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EFFECT OF LUBRICANT TEMPERATURE ON THE DYNAMIC AND STABILITY BEHAVIOR OF NANO-LUBRICATED JOURNAL BEARING

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ABSTRACT

The effect of lubricant temperature on the dynamic behaviour of Nano-lubricated finite length journal bearings has been investigated in the present work. Modified time dependent Reynolds equation include the effect of oil film temperature was perturbed in order to calculate the eight dynamic coefficients (four stiffness, and four damping) required to evaluate the dynamic characteristics of the journal bearing, the oil film temperature was obtained by solving numerically the energy and heat conduction equations simultaneously with the Reynolds equation using appropriate boundary conditions. Suitable viscosity temperature model has been used to consider the effect of oil film temperature. The bearing lubricated with oil containing Titanium dioxide (TiO_2) nanoparticles dispersed in to the base oil with various particle concentration. The effect of adding TiO_2 Nano-particles with various particle concentrations (0.1%, 0.5%, 1%, 1.5% and 2%) in order to enhance its viscosity has been discussed. A validation to the mathematical model and the computer program written in Fortran90 prepared to solving the governing equation has been carried out by comparing the result for the dynamic coefficients (stiffness and damping) obtained in the present work with that obtained by Sheeja and Prabhu . The results seen to be in a good agreement with percentage of error less than 2%. The results obtained in the present study show that the bearing stiffness coefficients increases by 30% while the damping coefficients increased by 27% when the bearing lubricated with Nano lubricant that contains TiO_2 Nano particles with particle concentration of (1.5%).
KEYWORDS: Journal Bearing, Thermo-hydrodynamic, Reynolds equation, dynamic coefficients, perturbation technique, stability.

دراسة تأثير درجة حرارة الزيت على التصرف الديناميكي والاستقرارية للمسند المقعدي المزيت بزيت حاوٍ على دقائق متناهية في الصغر

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الخلاصة

تم دراسة تأثير درجة حرارة الزيت على السلوك الديناميكي للمسند المقعدي المزيت بزيت حاوٍ على دقائق متناهية في الصغر وذو طول محدد. معادلة رينولدز المعتمدة على الزمن تشمل تأثير درجة حرارة طبقة الزيت من اجل حساب المعاملات الديناميكية الثمانية المطلوبة لتقييم الخصائص الديناميكية للمسند المقعدي، تم الحصول على درجة حرارة طبقة الزيت من خلال حل معادلات الطاقة والتوصيل الحراري عدديا مع معادلة رينولدز بأستخدام شروط حدية مناسبة. وقد تم استخدم نموذج مناسب درجة الحرارة–اللزوجة للنظر في تأثير درجة حرارة طبقة الزيت. تم تزييت المسند

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بزيت يحتوي على دقائق متناهية الصغر من ثنائي اوكسيد التيتانيوم المتناثرة في الزيت الاساس مع تركيزات الجزيئات المختلفة. وقد تمت مناقشة تأثير اضافة جزيئات نانوية (TiO_2) مع تراكيز جزيئات مختلفة (٠,١% الى ٢%) من اجل تعزيز اللزوجة. تم إجراء التحقق من صحة النموذج الرياضي وبرنامج الكمبيوتر الذي تم إعداده لحل المعادلات الحاكمة من خلال مقارنة نتائج المعاملات الديناميكية (الصلابة والتخميد) التي تم الحصول عليها في العمل الحالي مع تلك المستحصلة من الباحثان Sheeja و Prabhu. يبدو أن النتائج في اتفاق جيد مع نسبة خطأ أقل من ٢٪. تظهر النتائج التي تم الحصول عليها في الدراسة الحالية أن معاملات الصلابة للمسند تزداد بنسبة ٣٠٪ بينما زادت معاملات التخميد بنسبة ٢٧٪ عندما يكون المحمل مزيتاً بزيت نانوي الذي يحتوي على جزيئات TiO_2 النانوية بتركيز جزيئي قدره (١.٥٪) .

INTRODUCTION

A relative rotational or linear movement between two parts can be permitted by using journal bearing. It consists of two surfaces in relative motion with a thin lubricant film between them to allow hydrodynamic lubrication process. The static and dynamic behavior of such bearings have been investigated by many workers. **Babuh et al. (2014)** presented a mathematical model to represent the viscosity temperature relation for the multi grade engines lubricant (SAE15 W40) with dispersed aluminum and zinc oxide (Al_2O_3), (ZnO), Nanoparticles. This mathematical model was used for the calculation of statics performances characteristics of the bearing. The distribution of pressure and temperatures are obtained by solving modified Reynold's and energy equations by using finite element method (FEM). Thermal effect on rotor dynamic of continuous rotor shaft has been studied analytically by **Gu and Chu (2014).** An insight into the mechanisms for the rotor thermal vibration has been presented. The convection coefficients and the heat conductivity influences on the thermal vibrations have been considered in order to provide an insight into the management of thermal vibrations from the perspective of thermal aspects. **Kuznetsov and Glavatskih (2016)** used thermo-hydrodynamic model with mechanical and thermal deformations of the bearing surface to investigate there effects on the dynamic characteristics of two axial grooves. It has been found that thermal deformation increases the horizontal stiffness coefficients (Kyy) and (Kxy) and slightly reduces journal critical mass. **Lokhande and Prabhu (1987)** suggested a method to include the variable viscosity of the oil in journal bearing by solving suitable energy equation which is uncoupled with the Reynold's equation. The static and the dynamic performances of the partial journal-bearing were considered. **Michaud et al. (2007)** developed three dimensional transient thermo-hydrodynamic model to study the behavior of dynamically loaded journal bearing using finite element technique. Jacobson- Floberg-Olson model was used to predict the cavitation boundary. Bearings under sinusoidal loading have been studied using the proposed model. **Majumdar (1992)** considers the thermal effect to investigate the stability of submerged oil journal bearings. Jakobson–Floberg- Olson model was used to investigate the cavitation region of the bearing. A peso-viscous oil model was taken into consideration by using the exponential law to describe the variation of the oil viscosity with the temperature. **Paranjpe (1996)** using a transient thermo-hydrodynamic to study the dynamically loaded engine bearings. Oil film temperatures are found to vary considerably over time and space. The results obtained show that the adiabatic and simplified thermo-hydrodynamic analysis was well compared with the full thermo-hydrodynamic analysis. Significant improvement in the results has been obtained over the isothermal analysis. **Solghar (2015)** investigates the thermo-hydrodynamic behavior of single grooves journal bearing operating under steady loading. The performance of the journal bearing is obtained when it is lubricated with pure oil and that mixed with Al2O3 Nanoparticles. Finite volume scheme was used to solve the governing equations including momentum, continuity, and energy equations within the lubricants as well as within the solid bush. In the present work the combined effect of Nano-lubrication and oil film temperature on the dynamic and stability characteristics of the journal bearing is investigated.

GOVERNING EQUATIONS

In this work a finite length journal bearing that has geometric and coordinates system shown in fig. (1) is considered. It consists of journal rotating around its center O_j with angular speed ω and fixed bearing with center O_b. Single axial groove of 18^O width used to supply the lubricant inside the clearance gap of the bearing. The governing equations for the dynamic behavior of Nano lubricated journal bearing considering thermo-hydrodynamic analysis can be summarized as follow.

Reynold's Equastion

The following time dependent Reynold's equation for Newtonian, laminar, peso viscous flow modified to include thermal effect is adopted Costa, et al. (2003)

$$\frac{\partial}{R\,\partial x}\left(G\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(G\frac{\partial P}{\partial z}\right)=U\frac{\partial F}{R\,\partial x}+\frac{\partial h}{\partial t} \quad (1)$$

Where:

$$G=\int_0^h\left[\frac{y}{\mu}\left(y-\frac{\int_0^h y/\mu\,.dy}{\int_0^h 1/\mu\,.dy}\right)\right]dy \quad (2)$$

$$F=1-\frac{\int_0^h y/\mu\,.dy}{h\int_0^h 1/\mu\,.dy} \quad (3)$$

The following non dimensional groups can be used to generalize equation (1) **Boubendir and Larbi (2011)**.

$$\bar{P}=\frac{P\,c^2}{\mu_o\,\omega\,R_s R_{bi}},\quad \bar{T}=\frac{T}{T_o},\quad \bar{h}=\frac{h}{c},\quad \bar{y}=\frac{y}{h},\quad \theta=\frac{x}{R_{bi}},\quad \bar{z}=\frac{z}{L},\quad \bar{r}=\frac{r}{R_{bi}},\quad \tau=\omega t \text{ and } U=R\omega$$

Substituting the groups in Equation (1) result in the following non-dimensional Reynold's equation:

$$\frac{\partial}{\partial\theta}\left(\bar{h}^3\bar{G}\frac{\partial\bar{P}}{\partial\theta}\right)+\left(\frac{D}{2L}\right)^2\frac{\partial}{\partial\bar{z}}\left(\bar{h}^3\bar{G}\frac{\partial\bar{P}}{\partial\bar{z}}\right)=\frac{1}{2}\left(\frac{\partial\bar{F}}{\partial\theta}\right)+\frac{\partial\bar{h}}{\partial\tau} \quad (4)$$

Where:

$$\bar{G}=\int_0^1\left[\frac{\bar{y}}{\bar{\mu}}\left(y-\frac{\int_0^1 \bar{y}/\bar{\mu}\,.d\bar{y}}{\int_0^1 1/\bar{\mu}\,.d\bar{y}}\right)\right].d\bar{y} \quad (5)$$

$$\bar{F}=1-\frac{\int_0^1 \bar{y}/\bar{\mu}\,.d\bar{y}}{\bar{h}\int_0^1 1/\bar{\mu}\,.d\bar{y}} \quad (6)$$

thickness of the oil film for the aligned plain journal can be expressed in non-dimensional form as **Boubendir and Larbi (2011)**:

$$\bar{h}=1+\varepsilon\cos(\theta-\psi) \quad (7)$$

Oil viscosity is the most important physical property which couples the Reynold's and energy equations. It was presupposed to be variables across the fluid film, in the axial, and in circumferential directions. The exponential oil viscosity model can be used to describe the its dependence on the oil film temperature as follow

$$\mu = \mu_\circ e^{-\beta(T-T_0)} \tag{8}$$

Krieger–Dougherty viscosity model can be used to include the effect of the Nano-particles dispersed in the base oil on its viscosity. It can be expressed as follow **Binu et al. (2014)**:

$$\bar{\mu}_{nf} = e^{-\beta T_\circ(\bar{T}-1)}\left\{1-\frac{\varphi}{0.605}\left(\frac{a_a}{a}\right)^{1.2}\right\}^{-1.5} \tag{9}$$

where:

a_a, a: radius of aggregates and primary particles respectively.

The aggregate packing fraction for the dispersion of TiO_2 in engine oil has been measured and it was found to be $\left(\frac{a_a}{a}\right) = 7.77$ **Binu et al. (2013).**

Energy Equation

The lubricant inside the clearance gap was sheared due to the shaft rotational speed and the oil viscosity causes a considerable shear stress. The energy equation governs the distribution of temperature in the oils film can be written in the dimensionless form **Ferron et al. (1983)**.

$$P_e\left[\bar{u}\frac{\partial\bar{T}}{\partial\theta}+\left(\frac{\bar{v}}{c.\bar{h}}-\bar{u}\frac{\bar{y}}{\bar{h}}\frac{\partial\bar{h}}{\partial\theta}\right)\frac{\partial\bar{T}}{\partial\bar{y}}\right]=\frac{\partial^2\bar{T}}{\bar{h}^2\,\partial\bar{y}^2}+\frac{\bar{\mu}_{nf}}{\bar{h}^2}N_d\left[\left(\frac{\partial\bar{u}}{\partial\bar{y}}\right)^2+\left(\frac{\partial\bar{w}}{\partial\bar{y}}\right)^2\right] \tag{10}$$

The non-dimensional velocity components $\bar{u}$,$\bar{v}$, and $\bar{w}$ are defined as in **Roy (2009)**

$$\bar{u}=\frac{u}{U}=\bar{h}^2\frac{\partial\bar{P}}{\partial\theta}\left\{\int_0^{\bar{y}}\frac{\bar{y}}{\bar{\mu}_{nf}}\,d\bar{y}-\frac{\int_0^1\frac{\bar{y}}{\bar{\mu}_{nf}}.d\bar{y}.\int_0^{\bar{y}}\frac{1}{\bar{\mu}_{nf}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}_{nf}}.d\bar{y}}\right\}+\frac{\int_0^{\bar{y}}\frac{1}{\bar{\mu}_{nf}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}_{nf}}.d\bar{y}} \tag{11}$$

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

$$\bar{w}=\frac{w}{U}=\bar{h}^2\frac{\partial\bar{P}}{\partial\bar{z}}\left(\frac{1}{2\gamma}\right)\left\{\int_0^{\bar{y}}\frac{\bar{y}}{\bar{\mu}_{nf}}\,d\bar{y}-\frac{\int_0^1\frac{\bar{y}}{\bar{\mu}_{nf}}.d\bar{y}.\int_0^{\bar{y}}\frac{1}{\bar{\mu}_{nf}}d\bar{y}}{\int_0^1\frac{1}{\bar{\mu}_{nf}}.d\bar{y}}\right\}$$

Heat Conduction Equation

The distribution of the temperature through the metal of the bearing (bush) was evaluated by solving the following heat conduction equation **Ferron et al. (1983).**

$$\frac{\partial^2\bar{T}_b}{\partial\bar{r}^2}+\frac{1}{\bar{r}}\frac{\partial\bar{T}_b}{\partial\bar{r}}+\frac{1}{\bar{r}^2}\frac{\partial^2\bar{T}_b}{\partial\theta} \tag{12}$$

BOUNDARY CONDITIONS

Hydrodynamic Boundary Conditions

The following boundary conditions were used together with the Reynolds equation to define the pressure distribution inside the oil gap of the aligned journal bearing

$$\frac{\partial\bar{P}}{\partial\theta}=0.0 \;\&\; \bar{P}(\theta,z)=P_c=0.0 \text{ at } \theta=\theta_c$$

$$\bar{P}(\theta,z)=\bar{P}_s \text{ at } \theta=2\pi-\psi \tag{13}$$

$$\bar{P}(\theta,\pm1)=0.0 \text{ at } \bar{z}=0.0 \;\&\; \bar{z}=1$$

Thermal Boundary Conditions

The following boundary conditions were used with the energy equation to define the temperature distribution through the oil film.
The oil mixing temperature can be calculated as **Roy (2009)**

:$$T_{mix} = \frac{Q_r \bar{T}_r + Q_{in} \bar{T}_{in}}{Q_r + Q_{in}} \quad (14)$$

The recirculation flow rate (Q_r) can be calculated as:

$$Q_r = \int_0^h u.\,dy \quad (15)$$

which can be rewritten as follows:

$$Q_r = LUc \int_0^1 \bar{u}\,\bar{h}.\,d\bar{y} \quad (16)$$

The free convection from the outside surface of the bearing can be expressed as **Ferron, (1983).**

$$\left.\frac{\partial \bar{T}_b}{\partial \bar{r}}\right|_{\bar{R}_b = \frac{R_{bo}}{R_{bi}}} = -\frac{h_{conv}}{K_b} R_{bi} (\bar{T}_b - \bar{T}_a) \quad (17)$$

continuity of heat flux at the interface of the oil film and the bearing surface can be expressed as **Ferron, (1983)** .

$$\left.\frac{\partial \bar{T}_b}{\partial \bar{R}_b}\right|_{\bar{r}=1} = -\frac{K_{nf}}{K_b} \frac{R_{bi}}{c} \frac{1}{\bar{h}} \left.\frac{\partial \bar{T}}{\partial \bar{y}}\right|_{\bar{y}=0} \quad (18)$$

where

$$\bar{r} = \frac{r}{R_{bi}}$$

Ts = oil supply temperature =43^{o}C

DYNAMIC COEFFICIENTS

Figure (2) shows the main dynamic coefficients required to analyze the dynamic behavior of the bearing. They can be evaluated by calculating the bearing load components and finding the bearing parameters at the equilibrium position of the journal center. For this purpose the modified Reynolds equation (4) has been perturbed using suitable perturbation technique. The approach proposed by **Weimin et al.(2012), Roy and Kakoty (2013),** and finally followed by **Abass and Munier (2017)** was adopted to perturb such equation. The eccentricity ratio and the attitude angle can be perturbed as follows:

$$\varepsilon = \varepsilon_\circ + E_\circ e^{i\Omega t} \quad (19)$$

$$\psi = \psi_\circ + \Psi_\circ e^{i\Omega t} \quad (20)$$

Substitute equations (19) and (20) into equation (7) to get the perturbed oil film thickness as follow:

$$\bar{h} = \bar{h}_\circ + \bar{h}_1 e^{i\Omega\tau} = \bar{h}_\circ + (\varepsilon_\circ \cos\theta + \varepsilon_\circ \Psi_\circ \sin\theta) e^{i\Omega\tau} \quad (21)$$

The hydrodynamic pressure can be perturbed as **Weimin et al. (2012)**:

$$\bar{P} = \bar{P}_\circ + \bar{Q}_{10} e^{i\Omega\tau} + \bar{Q}_{20} e^{2i\Omega\tau} + \cdots \quad (22)$$

Substitute equations (21) and (22) into equation (1) and collecting the terms to get the following zero, first and second types of pressure ($\overline{P}_o, \overline{P}_1$ and $\overline{P}_2$) which can be expressed as follows:

$$\frac{\partial}{\partial\theta}\left(\bar{h}_\circ^3 \bar{G}\frac{\partial \bar{P}_\circ}{\partial\theta}\right)+\left(\frac{D}{2L}\right)^2\frac{\partial}{\partial\bar{z}}\left(\bar{h}_\circ^3\bar{G}\frac{\partial\bar{P}_\circ}{\partial\bar{z}}\right)=\left(\frac{\partial\bar{F}\bar{h}_\circ}{\partial\theta}\right) \tag{23}$$

$$\frac{\partial}{\partial\theta}\left(\bar{h}_\circ^3 \bar{G}\frac{\partial \bar{P}_1}{\partial\theta}\right)+\left(\frac{D}{2L}\right)^2\frac{\partial}{\partial\bar{z}}\left(\bar{h}_\circ^3\bar{G}\frac{\partial\bar{P}_1}{\partial\bar{z}}\right)+\frac{\partial}{\partial\theta}\left(3\bar{h}_\circ^2\cos\theta\,\bar{G}\frac{\partial\bar{P}_\circ}{\partial\theta}\right)+\left(\frac{D}{2L}\right)^2\frac{\partial}{\partial\bar{z}}\left(3\bar{h}_\circ^3\cos\theta\,\bar{G}\frac{\partial\bar{P}_\circ}{\partial\bar{z}}\right)=\left(\frac{\partial\bar{F}\cos\theta}{\partial\theta}\right)+i\Omega\cos\theta \tag{24}$$

$$\frac{\partial}{\partial\theta}\left(\bar{h}_\circ^3 \bar{G}\frac{\partial \bar{P}_2}{\partial\theta}\right)+\left(\frac{D}{2L}\right)^2\frac{\partial}{\partial\bar{z}}\left(\bar{h}_\circ^3\bar{G}\frac{\partial\bar{P}_2}{\partial\bar{z}}\right)+\frac{\partial}{\partial\theta}\left(3\bar{h}_\circ^2\sin\theta\,\bar{G}\frac{\partial\bar{P}_\circ}{\partial\theta}\right)+\left(\frac{D}{2L}\right)^2\frac{\partial}{\partial\bar{z}}\left(3\bar{h}_\circ^3\sin\theta\,\bar{G}\frac{\partial\bar{P}_\circ}{\partial\bar{z}}\right)=\left(\frac{\partial\bar{F}\sin\theta}{\partial\theta}\right)+i\Omega\sin\theta \tag{25}$$

It is clear that equation (23) is just the Reynolds equation for steady state operation of the bearing. It can be used to evaluate the bearing parameters at the equilibrium position of the journal center. The same boundary conditions (13) can be used through the solution. The finite difference technique (iterative procedure with successive over relaxation) has been used to solve equation (23). The load components can be calculated as:

$$\bar{F}_{x\circ}=-\int_0^1\int_0^{2\pi}\bar{P}_\circ.\cos\theta\ d\theta.d\bar{z} \tag{26}$$

$$\bar{F}_{y\circ}=-\int_0^1\int_0^{2\pi}\bar{P}_\circ.\sin\theta\ d\theta.d\bar{z} \tag{27}$$

The calculation of the equilibrium position is the key of calculating the dynamics coefficients. When the journal center forced to move out of its equilibrium position two additional dynamic pressures P_1 and P_2 are introduced. The linear stiffness coefficient can be found by integrating the linear pressure over the bearing surface as below, **Zhi (1995)**

$$\bar{F}_{x1}=\int_0^1\int_0^{2\pi}\bar{P}_1\cos\theta\,d\theta d\bar{z} \qquad \bar{F}_{y1}=\int_0^1\int_0^{2\pi}\bar{P}_1\sin\theta\,d\theta d\bar{z} \tag{28}$$

$$\bar{F}_{x2}=\int_0^1\int_0^{2\pi}\bar{P}_2\cos\theta\,d\theta d\bar{z} \qquad \bar{F}_{y2}=\int_0^1\int_0^{2\pi}\bar{P}_2\sin\theta\,d\theta d\bar{z} \tag{29}$$

the dynamic coefficients can be expressed as:

$$K_{xx}=\frac{\partial F_x}{\partial x}\quad K_{xy}=\frac{\partial F_x}{\partial y}\quad K_{yx}=\frac{\partial F_y}{\partial x}\quad K_{yy}=\frac{\partial F_y}{\partial y} \tag{30}$$
$$C_{xx}=\frac{\partial F_x}{\partial\dot{x}}\quad C_{xy}=\frac{\partial F_x}{\partial\dot{y}}\quad C_{yx}=\frac{\partial F_y}{\partial\dot{x}}\quad C_{yy}=\frac{\partial F_y}{\partial\dot{y}}$$

The linear dynamic coefficients can be expressed as:

$$\begin{aligned}\bar{K}_{xx}&=-\mathrm{Re}(\bar{F}_{x1}) & \bar{K}_{yx}&=-\mathrm{Re}(\bar{F}_{y1})\\ \bar{K}_{xy}&=-\mathrm{Re}(\bar{F}_{x2}) & \bar{K}_{yy}&=-\mathrm{Re}(\bar{F}_{y2})\\ \bar{C}_{xx}&=-Im(\bar{F}_{x1}) & \bar{C}_{yx}&=-Im(\bar{F}_{y1})\\ \bar{C}_{xy}&=-Im(\bar{F}_{x2}) & \bar{C}_{yy}&=-Im(\bar{F}_{y2})\end{aligned} \tag{31}$$

The obtained dynamic coefficients are substituted in the equation of motion for the standard Jeffcott model with rigid rotor shown in figure (3) **Zhi (1995)** to get the equivalent stiffness coefficient K_{eq}, and the whirl frequency ω_n as **:**

$$K_{eq} = \frac{(K_{xx} * C_{yy}) + (K_{yy} * C_{xx}) - (K_{xy} * C_{yx}) - (K_{yx} * C_{xy})}{(C_{xx} + C_{yy})} \quad (32)$$

$$\omega_n^{\ 2} = \frac{(K_{eq} - K_{xx}) + (K_{eq} - K_{yy}) - (K_{xy} * K_{yx})}{(C_{xx} * C_{yy}) - (C_{xy} * C_{yx})} \quad (33)$$

Also, the critical mass parameter can be evaluated as:

$$M = \frac{K_{eq}}{\omega_n^{\ 2}} \quad (34)$$

RESULTS AND DISCUSSION

The mathematical model of the bearing with geometrical and operating conditions shown in table(1) as well as the computer program prepared to solve the problem of the present wok have been verified by comparing the results for dynamic coefficients (stiffness and damping) obtained in the present work with that obtained by **Sheeja and Prabhu (1992)** as presented in figure (4). This figure clearly depict the good agreement between the results with percentage deviation less than 2%. The critical mass was also compared with that obtained by **Durany et al.(2010)** as shown in Figure (5). This figure also depicts a good agreement between the results. The deviation between the results has been computed and found to have average error of about 7%, this percentage is considered good for this validation because this paper considers the numerical solution of a transient thermo-hydrodynamic model. The effect of adding nanoparticles with different particle concentrations (0.1%, 0.5%, 1%, 1.5%, 2%) to the base oil on stiffness coefficients can be shown in figures (6a) to (6d). Figure (6a) shows the dimensional direct stiffness coefficient (Kxx) obtained in this work against the bearing eccentricity ratios. An increase in stiffness coefficient (Kxx) can be noticed for the bearing with higher eccentricity ratios. This is due to the higher load carried by the bearings in this case. Also this figure show that the higher stiffness coefficients (Kxx) can be obtained when the bearing lubricated with oil with higher particles concentrations. This is due to the higher oil viscosity when nanoparticles dispersed in the base oil with higher particle concentrations, hence, causes higher load carrying capacity and higher oil film stiffness. A percentage increase in (Kxx) of 9% and 28% has been obtained for a bearing working with eccentricity ratio of 0.4 lubricated with Nano-lubricant with particle concentration of 0.5% and 1.5% respectively. Figure (6b) shows that (Kxy) increases for the bearing with higher ratios of eccentricity when its lubricated with Nano-lubricant that has higher particle concentrations. A percentage increase of 27% has been calculated for a bearing working at eccentricity ratio of 0.4 lubricated with Nano-lubricant containing 1.5% particle concentrations of TiO_2 Nano-particles. The cross coupled stiffness (Kyx) shows negative values for all eccentricity ratios as can be shown from Figure (6c). This figure also depicts that the negative values of the (Kyx) slightly decreases for the bearing works at eccentricity ratios less than 0.4 after that it clearly decreased. The direct stiffness coefficient (Kyy), increases as the bearing lubricated with Nano-lubricant with higher particle concentrations as can be shown in figure. (6d). Figures (7a-c) show the variation of the damping coefficients with the eccentricity ratios when the bearing was lubricated with Nano-lubricant that has different particle concentrations. It can be seen from these figures that the damping coefficients (Cxx, Cxy = Cyx, Cyy) always increasing for the bearing with higher eccentricity ratios. This can be attributed to the higher load carried by the bearing in this case. Also it is obvious from these figures that the damping coefficients increase when the bearings lubricated with Nano-lubricants with higher particle concentrations of the TiO_2 nanoparticles. The percentage increase in damping coefficients (Cxx, Cxy = Cyx, Cyy) for a bearing working at an ratio eccentricity of 0.7 lubricated with

Nano-lubricant that has 0.5%, 1% and 1.5% particles concentrations of the nanoparticles has been computed and was found to be, 26%, 27% and 27.5%, respectively when compared with that lubricated with pure oil. The stiffness and damping coefficients were used to discuss the stability of the rotor bearing system. The results obtained are presented in terms of equivalent stiffness and critical mass against bearing eccentricity ratios. The effect of lubricating the bearing with Nano-lubricant containing different particle concentrations of TiO_2 nanoparticle on the equivalent stiffness coefficient can be shown in figure (8). This figure shows that the equivalent stiffness coefficient increases when the bearing lubricated with Nano-lubricant with higher particle concentrations. This can be attributed to the increasing of the direct stiffness coefficients of the oil film discussed previously which represents the main component of the equivalent stiffness as can be seen from equation (32). The percentage increase in equivalent stiffness coefficient has been calculated for a bearing working at an eccentricity ratio of 0.7 lubricated with Nano-lubricant that has 0.5% and 2% TiO_2 nanoparticles and was found to be 9% and 40% respectively. The stability of the bearings in terms of critical mass has been studied and presented in figure (9). This figure depicts that the critical mass becomes higher and hence the stability zone when the bearings lubricated with Nano-lubricant with higher particles concentrations of the nanoparticles. A 7.5% increase in critical mass was calculated for a bearing working at an eccentricity ratio of 0.5 when lubricated with oil containing 0.5% nanoparticle concentration in comparison with that lubricated with pure oil wile it becomes 39% when it was lubricated with Nano-lubricant that has 2% particle concentration. This can be explained by knowing that the critical mass is directly proportional to the equivalent stiffness coefficient as can be shown from equation (34). The figure also shows a little effect for the low particle concentrations of the nanoparticles added to the oil on the critical mass. The lubricant seems to behave like pure oil in this case. A comparison between the critical mass obtained when the oil film temperature has been considered with that obtained from the isothermal solution, and the same thing for the equivalent stiffness can be shown in figures (10&11). It is clear from these figures that the oil film temperature has a negative effect on the critical mass supported by the rotor and on the equivalent stiffness. Theoretically speaking both of the critical mass and the equivalent stiffness decreases when the oil film temperature effect was considered for the bearing works at higher eccentricity ratios $(\varepsilon \geq 0.5)$, while a little effect can be noticed when the bearings works at lower eccentricity ratios.

CONCLUSIONS

The above discussions for the obtained results lead to the following conclusions

1. The dynamic coefficients Kxx, Kxy and Kyy increase as the bearing lubricated with oil contains TiO_2 nanoparticles with higher particle concentrations. A percentage increase of 28% and 27% in Kxx and Kxy has been obtained when the bearing works at ratio eccentricity of 0.4 lubricated with Nano-lubricant that has particle concentration of 1.5%.
2. The dynamic coefficient Kyx shows negative values and decreasing as the bearings works at higher ratios eccentricity. The percentage decrease becomes higher when the bearing lubricated with oil containing higher particle concentration of the nanoparticles.
3. The equivalent stiffness coefficient increases as the bearings lubricated with oil containing higher particle concentrations of Nano-particles. An increase of 9% has been obtained for the bearing works at ratio eccentricity of 0.7 lubricated with oil containing 0.5% particle concentration of TiO_2 Nano-particles.
4. An increase of 7.5% in critical mass has been obtained for a bearing works at eccentricity ratio 0.5 lubricated with oil containing 0.5% particle concentration of the Nano-particles in comparison with that lubricated with pure oil.

5. The damping coefficients (Cxx, Cxy = Cyx, Cyy) always increasing when the bearing works at higher eccentricity ratios. Also these coefficients increase when the bearing lubricated with Nano-lubricant with higher particles concentration. Percentage increase of 26% and 27% in above damping coefficient have been obtained for the bearing works at eccentricity ratio of 0.7 lubricated with oil containing 0.5% and 1% particle concentration of TiO_2 Nano-particles in comparison with that lubricated with pure oil.
6. Considering the effect of the oil film temperature shows that it has a negative effect on the dynamic characteristics and the critical mass supported by the rotor especially for bearing works at $\varepsilon \geq 0.5$.

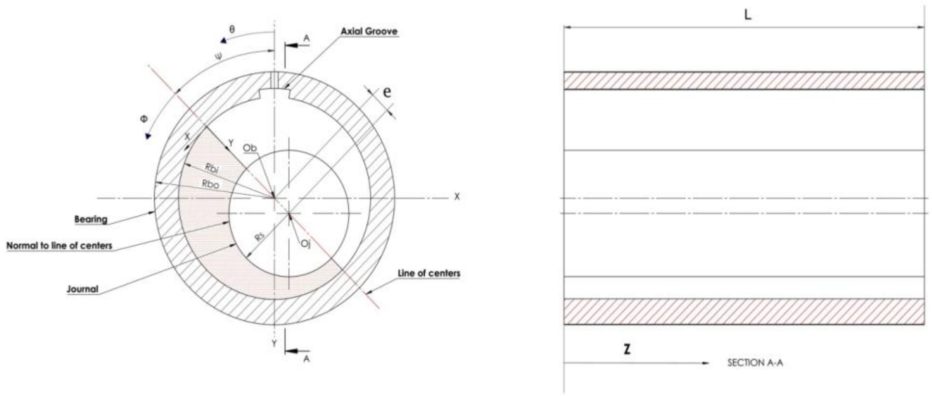


Fig.(1): Journal bearing geometry

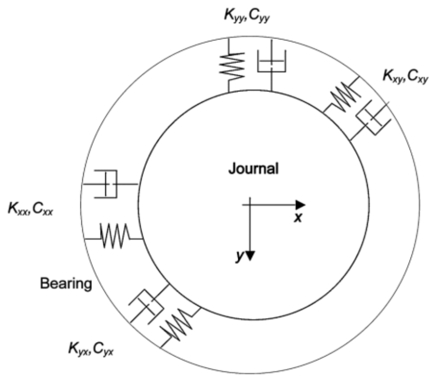


Fig.(2): Linear dynamic coefficients

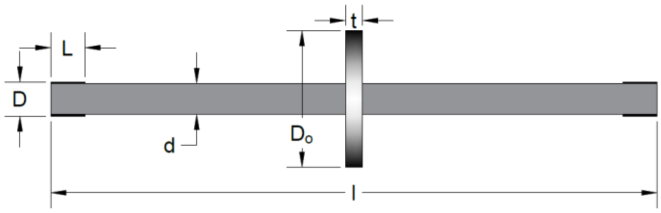


Fig.(3): Jeffcott model

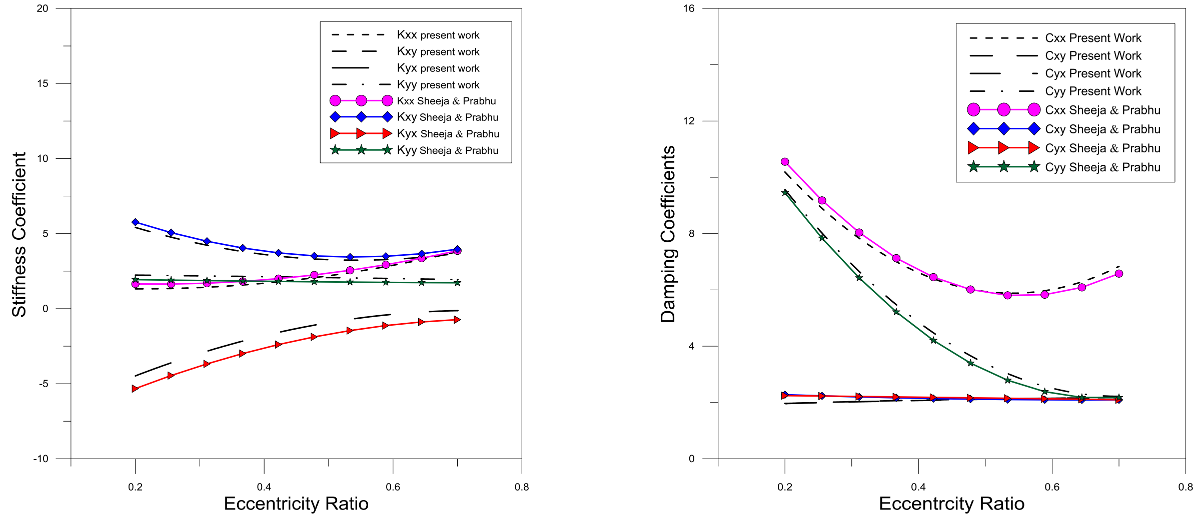


Fig.(4): Validation of the dynamic coefficients obtained in the present work

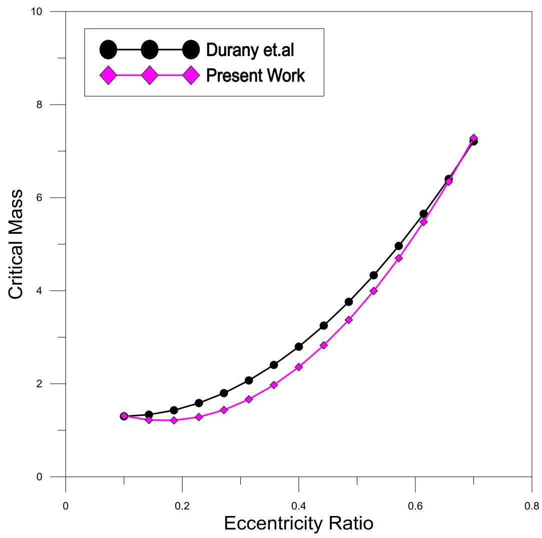
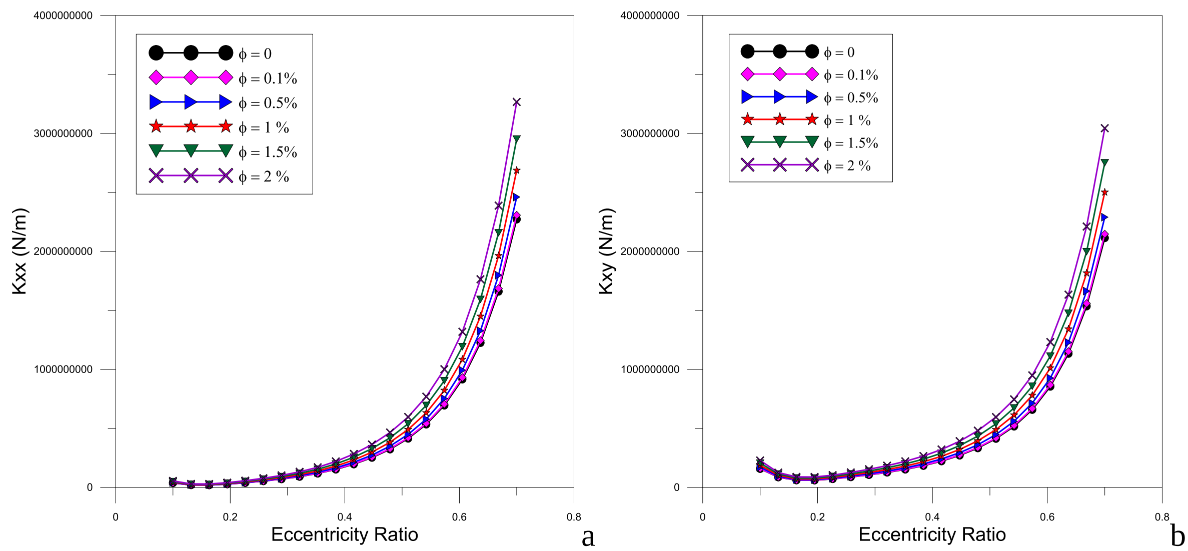


Fig.(5): Validation of the critical mass obtained in the present work



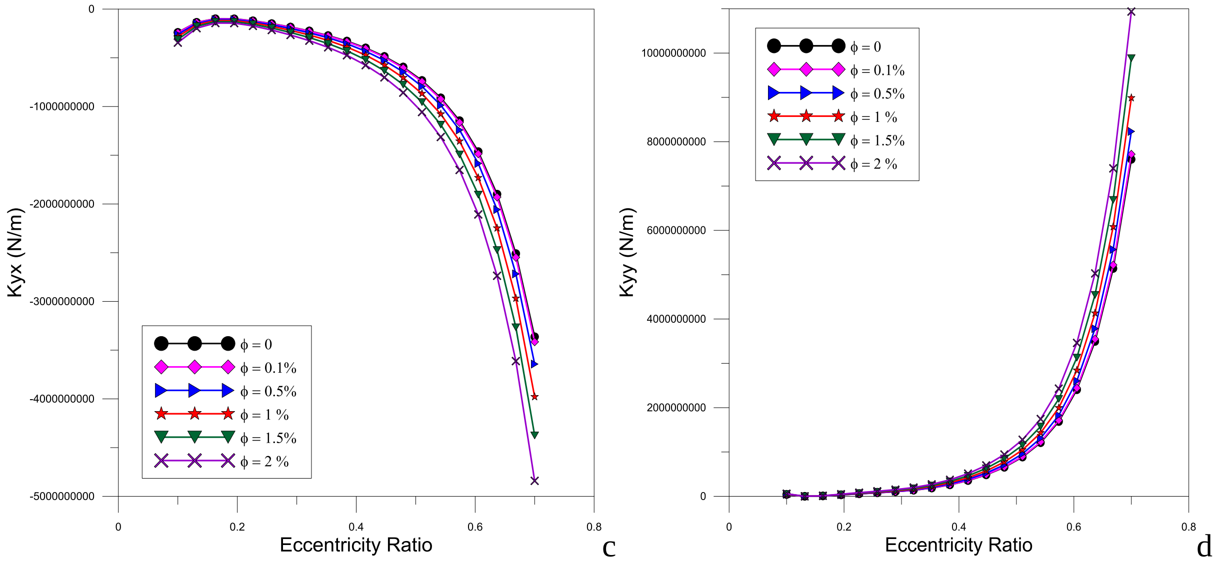


Fig.(6): Stiffness coefficients for different nanoparticle concentrations.

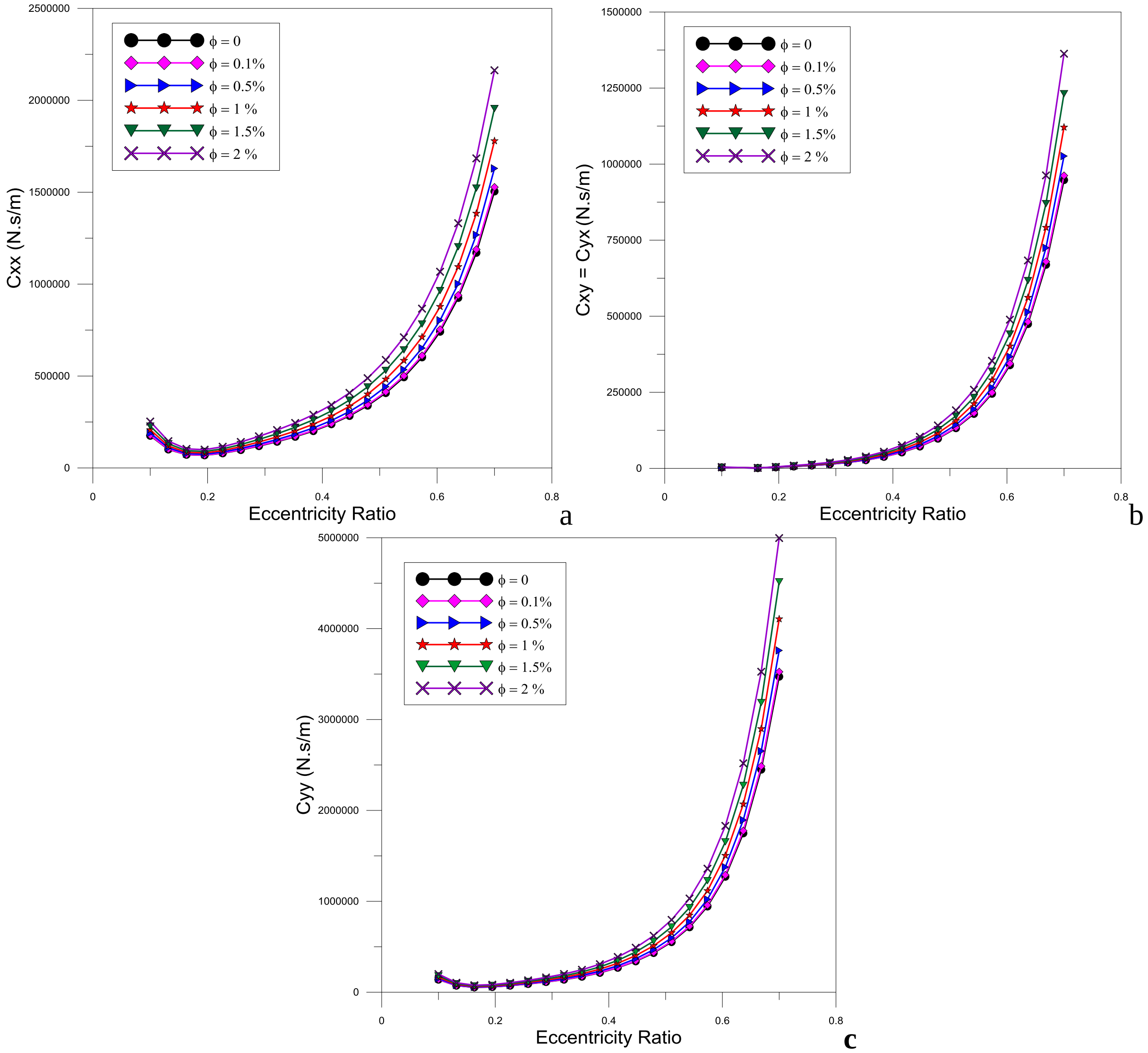


Fig.(7): damping coefficients as a function of eccentricity ratios and different nanoparticle concentrations.

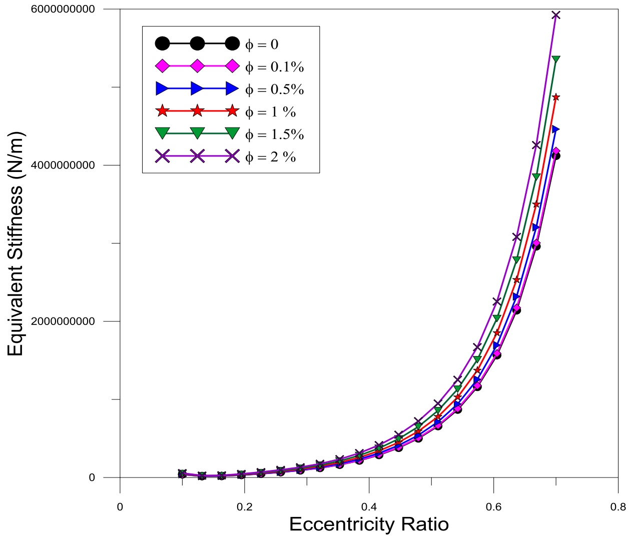


Fig.(8): Equivalent stiffness

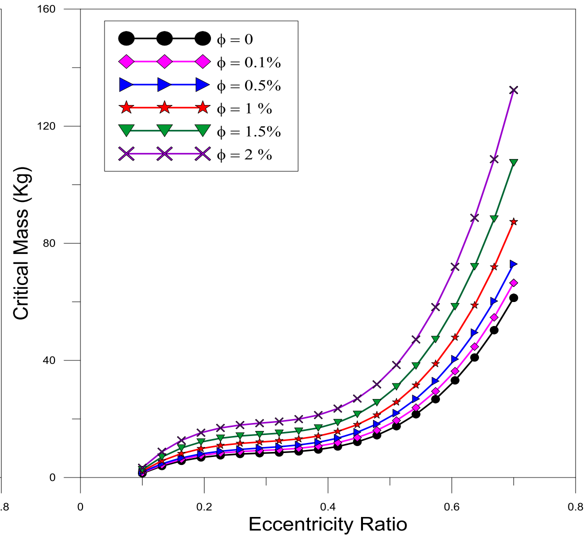


Fig.(9): critical mass

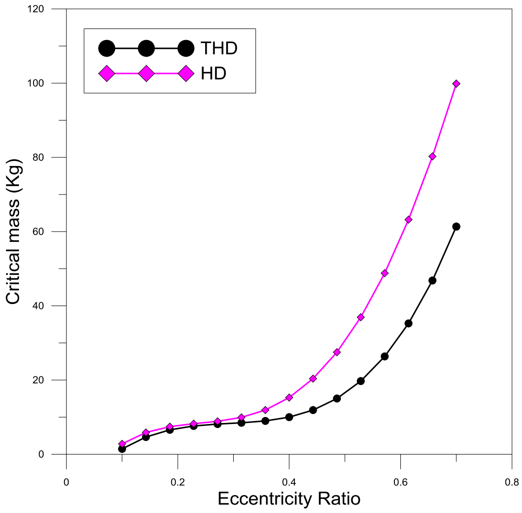


Fig.(10): Dimensional critical mass

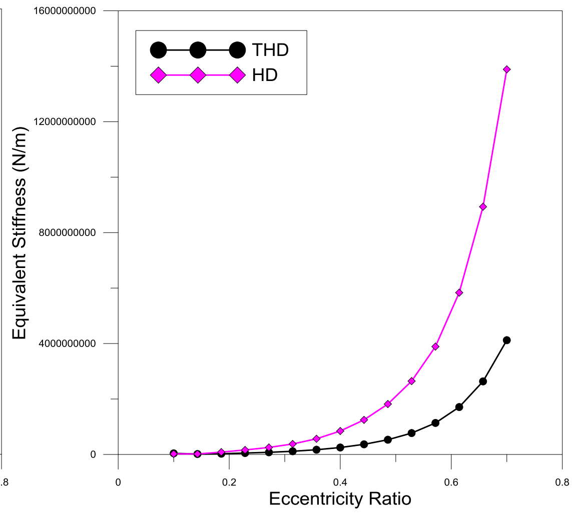


Fig.(11): Dimensional Equivalent stiffness

Table(1) : Geometric and operation parameters of the journal bearing **Ferron, et.al (1983)**

Parameter	Symbol	Value and Unit
Bearing length	L	0.08 m
External bearing radius	R_{bout}	0.01 m
Shaft radius	R_s	0.05 m
Radial clearance	c	0.0000152 m
Inlet lubricant temperature	T_{in}	40 °C
Ambient temperature	T_a	40°C
Inlet lubricant pressure	P_{in}	70000 Pa
Rotation speed	N	2000 -8000rpm
Lubricant density at inlet temperature	ρ	860 kg/m^3
Lubricant viscosity at inlet temperature	μ_o	0.0277Pa.s
Oil thermo-viscosity parameter	β	0.034
Lubricant specific heat	C_o	2000 J/kg.°C
Lubricant thermal conductivity	K_f	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	θ_g	18 deg
Nano particle density	ρ_{np}	4250 kg/m^3
Nano particle thermal conductivity	K_{np}	8.9538 W/m .°C
Nano particle specific heat	$C_{p\ np}$	686.2 J/kg.°C

NOMENCULATURE

Symbol	Description	Units
c	Radial clearance	M
$C_{i,j}$	Damping Coefficient	$N.s/m$
E	Complex number	-
F	Hydrodynamic Force	N
$h, \bar{h}$	Dimensional and non-dimensional oil film thicknesses	M
$K_{i,j}$	Stiffness Coefficient	N/m
L	Bearing length	M
O_b, O_j	Bearing, journal centers	M
P	Oil film pressure	N/m^2
$\bar{P}$	Non-dimensional oil film pressure	-
R_j	Radius of the journal	M
Q_{in}	The supply oils flow rate	m^3/s
Q_r	The Recirculation flow rate	m^3/s
R_b	Bearing radius	M
R_{bi}	Inner radius of the bearing	M
$\bar{T}_r$	The temperature of recirculation.	-
$\bar{T}_{in}$	Oil temperature at the bearing inlet.	-
$T, \bar{T}$	Dimensional and non-dimensional oil film temperature	°C
$T_\circ$	Temperature at the bearing outlet	°C
T_{mix}	Mixing oil film temperature	
U	Journal (shaft) speed	m / s
$u, \bar{u}$	Dimensional and non-dimensional fluid velocity component in x-direction	m / s
$v, \bar{v}$	Dimensional and non-dimensional Fluid velocity component in y-direction	m / s
$w, \bar{w}$	Dimensional and non-dimensional fluid velocity component in z-direction	m / s

GREEK SYMBOLS

Symbol	Description	Units
α	Dimensional ratio	-
β	Thermo-viscous coefficient	1/°C
ε	Eccentricity ratio	-
μ	Pure oil viscosity	Pa. s
$\bar{\mu}$	Dimensionless pure oil viscosity= $\frac{\mu}{\mu_\circ}$	-
$\mu_\circ$	Inlet lubricant viscosity	Pa. s
θ	Angular coordinate	Deg.
ψ	Attitude angle	Deg.
ω	Journal rotational speed	Rad/s

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