delta=0.4$) lubricated with base oil lubricant and different nanolubricants

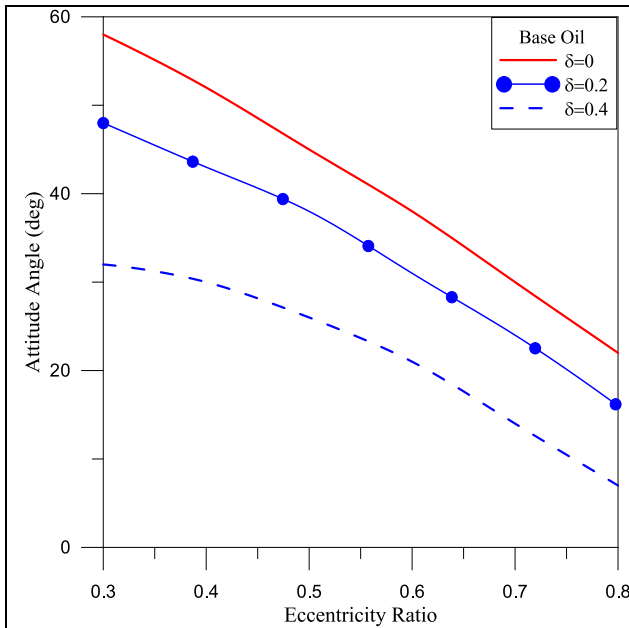


Fig (17) Attitude angle verses eccentricity ratio for a worn journal bearing lubricated with base oil lubricant.

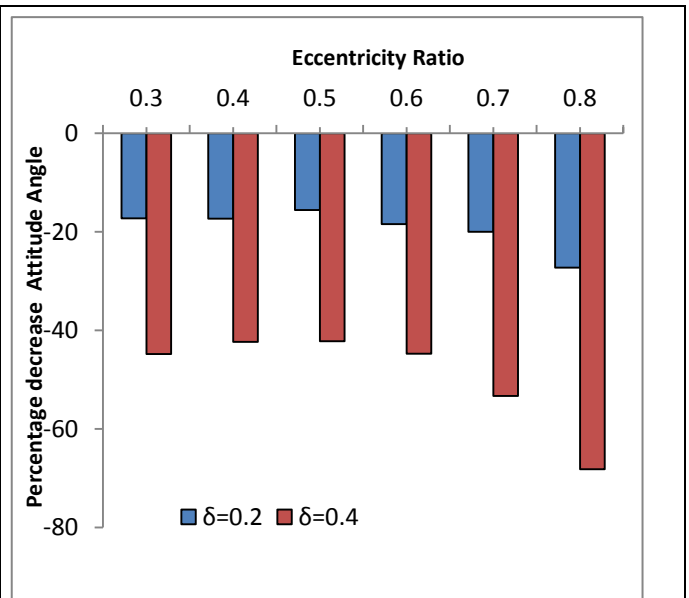


Fig (18): Percentage decrease in attitude angle for a worn journal bearing lubricated base oil lubricant with different wear.

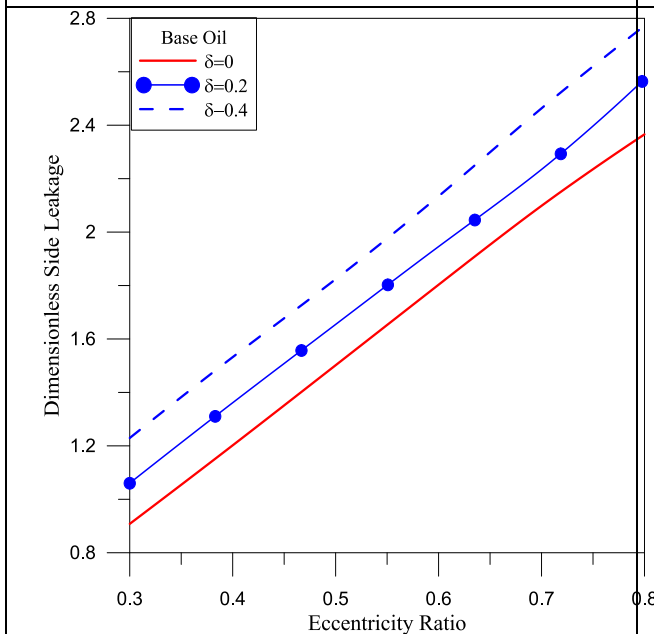


Fig (19) Dimensionless side leakage verses eccentricity ratio for a worn journal bearing lubricated with base oil lubricant.

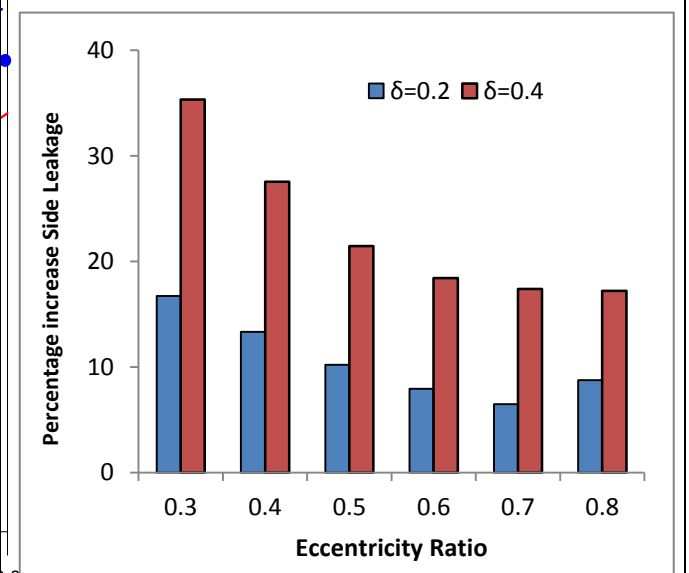


Fig (20): Percentage increase in dimensionless side leakage for a worn journal bearing lubricated base oil lubricant with different wear

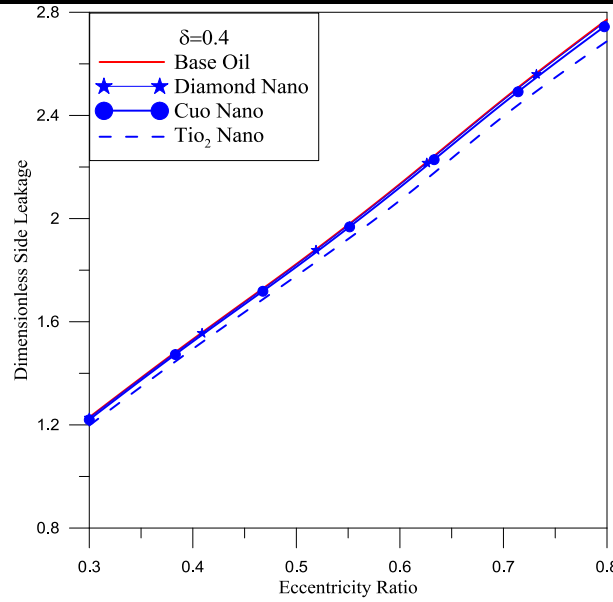


Fig (21): Dimensionless side versus eccentricity ratio for a worn journal bearing ($\delta=0.4$) lubricated with base oil lubricant and different nanolubricants.

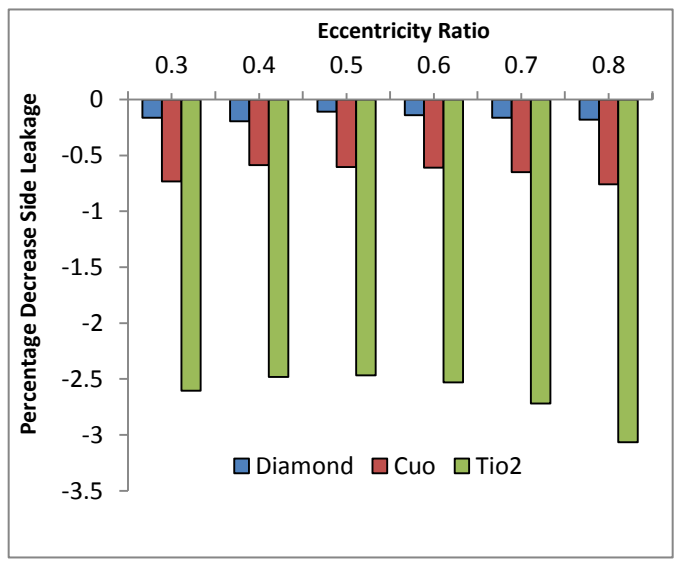


Fig (22): Percentage decrease in dimensionless side leakage for a worn journal bearing ($\delta=0.4$) lubricated base oil and different nanolubricants

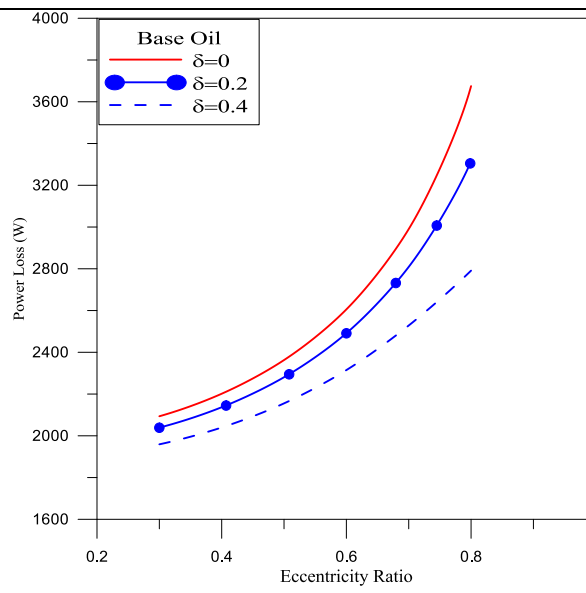


Fig (23) Power loss versus eccentricity ratio for a worn journal bearing lubricated with base oil lubricant.

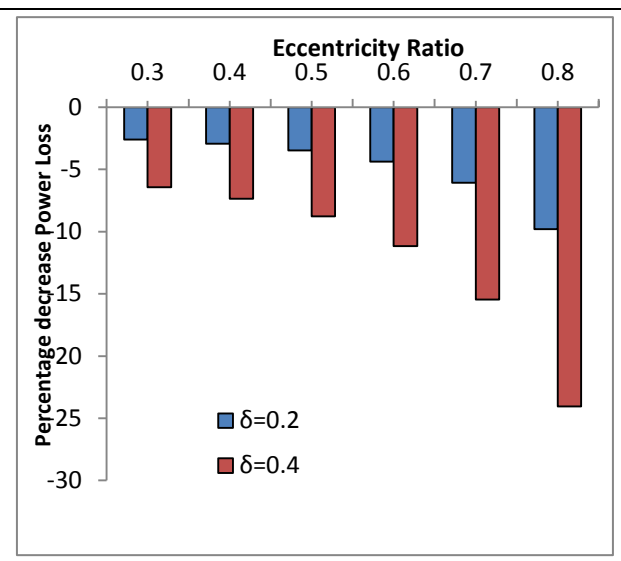


Fig (24): Percentage decrease in dimensionless power loss for a worn journal bearing lubricated base oil lubricant with different wear

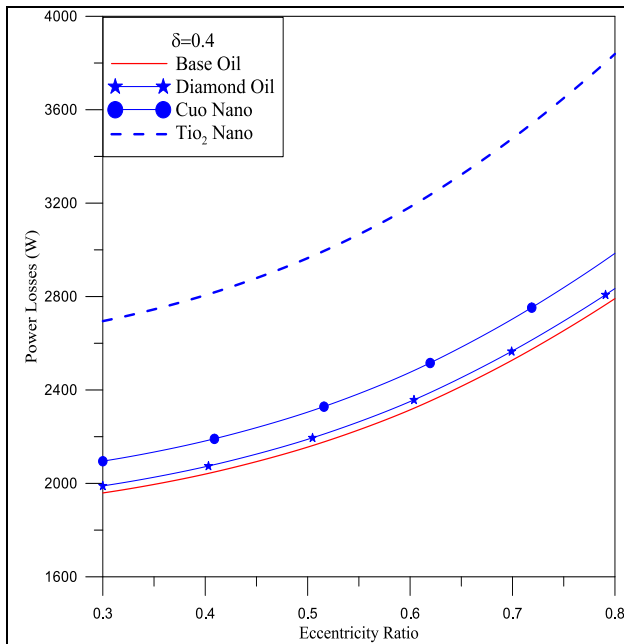


Fig (25): Power loss versus eccentricity ratio for a worn journal bearing ($\delta=0.4$) lubricated with base oil lubricant and different nanolubricants.

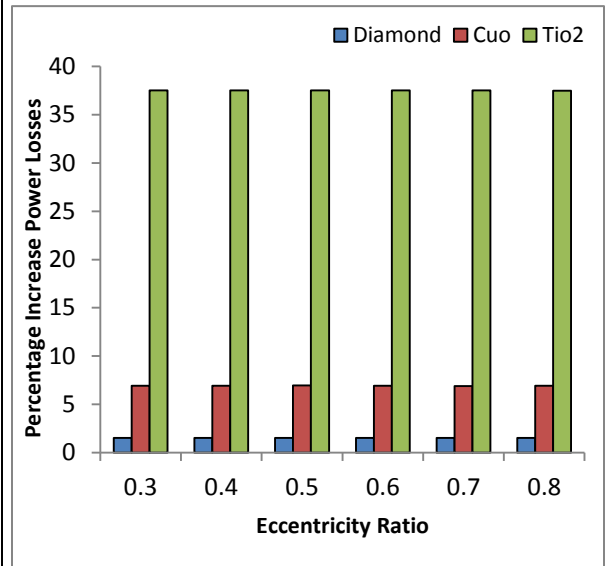


Fig.(26): Percentage increase in power loss for worn journal bearing .

Nomenclature

d	Wear depth
C	Radial clearance (m)
\bar{f}_r	Dimensionless friction force
\bar{h}	Non-dimensional oil film thickness
L	Bearing length (m)
N	Number of revolution per minute (rpm)
\bar{P}	Non-dimensional oil film pressure
\bar{Q}_s	Non-dimensional Side leakage flow
R_s	Journal Radius (m)
R_{bi}	Bush inner radius (m)
R_{bo}	Bush outer radius (m)
\bar{W}	Dimensionless load carrying capacity of Worn journal bearing
\bar{W}_r	Dimensionless radial load component
\bar{W}_t	Dimensionless tangential load component
x, y, z	Coordinate system (m)

Greek symbols

μ	Pure lubricant viscosity (Pa.sec.)
$\bar{\mu}$	Dimensionless pure lubricant viscosity = $\frac{\mu}{\mu_o}$
ε	Eccentricity ratio
ρ	Pure lubricant density (kg / m ³)
ϕ	Attitude angle, Deg.
θ	Angular coordinate, Deg.
ω	Journal rotational speed, rad / sec.
δ	Dimensionless wear depth

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