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RESEARCH

Effect of Cavitation on the Performance of Journal Bearings Lubricated with Non-Newtonian Lubricant

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A B S T R A C T

Journal bearings work under severe conditions of speeds and loads are far away from working at Newtonian condition due to the high shearing rate of the fluid in this case with cavitation at its divergent zone. The main goal of the present work is to study the journal bearing performance considering combined effects of nonlinearity behavior of the working fluid, cavitation, and the bearing liner elasticity. For this purpose, a finite length cylindrical journal bearing working with non-Newtonian lubricant modeled and analyzed numerically considering the effect of parameters. The analysis was implemented by solving the main governing equations using a suitable prepared computer program. The flow of lubricant assumed laminar, incompressible, and isothermal. Lubricant shear stress assumed to vary with a shear rate according to the power law with index coefficient (λ) varies from 0.7 to 1.3 while the cavitation effect included by implements Elrod's cavitation algorithm (ECA). The elastic deformation is considered by calculating the radial component of the bearing liner elastic deformation which is a function of the elasticity coefficient κ. Different parameters investigated in static analyses and the numerical model validated by comparing the results of pressure and density ratio obtained in the present work with that obtained by Ye et al., (1994) and they are found in a good agreement. The results obtained showed an increase in oil film pressure and the load carried by the bearing while the coefficient of friction decreases when the bearing lubricated with a non-Newtonian lubricant of a higher index.

1. INTRODUCTION

Journal bearings are simple machine elements used to support the applied load during the working of high-speed industrial machines. Reduction of power losses is a great challenge in designing such bearings. The problem of energy-saving is complicated by using multigrid oils with both temporary and permanent shear thinning and make the lubricant to behave as a Non-Newtonian fluid [1]. Remarkable works were implemented to study the influence of the non-Newtonian fluid on the performance of journal bearings. Sbeeja et al. [2] analyzed theoretically and experimentally the influence of the first normal stress difference on hydrodynamic lubrication using a cubic law fluid model for the mathematical model. The predicted results are found to be in good agreement with that obtained experimentally. A cavitation model based on the centrifugal force and surface tension which must be considered as the primary cause for the lubricant film to be in equilibrium in this zone has been proposed by Mistry et al [3]. It was found that the supply pressure has a considerable effect on the film reformation. Sharma et al. [4] studied theoretically the effect of non-Newtonian behavior of lubricant with cubic shear stress law on the elastohydrodynamic performance of slot entry journal bearing. It has been noticed that the effect of nonlinearity and flexibility coefficients considerably affect the performance parameters of such bearings. Many works are implemented to drive the compressible Reynolds equation to solve the flow at the cavitation zone of the journal bearing in order to overcome the primary formulation associated drawbacks [5-8]. All these proposed equations have been validated against available experimental or CFD published results. Computational fluid dynamics and fluid-structure interaction approaches based on actual physical models are widely used to analyze the effects of both cavitation and non-Newtonian behavior on the performance of journal bearings [9-11]. The physical models were validated by comparing the predicted results with the available experimental published results and it was found in a good agreement. Kango et al. [12] studied the performance of micro-textured journal bearings under combined influences of viscous heat dissipation and non-Newtonian rheology of lubricant. It has been noticed that the existence of the micro-texture causes a reduction in the average temperature of lubricant in comparison with the smooth surface of the bearing. Kushare and Sharma [13] present a thermo-hydrodynamic solution of two lobe journal bearing lubricated with non-Newtonian lubricant modeled by cubic law. The results obtained show that the variation of lubricant cubic law viscosity due to temperature and geometry caused a significant change in bearing performance parameters. Mehala et al. [14] Analyze numerically the performance of journal bearing coated with high tin layer lubricated with non-Newtonian lubricant working with severe conditions considering thermal effect. Numerical findings showed that both the oil film pressure and temperature are high when the lubricant behaves as a viscoelastic material ($n>1$). Meng, [15] developed a thermo-hydrodynamic model to analyze the effect of viscosity wedge associated with dimple surfaces considering cavitation. The analysis shows that the viscosity wedge effect has a pronounced influence than the cavitation effect. Miszczak [16] considered the effect of surface roughness on the flow and operating parameters of the journal bearing lubricated with non-Newtonian lubricant. It has been reviled that the transverse surface roughness has a significant impact on the operating parameters of the bearing. Sun et al. [17] studied the cavitation and whirl phenomena to evaluate the static and dynamic characteristics of journal bearing. It has been noticed that an efficient evaluation of the static and dynamic performance was obtained. Dang et al. [18] in his review on the effect of most important types of non-Newtonian lubricants on the performance of the journal bearing noticed that an improvement in the bearing performance was obtained when using such lubricants rather than Newtonian one. Lampaert and Ostayen [19] presented an exact thin-film lubrication simulation for Bingham plastic fluid. The model was validated by comparing the results obtained for the finite and infinite length journal bearings with that published in literatures and it was found in a good agreement. Bertocchi et al. [20] extended the commonly used cavitation algorithm to include the effects of fluid compressibility, piezo viscosity, and non-Newtonian behavior of the lubricant and applied it to analyze the two-dimensional problems. The proposed method was validated by comparing the obtained results with that obtained from CFD analysis. The effect of using ferrofluid, which displays non-Newtonian behavior, on the performance of journal bearings was investigated by Osman et al. [21]. It was concluded that the flow index, has a large effect on the bearing performance. Dien and Elrod [22], perturbed the film pressure and velocity fields for the non-Newtonian fluid in order to drive the Reynolds equation for a finite length bearing. Ye et al. [23]

developed a new numerical algorithm to predict cavitation in several fluid-film bearings. The algorithm has solution speed and higher computational precision. Chetti and Zouggar, [24] studied numerically the effect of elastic deformation on the performance characteristics of a finite-width journal bearing working with non-Newtonian lubricants following the power law model. Chen et al. [25] investigated elastohydrodynamic lubrication of journal bearing including cavitation and the bearing elasticity using the fluid-structure interaction method. The effects of shaft angular speed and the couple stress fluid on the cavitated finite journal bearing are investigated by Mahdi et al. [26]. Elrod cavitation algorithm (ECA) was used to solve the Reynolds equation. The results show that the non-Newtonian fluids (couple stress fluids) enhance the film pressure and the load carried by the bearing, with a small drop in the side leakage flow. Effects of non-Newtonian models such as Rabinowitsch and couple stress fluid together with the elastic deformation of the bearing liner on the journal bearing performance have been extensively studied [27-29]. Results obtained show that bearing deformation and non-Newtonian behavior have significant effects and cannot be ignored. Saber and El-Gamal [30] Studied the effect of bearing elastic deformation on its stability. They show that the shorter bearing is less stable than the longer one for any value of the elastic coefficient. Aki Linjamaa et al. [31] developed a parameterized calculation model for hydrodynamic journal bearing considering the elastic and thermal deformations of the bearing liner and the shaft surfaces. It is concluded that elastic and thermal deformations are partly canceling each other out at the loaded side of the bearing. CFD-FSI technique used to study the combined effects of cavitation, elastic deformation, and oil film temperature on the oil film pressure generated in journal bearings lubricated with different types of lubricants [32-34]. The results show that the thermal effect has a significant effect than the elastic deformation. The main goal of the present work is to study the combined effects of bearing elastic deformation on the performance of journal bearing lubricated with non-Newtonian lubricant considering the cavitation effect. Kumar et al. [18] recommend that the implementation of cavitation and elastic deformation with non-Newtonian behavior of the lubricant are very important to investigate the performance of such bearing. Also, Chetti and Zouggar [24] studied the effect of elastic deformation and non-Newtonian behavior of the lubricant on the performance of the journal bearing but without considering the cavitation effect which is covered in the present work.

2. THEORETICAL MODEL

Figure 1 shows the geometry and the coordinates system of the journal bearing used in the present work. In the power-law model, implies the power-law index λ (presents flow behavior), under the non-Newtonian behavior of the lubricant with viscosity model ($\mu = \mu_o\left(\partial v_x / \partial y\right)^{(\lambda-1)}$), the modified Reynolds equation for isothermal and laminar flow in such bearing operating with Non-Newtonian modeled by power law is expressed as [21,22]:

$$\frac{\partial}{\partial x}\left(\frac{\rho h^{\lambda+2}}{\mu_o}\frac{\partial P}{\partial x}\right)+\lambda\frac{\partial}{\partial z}\left(\frac{\rho h^{\lambda+2}}{\mu_o}\frac{\partial P}{\partial z}\right)=6\lambda U\frac{\partial \rho h}{\partial x} \quad (1)$$

where; λ is the power-law index.
It can be observed that when $\lambda = 1$ equation (1) reduces to the Newtonian Reynolds equation.

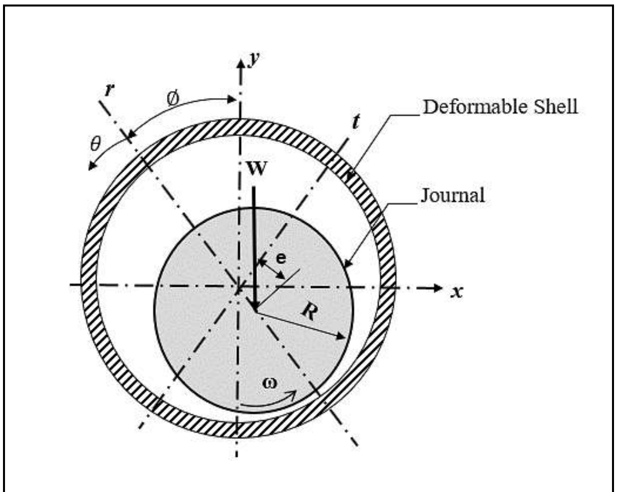


Fig. 1. Journal bearing geometry.

The fluid density assumed to be constant at the active zone of the bearing and equation (1) can be written as:

$$\frac{\partial}{\partial x}\left(h^{\lambda+2}\frac{\partial P}{\partial x}\right)+\lambda\frac{\partial}{\partial z}\left(h^{\lambda+2}\frac{\partial P}{\partial z}\right)=6\lambda\mu_o U\frac{\partial h}{\partial x} \quad (2)$$

The pressure remains constant at the cavitation zone (i.e., $P = P_{cav}$) and equation (1) can be expressed as:

$$6\lambda U \frac{\partial \rho h}{\partial x} = 0 \quad (3)$$

The mass-conserving Jakobsson, Floberg Olsson (JFO) boundary condition [35-36] that covers the oil film rupture and reformation region was adopted in the numerical analysis of different workers such as Elrod and Elord and Adams [37]. They obtained a simple equation that combined the oil film and cavitation zones depending on the fluid continuity by the pressure – density relationship. They introduced a new parameter called the fractional film content (φ). The film pressure (p), density (ρ), and the switch function linked through the bulk modulus of lubricant as [37]:

$$g\beta = \rho \frac{\partial P}{\partial \rho} = \varphi \frac{\partial P}{\partial \varphi} \quad (4)$$

The above equation can be integrated and gives:

$$P = P_{cav} + g\beta \ln(\varphi) \quad (5)$$

Where

$$\varphi = \frac{\rho}{\rho_{cav}} \quad (6)$$

Where P_{cav} and ρ_{cav} are the lubricant pressure and density at the cavitation region. By using the cavitation index (switch function $g(\varphi)$), the terms of film pressure to be ignored in the rupture zone or to be kept in the active zone. The switch function proposed is given as:

$g = 0$ in the inactive zon $\varphi < 1$

$g = 1$ in the active zone $\varphi \geq 1$

The density ratio φ in the active zone is somewhat larger than unity, therefore, equation (4) can be expressed in a simpler formula as:

$$P = P_{cav} + g\beta(\varphi - 1) \quad (7)$$

Introducing equations (5) and (6) into equation (1), the resulting mass-conservative form of the Reynolds equation which considered non-Newtonian and cavitation effects can be obtained as follows:

$$\frac{\partial}{\partial x}\left(\frac{g\beta h^{\lambda+2}}{\mu}\frac{\partial \varphi}{\partial x}\right) + \lambda \frac{\partial}{\partial z}\left(\frac{g\beta h^{\lambda+2}}{\mu}\frac{\partial \varphi}{\partial z}\right) = 6\lambda U \frac{\partial}{\partial x}(\varphi h) \quad (8)$$

In equation (8) the only unknown is φ rather than oil film pressure which can be defined as the fractional-film content for the lubricant film in the cavitated zone and dimensionless density ratio in the active zone. It satisfies the pressure conditions at the cavitation and full film region and also automatically implements the oil film reformation and rupture.

By using the following relations; $x = R\theta$, $z = \bar{z}L$, $\beta = \mu_o UR\bar{\beta}/C^2$, $\mu = \mu_o \bar{\mu}$, $h = \bar{h}C$, equation (8) can be re-written in dimensionless form as :

$$\frac{\partial}{\partial \theta}\left(g\bar{h}^{\lambda+2}\frac{\partial \varphi}{\partial x}\right) + \lambda \left(\frac{R}{L}\right)^2 \frac{\partial}{\partial \bar{z}}\left(g\bar{h}^{\lambda+2}\frac{\partial \varphi}{\partial \bar{z}}\right) = 6\lambda \frac{1}{\beta}\frac{\partial}{\partial \theta}(\varphi \bar{h}) \quad (9)$$

It is clear from equation (9) as λ=1, the Reynolds equation converts into the classical mass-conservative one for Newtonian lubricant. It can be solved numerically by using the finite-difference approach to calculate the φ distribution which is transformed later to the pressure field using equation (7).

The boundary condition at the bearing edges is given as:

$$\varphi = \exp\left(\frac{\bar{P}_a - \bar{P}_{cav}}{\bar{\beta}}\right) \quad (10)$$

Where; $\bar{P}_a = 0$ and $\bar{P}_{cav} = 0$ at $\bar{z} = 0$ and 1.

Finally, the pressure distributions of the lubricant at each zone of the bearing are found from the fractional-film content, φ, as:

$$\bar{P} = \begin{cases} \bar{P}_{cav} + \bar{\beta}\ln(\varphi) & \text{if } \varphi \geq 1 \\ \bar{P}_{cav} & \text{if } \varphi < 1 \end{cases} \quad (11)$$

2.1 FLUID FILM THICKNESS

The fluid film thickness at any point in the bearing considering liner elasticity effect can be evaluated in dimensionless form as [24]:

$$\bar{h} = h / c = 1 + \varepsilon \cos\theta + \kappa \bar{P} \quad (12)$$

where κ is the elastic coefficient described by:

$$\kappa = \frac{\mu URt(1-\nu^2)}{EC^3} \quad (13)$$

2.2 BEARING PARAMETERS

The load capacity components of bearing in the *r* and *t* directions are defining as [24]:

$$\overline{W}_r = \int_0^1 \int_0^{2\pi} \overline{P} \cos\theta d\theta d\overline{z} \quad (14)$$

$$\overline{W}_t = \int_0^1 \int_0^{2\pi} \overline{P} \sin\theta d\theta d\overline{z} \quad (15)$$

The load capacity and attitude angle ϕ can be determined from the following equations:

$$\overline{W} = \sqrt{\overline{W}_r^{\,2} + \overline{W}_t^{\,2}} \quad (16)$$

$$\phi = \tan^{-1}\left(-\frac{\overline{W}_t}{\overline{W}_r}\right) \quad (17)$$

The lubricant side-leakage flow of bearing can be obtained from the following equation [24]:

$$\overline{Q}_s = \int_0^{2\pi} \frac{\overline{h}^{\lambda+2}}{12} \left.\frac{\partial\overline{P}}{\partial\overline{z}}\right|_{\overline{z}=0} d\theta \quad (18)$$

The viscous friction force of shearing film fluids can be computed from [24]:

$$\overline{F}_f = \int_0^1 \int_0^{2\pi} \left(\frac{1}{\overline{\overline{h}}} + \frac{\overline{h}^{\lambda}}{2\lambda} \frac{\partial\overline{P}}{\partial\theta}\right)^{\lambda} d\theta d\overline{z} \quad (19)$$

The coefficient of the friction can be calculated as:

$$C_f = \frac{\overline{F}_f}{\overline{W}} \quad (20)$$

3. COMPUTATIONAL ALGORITHM

The numerical study of a journal bearing considering the effects of cavitation, deformation of the bearing liner, and non-Newtonian behavior of the lubricant required the solution of equation (9) with appropriate boundary conditions. The finite-difference technique was adopted to solve the Reynolds equation using a direct iterative procedure with successive under relaxation (SUR) to find the φ distributions. The computational process involves the following steps:

1. Discretize the bearing to a suitable number of nodes in circumferential and axial directions. In the present work, a mesh size of 120×20 in the circumferential and axial directions has been adopted.
2. Input the operating conditions and lubricant properties.
3. Calculate the initial value of the attitude angle ∅ just for the first step of the solution process. It can be predicted as:

$$\phi = \tan^{-1}\left(\frac{\pi}{4\varepsilon}\left(1-\varepsilon^2\right)^{1/2}\right) \quad (21)$$

4. Assume an initial value of φ and g in the active region for the first iteration.
5. Assume an initial guess for the circumferential oil film pressure distribution in order to obtain the film thickness at each node from equation (12).
6. Compute the new distributions of φ by an iterative solution of equation (9) using a suitable under - relaxation factor. The switch function at all points is updated after each iteration based on new values of φ. The iteration is stopped when the following convergence criterion of the variable φ is satisfied:

$$\left|\frac{\sum\left(\varphi_{i,j}^{m+1} - \varphi_{i,j}^{m}\right)}{\sum \varphi_{i,j}^{m+1}}\right| \quad (22)$$

Where m is the iteration number.

7. Calculate the film pressure from equation (11), and then the updated film geometry.
8. Compute the components of the load capacity from the equations (14) and (15), then a new value of attitude angle ϕ can be obtained from equation (17). The iteration is stopped when the convergence criterion of ϕ reaches 10^{-4}.

$$\left|\phi^{m+1} - \phi^{m}\right| \le 10^{-4} \quad (23)$$

9. Calculate the performance characteristics of bearing.

The global procedure for the numerical computation is described in the flow chart shown in Appendix (A).

4. RESULTS AND DISCUSSION

The results for the performance analysis of cavitated, deformable finite length journal bearing lubricated with non-Newtonian lubricant (λ = 0.7, 1.3) working at different values of eccentricity ratios ($\varepsilon = 0.2-0.8$) are presented and discussed here. The power-law index λ=0.7 refers to shear thinning (pseudo-plastic) lubricant while λ=1 and 1.3 refer to the Newtonian and dilatant lubricants, respectively. The mathematical model and the computer program of the present work have been validated by comparing the predicted results of the film fraction φ and pressure distributions for the journal bearing of the present work lubricated with Newtonian oil that has a bulk modulus $\bar{\beta}=40$ working at an eccentricity ratio of 0.6 with that obtained by Yu and Keith [23] as presented in Figs. 2 and 3. These figures show obviously that the results are in a good agreement and the computer program can be used with high confidence to study the bearing performance.

Table 1. Geometric and operating parameters of the journal bearing used in this work.

Bearing Length	L=60
Journal Diameter	D=60
Radial clearance	$C=150\mu m$
Rotational speed	2000 rpm
Atmospheric pressure	$P_a=0.0$
Inlet lubricant viscosity	$\mu_o=0.0277$ pa. s
Cavitation pressure	$P_{cav}=0.0$ pa
Dimensionless Bulk modulus	$\bar{\beta}=40$
Power law index	$\lambda=0.7,1.0,1.3$
Elastic deformation parameter	$\kappa=0.0,0.02$
Eccentricity ratio	$\varepsilon=0.2-0.8$

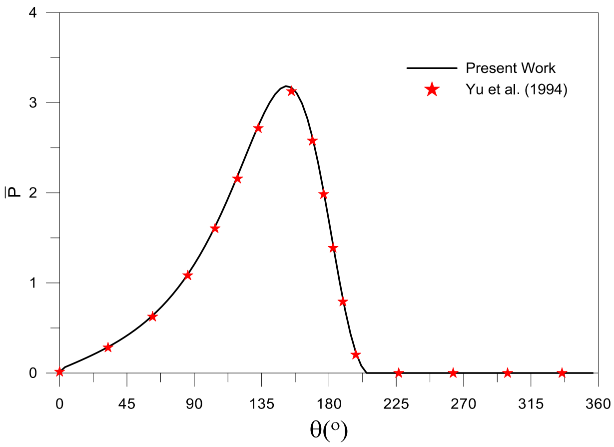


Fig. 2. Comparison between the predicted pressure distributions with that obtained by Yu et al. (1994).

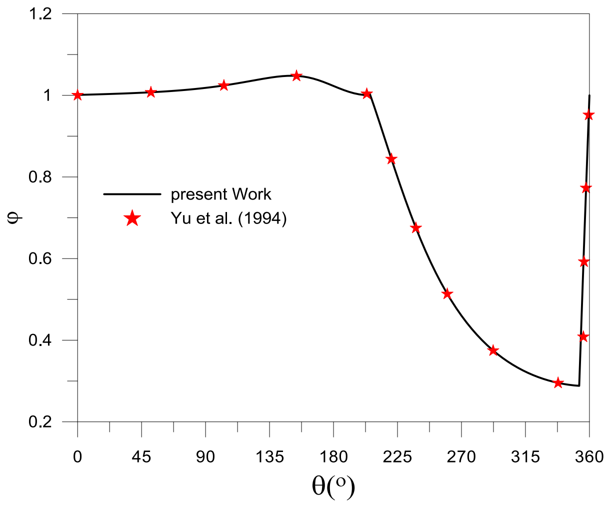


Fig. 3. Comparison between the predicted φ distributions with that obtained by Yu et al. (1994).

Figure 4 shows a comparison between the circumferential pressure distribution of rigid and elastic bearings lubricated with non-Newtonian lubricants that have different power-law indices. It can be observed from this figure that the oil film pressure increases by 38.2 % and decreases by 32 % for the bearing lubricated with shear thickening and shear thinning oil in comparison with that lubricated with Newtonian oil. This is can be attributed to the higher viscosity of the shear thickening lubricant and lower viscosity of the shear-thinning one in comparison with Newtonian lubricant. The percentage increase in oil film pressure becomes 20 % when the bearing elastic deformation is considered for the bearing lubricated with shear-thickening lubricant while it becomes negligible when the bearing lubricated with shear-thinning lubricant. This can be attributed to the sensitivity of the oil film pressure to the change in oil film thickness.

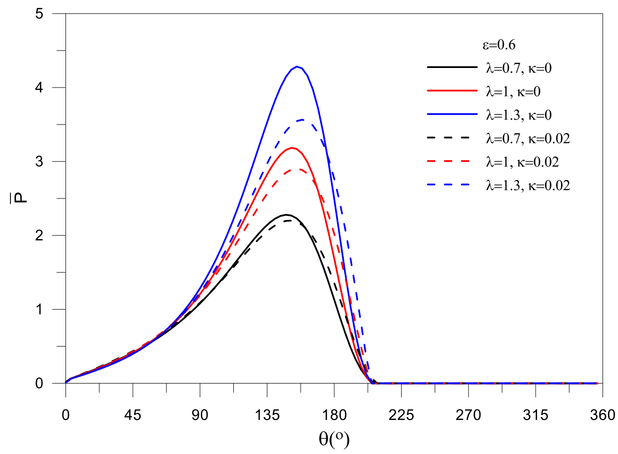


Fig. 4. Dimensionless full-film pressure distribution for various values of index λ for rigid and deformable bearing.

Figure 5 displays the fractional film content distributions at the mid-plane of the bearing when it works at eccentricity ratio $\varepsilon = 0.6$, and lubricated with a non-Newtonian lubricant that has different values of λ. Two areas can be distinguished in this figure, the first where $\varphi \geq 1$ which represents full film or active region, and the second where $\varphi \leq 1$ which represents the cavitation zone.

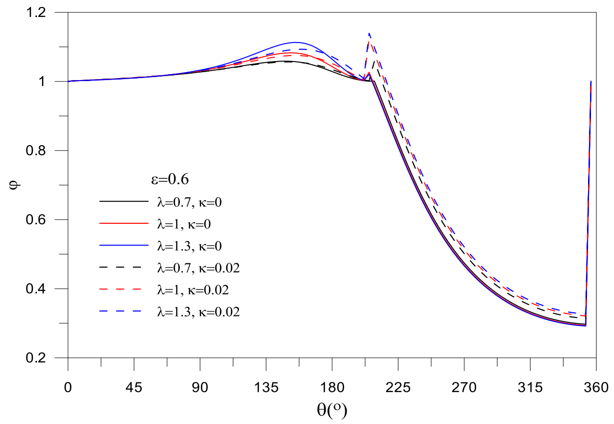


Fig. 5. Fractional film content distribution for various values of index λ for rigid and deformable bearings.

The variation of φ in the active region is very small due to the large bulk modulus, while it drops significantly to $\varphi = 0.4$ in cavitation zone. due to the increase of the gas phase to the liquid phase in this case. This figure also revealed that φ increases when the bearing lubricated with non-Newtonian lubricant that has a higher value of λ, in the active zone for both rigid and elastic bearings, while it is not affected in the cavitation zone. Also, it is obviously shown that the values of φ for the rigid bearing in the active and cavitation zones is lower than that for deformable one.

Figure 6 presents the variations of dimensionless load-carrying capacity $\bar{W}$ with the eccentricity ratio for rigid and elastic bearings lubricated with non-Newtonian lubricant. It can be observed that $\bar{W}$ growths as the bearing working at higher ε and λ. The load carried by the bearing works at an eccentricity ratio of 0.6 increased by 36.3 % when it is lubricated with shear-thickening oil while it decreases by 10 % when the bearing lubricated with shear-thinning lubricant in comparison with that lubricated with Newtonian lubricant. This can be attributed to the higher viscosity of the shear thickening oil in comparison with that of shear-thinning lubricants. Also, it can be noticed that the $\bar{W}$ for the rigid bearing is greater than that of flexible (deformable) one because of the higher film pressure generated in this case. The variation of the $\bar{W}$ becomes appreciable at a higher value of ε (i.e. 0.8) and for λ=1.3 (viscoelastic lubricant).

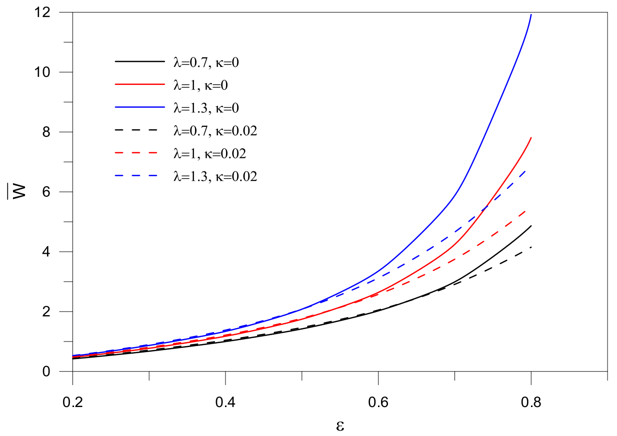


Fig. 6. Effect of λ on dimensionless load carrying of rigid and deformable bearing

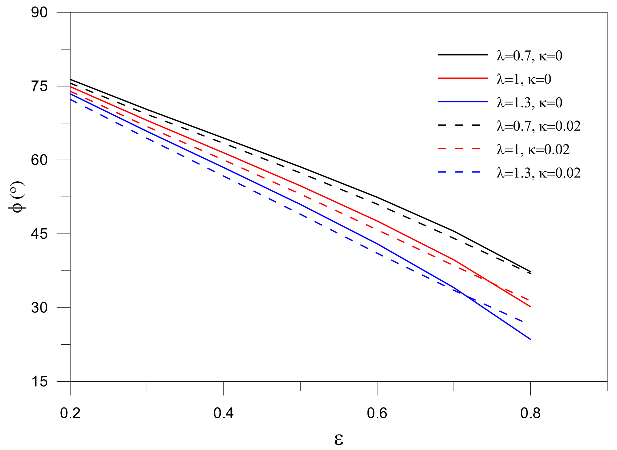


Fig. 7. Effect of λ on attitude angle of rigid and deformable bearing

The variation of the attitude angle ϕ for a bearing working at different eccentricity ratios lubricated with a non-Newtonian lubricant that has different values of λ for rigid and flexible bearings is presented in Fig. 7. This figure shows that the attitude angle decreases as the bearing works at higher ε and lubricated with oil that has higher values of λ. The percentage decrease in attitude angle for a bearing working at an eccentricity ratio of 0.6 and lubricated with shear-thickening oil has been calculated and found to be 12 % while it increases by 9 % when the bearing lubricated by shear thinning lubricant in comparison with that lubricated with Newtonian one. It can also be observed from this figure that the elasticity coefficient of the bearing has little effect on the attitude angle values.

Figure 8 shows the variation of the friction coefficient with ε for a journal bearing lubricated with non-Newtonian lubricant. It is observed that the friction coefficient decreases with an increase of ε and the power law index of the lubricant due to the higher pressure generated in this case and hence higher $\bar{W}$. This figure also shows that the friction coefficient at a higher value of ε of the non-rigid bearing is higher than that of rigid bearing and this is more noticeable when the bearing lubricated with shear-thickening lubricant compared with those lubricated with shear-thinning lubricant due to the greater oil film thickness which causes a decrease in the load carried by the bearing in this case.

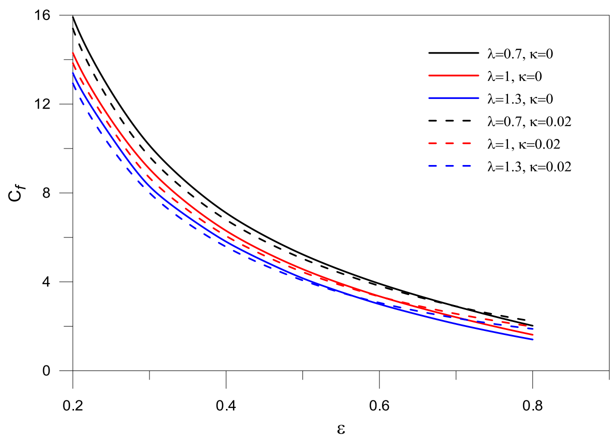


Fig. 8. Effect of λ on coefficient of friction for rigid and deformable bearing.

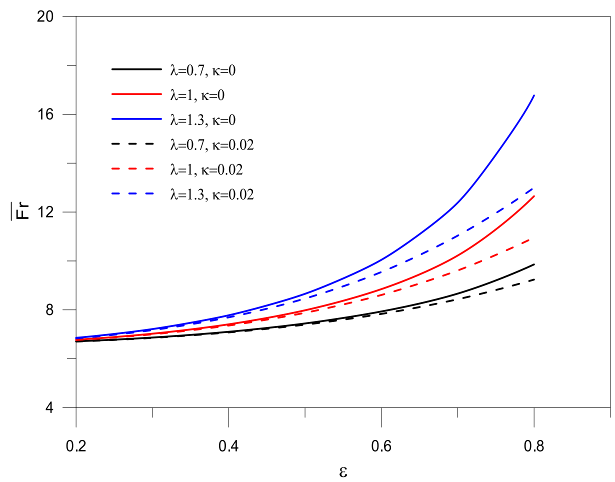


Fig. 9. Effect of λ on dimensionless friction force of rigid and deformable bearing.

The effect of lubricating the bearing with a non-Newtonian lubricant that has different values of λ on the friction force $\bar{Fr}$ is presented in Fig. 9. This plot shows the variation of friction force against ε for flexible and rigid bearings. It can be seen from this figure that $\bar{Fr}$ rises with the increase of ε and λ. This is due to the higher oil film pressure generated at a higher eccentricity ratio and higher shear stress induced in the oil film as a result of higher oil viscosity. It seems that the friction force decreases when considering the elastic deformation of the bearing liner due to the increase in oil film thickness which causes lower oil film pressure and shear rate effect of the bearing elasticity.

Figure 10 displays the increasing of the dimensionless side leakage flow $\bar{Q}_s$ with the ε for non-deformable and deformable bearings lubricated with the non-Newtonian lubricant of different λ due to the increase of oil film pressure in this case. Also, it is observed that the growth in $\bar{Q}_s$ is significant for a bearing lubricated with shear-thinning fluid rather than lubricated with shear thickening fluid. It can be observed that the bearing side leakage flow increases by 13%-21.3% when the bearing works at eccentricity ratios 0.4 and 0.6 and lubricated with a lubricant with λ=0.7 and 1.3 respectively. Also, it can be seen that κ has an insignificant effect on the $\bar{Q}_s$. This can be better from the heat removing point of view.

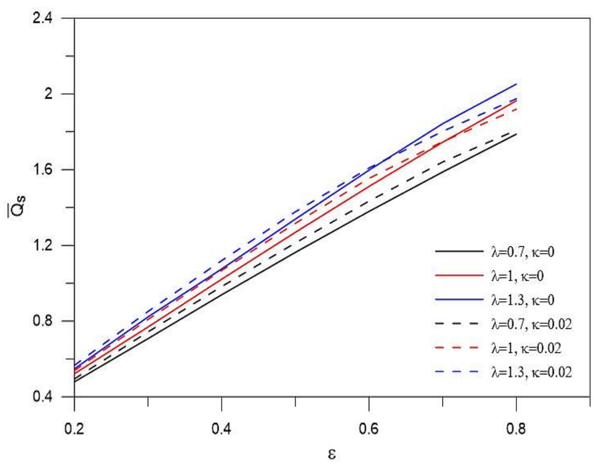
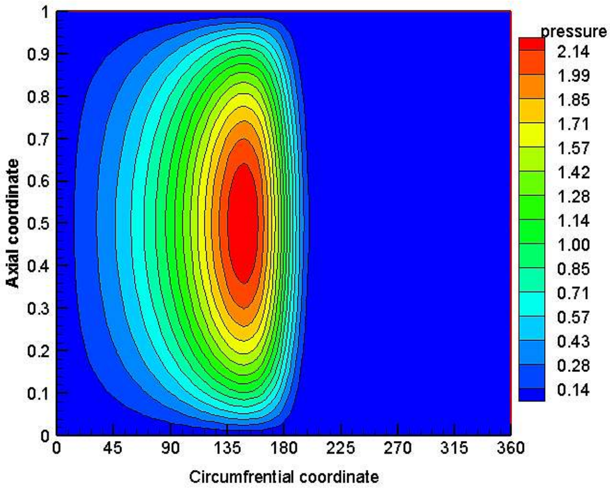
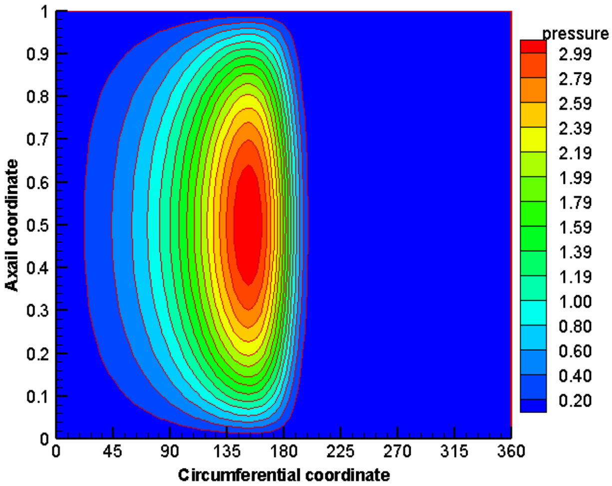


Fig. 10. Effect of λ on dimensionless side leakage of rigid and deformable bearing.

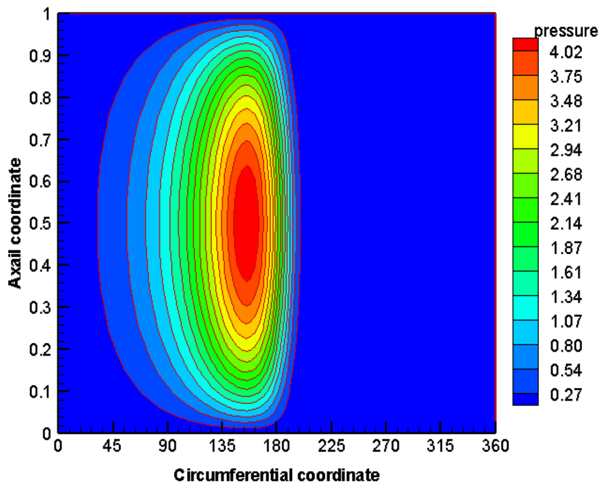
Figures 11 and 12 compare the circumferential and axial distributions of dimensionless oil-film pressure $\bar{P}$ for both rigid and deformable bearings working at ε = 0.6 lubricated with the non-Newtonian lubricant of different λ. It is observed that $\bar{P}$ grows up with increasing λ for both rigid and deformable bearings due to the higher oil viscosity in this case. However, the pressure distributions for the rigid bearing is larger than the deformable one due to the smaller oil film thickness in this circumstance.



(a) λ = 0.7

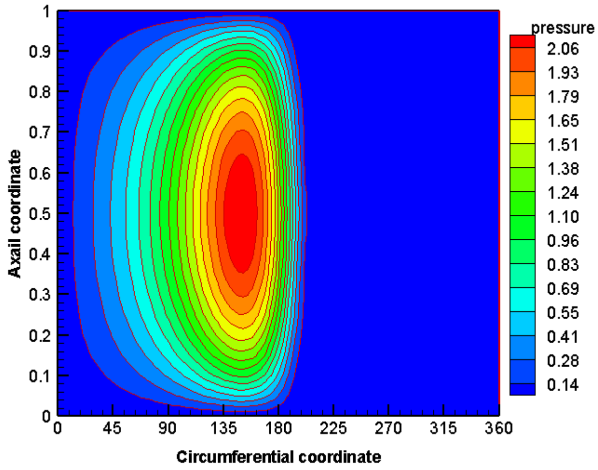


(b) λ=1.0

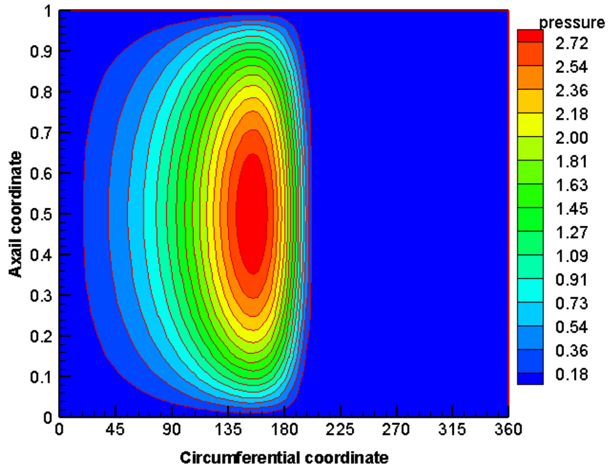


(c) λ=1.3

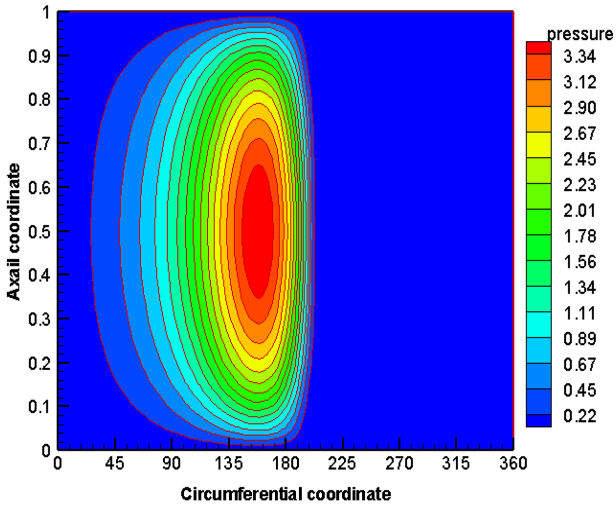
Fig. 11. Distribution of dimensionless pressure $\bar{P}$ for a rigid bearing (κ=0) lubricated with: (a) pseudo-plastic lubricant, (b) Newtonian lubricant, and (c) dilatant lubricant.



(a) λ= 0.7



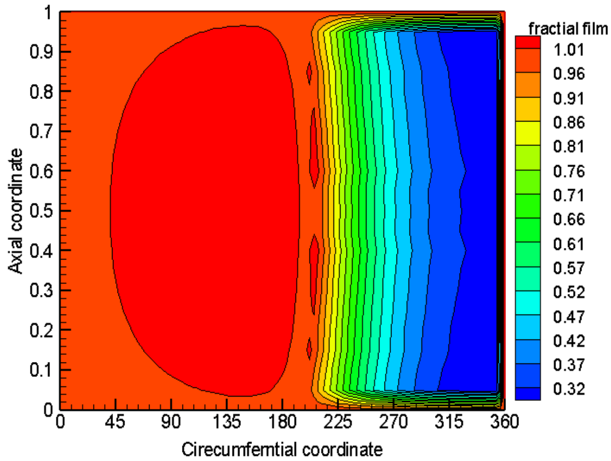
(b) λ=1.0



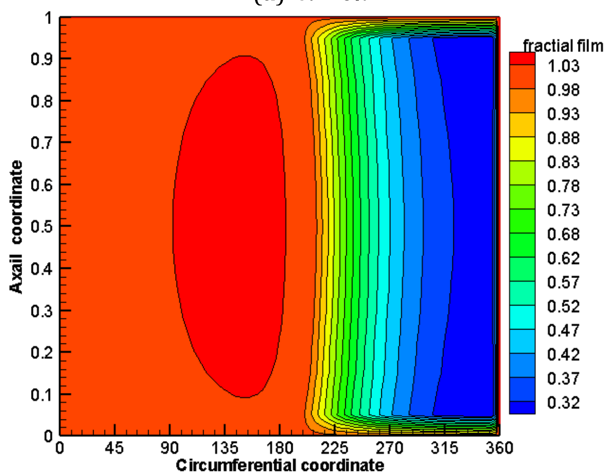
(c) λ=1.3

Fig. 12. Distribution of dimensionless pressure $\bar{P}$ for a flexible bearing (κ = 0.02) lubricated with: (a) pseudo-plastic lubricant, (b) Newtonian lubricant, and (c) dilatant lubricant.

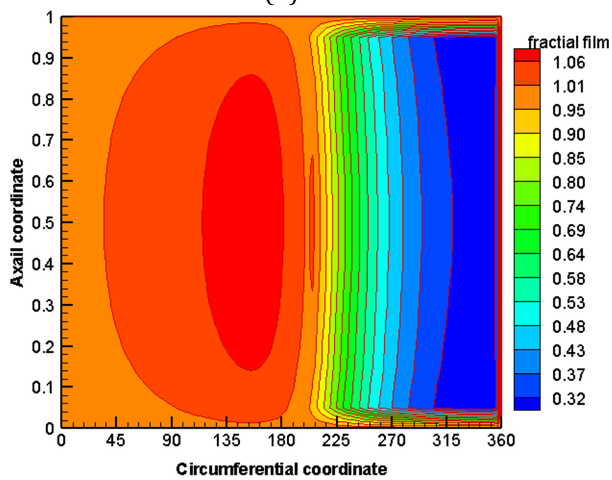
Figures 13 and 14 show the variation of φ in axial and circumferential directions for both rigid and flexible bearings operating at an eccentricity ratio of 0.6 and lubricated with Newtonian and non-Newtonian lubricants. As shown in figures, the contours divided into two areas, one is the convergent (active or full film) region and the second is the divergent (cavitation) zone.



(a) λ = 0.7

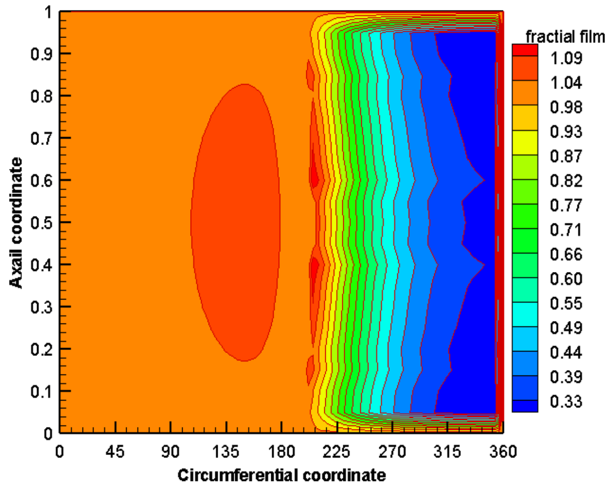


(b) λ=1.0

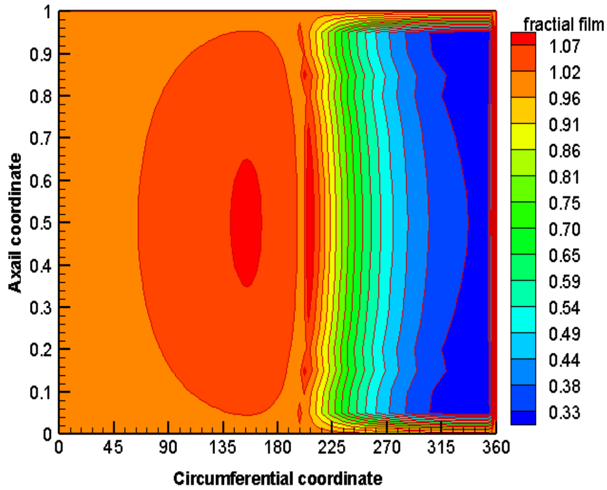


(c) λ=1.3

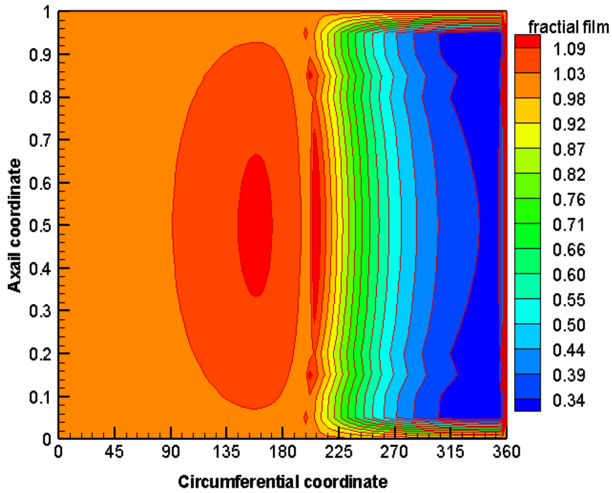
Fig. 13. Distribution of fractional film content φ for a rigid bearing (κ = 0) lubricated with: (a) pseudo-plastic lubricant, (b) Newtonian lubricant, and (c) dilatant lubricant



(a) λ= 0.7



(b) λ=1.0



(c) λ =1.3

Fig. 14. Distribution of fractional film content φ for a flexible bearing ((κ = 0.02) lubricated with: (a) pseudo-plastic lubricant, (b) Newtonian lubricant, and (c) dilatant lubricant.

Figure 13 shows that φ (dimensionless density) growths at the active zone of the rigid bearing with the increase of λ while, it was not affected at the cavitation zone.

For flexible bearing, the values of φ in the active zone are seen to be lower than that in the rigid bearing. It can also be observed that the values of φ are larger than that in the rigid bearing with more pronounced when the bearing lubricated with dilatant lubricant λ=1.3.

5. CONCLUSION

From the previous discussion of results for the performance characteristics of an elastic journal bearing lubricated with non-Newtonian lubricant considering cavitation effect, the following concluding remarks are drawn:

1. The power law index λ has a considerable effect on the full-film pressure distributions. Oil film pressure obviously increases when rigid or deformable bearing lubricated with a lubricant that has higher values of λ.
2. The hydrodynamic film pressure, dimensionless load-carrying capacity, attitude angle, and the friction force are decreased for the elastic bearing lubricated with both Newtonian and a non-Newtonian lubricant in comparison with the rigid one.
3. The dimensionless density φ of rigid bearing is enhanced with the increase in λ and this becomes more apparent when used dilatant lubricant while it remains constant at the cavitation zone.
4. The distribution of dimensionless density φ in the rupture zone of a deformable bearing is greater than that of the rigid bearing.
5. Bearing liner elastic deformation has a more pronounced effect on the performance characteristics of the bearing lubricated with non-Newtonian lubricant works at higher eccentricity ratios.

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NOMENCLATURE

c : Radial clearance (m)
C_f : Friction coefficient
e : Eccentricity (m)
E : Bearing modulus of elasticity
$\bar{F}_r$: Dimensionless viscous friction force
g : Switch function
h : Film thickness (m)
$\bar{h}$: Dimensionless lubricant film thickness, $\bar{h} = h/c$
L : Length of the bearing (m)
P : Hydrodynamic oil film pressure (pa)
$\bar{P}$: Dimensionless hydrodynamic pressure,
P_a : Ambient pressure (Pa)
P_{cav} : Cavitation pressure (Pa)
$\bar{Q}_s$: Dimensionless Leakage flow rate
R : Radius of the journal (m)
t : Thickness of bearing liner, m
v_x : velocity component in the circumferential direction
$\bar{W}$: Total load capacity (N)
$\bar{W}_r, \bar{W}_t$: Dimensionless Radial and tangential load components, respectively
U : Journal linear speed ($U = \omega R$), (m/s)
x, y, z Coordinates system (m)
$\bar{z}$: Dimensionless axial coordinates
φ : Fractional film content in the inactive zone; non-dimension density in the active zone
β : Oil bulk modulus (Pa.)
ε : Eccentricity ratio, $\varepsilon = e/C$
λ : Power-law index
κ : Elastic coefficient
θ : Circumferential coordinate, x/R, (deg.)
ν : Poisson's ratio
μ : Viscosity of the lubricant (pa. s)
μ_o : reference lubricant viscosity, (Pa.s)
ρ : Lubricant density (kg/m^3)
ϕ : Attitude angle (deg.)
ω : Angular velocity of the journal, (rad/s)

Appendix –I: Flow chart of the general computational procedure