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Influence of couple stress lubricants on hole-entry hybrid journal bearings

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HIGHLIGHTS

- *Minimum fluid film thickness increases for the bearings operate under couple stress lubricants.*
- *Stiffness and damping coefficients are larger for the couple stress fluids lubricated bearings.*
- *Threshold speed improves when the bearings are lubricated by couple stress lubricants.*

ABSTRACT

The analysis of symmetric and asymmetric hole-entry hybrid journal bearings under couple stress lubrication is presented in the present investigation. Equation of Reynolds theorem has been modified for the couple stress lubricant's flow in the journal and bearing's clearance space and Newton Raphson and finite element techniques are used for solving this Reynolds equation. For different parameter values of couple stress lubricant and for external load, the outcomes are simulated. Presented result show that at constant load for hybrid journal bearings, couple stress lubricant influences the increase in the value of minimum thickness of fluid film, coefficients of stiffness and damping. It is also observed that for asymmetric bearing under couple stress lubricant, the threshold speed value increases by 14.47% for lower external load when operates couple stress parameter value 5 than similar bearing under Newtonian lubricant.

Keywords:
| Fluid film bearings | Lubricants | Restrictors | Finite element technique |

NOMENCLATURE

a_b	Width of bearing land
a_O	Diameter of orifice, mm
c	Radial clearance
$\bar{C}_{ij}$	Damping coefficients, $C_{ij}\left(c^3/p_s R_j^4\right)$
D	Diameter of the bearing
e	Eccentricity
F_o	Fluid-film reaction
$\overline{F}_o$	Fluid-film reaction, $\overline{F}_o/p_s R_j^2$
h	Fluid film thickness
h_{min}	Minimum thickness of fluid film, mm
$\overline{h}$	h/c
L	Bearing length
l	Characteristics length of additives
$\bar{l}$	Couple stress parameter, c/l
p	Pressure
p_s	Supply pressure, Pa
Q	Flow of lubricant in bearing
$\overline{Q}$	$Q\ (\mu/c^3 p_s)$
R_j	Radius of journal
$\bar{S}_{ij}$	Stiffness coefficients$S_{ij}\left(c/p_s R_j^2\right)$
$\overline{C}_{s2}$	Restrictor design parameter for orifice restrictor, $\left[\frac{1}{12}\left(\frac{3\pi a_o^2 \mu_r \psi_d}{c^3}\right)\right]\left(\frac{2}{\rho\rho_s}\right)^{1/2}$
W_o	External load
$\overline{W}_o$	$W_o/p_s R_j^2$
X_J, Z_J	Journal centre coordinate
$\overline{X}_J$	X_J/c
$\overline{Z}_J$	Z_J/c
α	X/R_J, circumferential coordinates
β	Y/R_J, axial coordinates
β^*	p^*/p_s, concentric design pressure ratio
ε	Eccentricity ratio
η	Material coefficient responsible for couple stress property
μ	Dynamic viscosity of lubricant
λ	L/D, aspect ratio
Ω	$\omega_j\left(\mu R_j^2/c^3 p_s\right)$, speed parameter
ψ_d	Coefficient of discharge for orifice
ω_{th}	Threshold speed, rad s^{-1}

1.0 INTRODUCTION

The theory of couple stress fluids considers the fluid particle's size and explains the behaviour of the lubricants containing long chain polymer molecules. It was proposed by Stokes (1966). Ariman and Cakmak (1967) describes the applications of theory of couple stress to the problems of couettee flow and poiseuille flows between two parallel plates and also compared the results given by Stokes. Schijve (1966) discussed the physical significance of couple stresses. Chaturani (1977) worked on Poiseulle and pulsatile flows of couple stress fluid for the application of blood flow. Later on, Bhattacharjee and Das (1994) presented the characteristics of porous bearing using couple stress and Newtonian lubricants. Ramanaiah and Sarkar (1978) studied theoretically the squeeze film bearings lubricated with couple stress fluids. Lin and Yu (2004) presented the effects of couple stresses on steady state performance of slider bearings having wide parabolic shaped. They found that the effects of couple stresses characterized by the couple stress parameter signify an improvement in the steady state performance. Findings show that the parabolic shaped slider bearing under couple stress fluids results in a higher value of load carrying capacity and a smaller required volume flow rate especially for the bearing designed at a larger value of the profile parameter as compared to the inclined plane bearing. Alyaqout and Elsharkawy (2011) used SQP to optimize the slider bearing's shape lubricated with couple stress lubricant utilizing COMSOL software. Sinha et al. (1981) reported the reduction in coefficient of friction value due to use of couple stress lubricant. Later on, Lin (2001) found an increase in fluid film stiffness and damping characteristics as compared to Newtonian lubricants. Wang et al. (2002) considered both thermal and cavitation effects for the analysis of performance of journal bearing lubricated with couple stress. It was shown that an increase in load carrying capacity, decrease in friction coefficient and also formation of lower bearing temperature field is achieved when using lubricant with couple stress as compared to Newtonian lubricant. Ma et al. (2004) investigated the dynamic performance of bearings operate under couple stress fluids. It was concluded that the couple stress fluids lubrication improves the bearing performance under dynamic loads. Further, it is reported that the lubricants with couple stresses increase the fluid film pressure and the attitude angle as well as reduce the friction coefficient as compared with Newtonian lubricant. Guha (2004) reported the performance of hydrodynamic bearing under couple stress fluid. From the out coming results it was suggested that there is an increase in dynamic performance of journal bearing system due to the influence of couple stress. Liao et al. (2005) reported that the stiffness and damping coefficients of bearing increases under couple stress fluid at eccentricity ratio values. The inertia of fluid and recess volume fluid compressibility on hydrostatic step thrust bearing was presented by Lin (1999). The characteristics of finite bearings using couple stress lubricant were presented by Mokhiamer et al. (1999) under elastic deformation. It was found that in the presence of couple stresses as the elasticity effect reduces there is an increase in the load carrying capacity and the friction factor is higher for the bearing. Lahmar and Bou-Said (2008) analyzed connecting rod's big end bearings in diesel and petrol engines under couple stress fluids. It was found that the couple stress effects generate more oil film thickness. Later on, Jian et al. (2010) proposed the hybrid squeeze-film bearing with active control lubricated with couple stress fluid. The pressure distribution and the dynamics of a rigid rotor supported by such bearing were studied. Mouassa et al. (2015) analyzed the static performance of compliant bearing under piezoviscous polar fluid as lubricant. Further, the several studies concerning the performance of hole-entry hydrostatic and hybrid journal bearings have been carried out (Rowe et al., 1982; Cheng and Rowe, 1995; Sharma et al., 1990; Ram, 2016a; Ram, 2016b; Sharma and Ram, 2012). The performance characteristic of hole-entry journal bearing was analyzed by Rowe, et al. (1982). Its performance for various values of power ratio was compared to recessed and slot-entry bearings. Later on, a selection strategy was reported to design the hydrostatic bearings by Cheng and Rowe (1995). Sharma et al. (1990) reported the effects of orifice restrictor on hole-entry journal bearing considering deformation coefficients. Recently, Ram (2016a; 2016b) reported the influence of miropolar and turbulent lubrication on symmetric/asymmetric hole-entry bearings. Later on, Sharma and Ram (2012) determined the influence of orifice restrictor and couple stress lubricants on hole-entry bearings. Recently, Naduvinamani et al. (2017) presented the effect of magnetic field and couple stress lubricant on slider and parabolic bearings. They observed the superior performance of parabolic bearing than the slider bearing.

A thorough evaluation of the literature pertaining to the hydrodynamic journal bearings indicates that the performance of these bearings is significantly affected by the non-Newtonian behavior of couple stress lubricant. Further, it has been observed that many studies are concerned with the effect of couple stress lubricant exist for mainly the hydrodynamic journal bearings. Therefore, the hole-entry hybrid journal bearing under couple stress lubrication has been analyzed in the present paper. The work is required in the area of hole-entry hybrid journal bearings under couple stress lubricant to provide the needful information to the designer of bearings.

2.0 ANALYSIS

The Reynolds equation for couple stress lubricant has been modified for hole-entry journal bearing is given as:

$$\frac{\partial}{\partial \alpha}\left(\frac{\bar{h}^3}{12}\bar{\Phi}(\bar{l},\bar{h})\frac{\partial \bar{p}}{\partial \alpha}\right)+\frac{\partial}{\partial \beta}\left(\frac{\bar{h}^3}{12}\bar{\Phi}(\bar{l},\bar{h})\frac{\partial \bar{p}}{\partial \beta}\right)=\frac{\Omega}{2}\frac{\partial \bar{h}}{\partial \alpha}+\frac{\partial \bar{h}}{\partial \bar{t}} \tag{1}$$

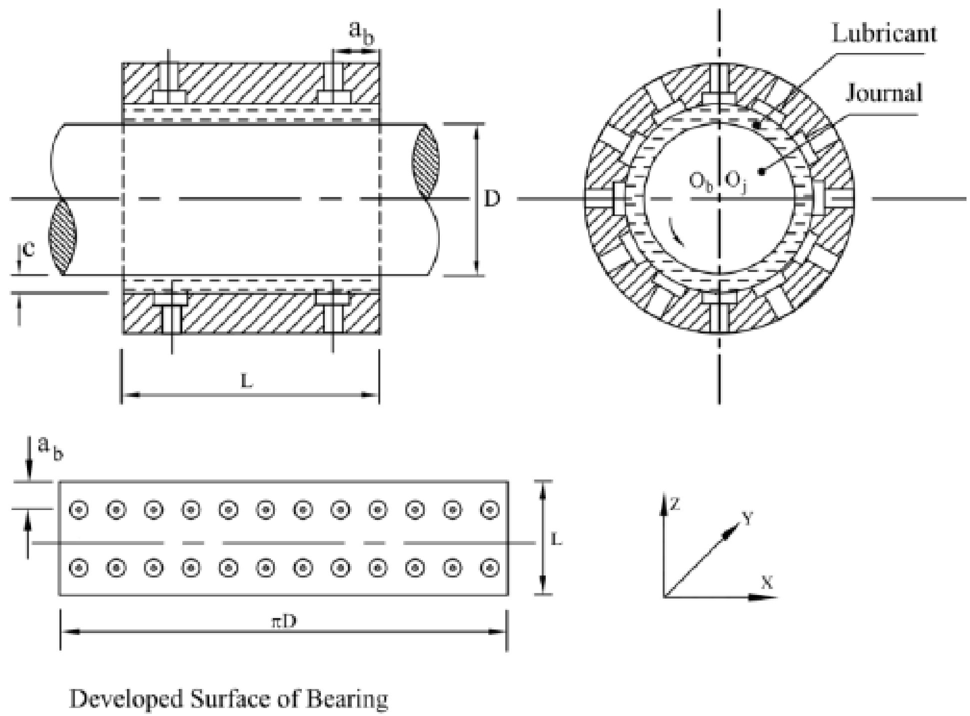
Where, $\bar{\Phi}(\bar{l},\bar{h}) = 1-\frac{12}{\bar{l}^2\bar{h}^2}+\frac{24}{\bar{l}^3\bar{h}^3}tanh\left(\frac{\bar{h}\bar{l}}{2}\right)$

$\bar{l}$ is the parameter for couple stress lubricant used in bearing and different from Newtonian one. As the $\bar{l}$ decreases the couple stress increases and lubricant behaves as a Newtonian lubricant when the $\bar{l}$ reaches to infinity.

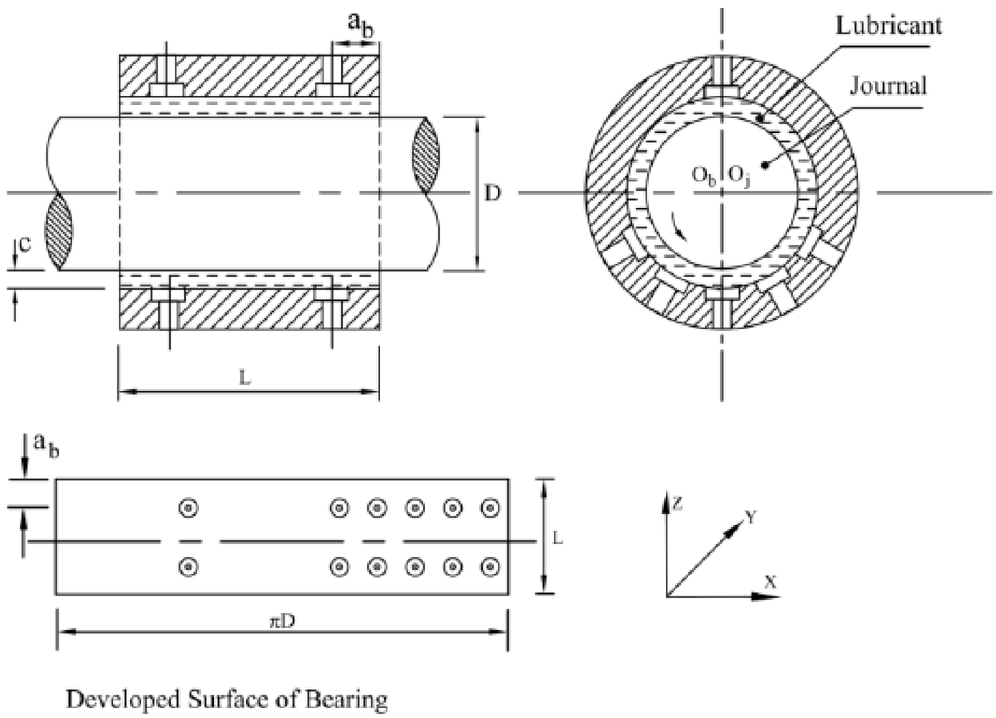
2.1 Fluid Film Thickness

The fluid film thickness $\bar{h}$ depends on journal centre's coordinates X_J and Z_J, shown in Figure 1, is given as (Sharma et al., 1990; Ram, 2016b):

$$\bar{h} = 1 - \bar{X}_J \cos\alpha - \bar{Z}_J \ sin\alpha \tag{2}$$



(a)



(b)

Figure 1: (a) Symmetric hole-entry journal bearing and (b) asymmetric hole-entry journal bearing

2.2 Restrictor Flow Equation

The orifice restrictor has been used in hole-entry bearing and lubricant flow through this restrictor is given as (Sharma et al., 1990):

$$\overline{Q}_R = \overline{C}_{s2}\left(1 - \overline{p}_c\right)^{1/2} \tag{3}$$

2.3 Finite Element Formulation

Four noded quadrilateral iso-parametric elements has been used for the discretization of lubricant domain and the equations are derived in the matrix form with the help of Galerkin's technique as:

$$[\bar{F}]^e\,\{\overline{p}\}^e = \{\bar{Q}\}^e + \Omega\,\{\bar{R}_H\}^e + \dot{\bar{X}}_j\{\overline{R}_{xj}\}^e + \dot{\bar{Z}}_j\{\overline{R}_{zj}\}^e \tag{4}$$

Where,

$$\overline{F}_{ij}^e = \int_{A^e}\int\left[\frac{\bar{h}^3}{12\,\bar{\mu}}\,\overline{\Phi}(\bar{l}\,,\bar{h})\,\frac{\partial N_i}{\partial\alpha}\,\frac{\partial N_j}{\partial\alpha} + \frac{\bar{h}^3}{12\bar{\mu}}\,\overline{\Phi}(\bar{l}\,,\bar{h})\,\frac{\partial N_i}{\partial\beta}\,\frac{\partial N_j}{\partial\beta}\right]d\alpha d\beta \tag{5}$$

$$\overline{Q}_i^e = \int_{\Gamma^e} \left\{ \left[\left(\frac{\overline{h}^3}{12\,\overline{\mu}} \overline{\Phi}(\overline{l}\,,\overline{h}) \frac{\partial \overline{p}}{\partial \alpha} \right) - \frac{\Omega}{2} \overline{h} \right] l + \left(\frac{\overline{h}^3}{12\,\overline{\mu}} \overline{\Phi}(\overline{l}\,,\overline{h}) \frac{\partial \overline{p}}{\partial \beta} \right) m \right\} N_i d\Gamma \qquad (6)$$

$$\overline{R}_{Hi}^e = \int_{A^e} \int \frac{\overline{h}}{2} \frac{\partial N_i}{\partial \alpha} \, d\alpha d\beta \qquad (7)$$

$$\overline{R}_{xji}^e = \int_{A^e} \int N_i \, cos\alpha \;\; d\alpha d\beta \qquad (8)$$

$$\overline{R}_{Zji}^e = \int_{A^e} \int N_i \, sin\alpha \;\; d\alpha d\beta \qquad (9)$$

2.4 Boundary Conditions

Necessary assumptions for solving Equation (4) are:

1. Zero relative pressure on the nodes are situated on the outer boundary of bearing
   i.e. $\bar{p}|_{\beta=\mp 1.0} = 0.0$.
2. Nodes situated on holes have equal pressure.

3.0 SOLUTION PROCEDURE

An efficient and suitable iterative procedure is employed for determining the performance of bearings using couple stress lubrication and compensated by an orifice restrictor. For steady state, the lubricant flow field system Equation (4) along with an orifice restrictor flow Equation (3) satisfying the required boundary conditions has been solved for obtaining the fluid film pressure field. Modified system of equations for a lubricant flow becomes non-linear, when the bearings are compensated by an orifice restrictor. Therefore, for solving these equations of non-linear type, a Newton Raphson method has been used in the present work. On the convergence of the solution, the performance of hybrid bearings has been computed using the expressions from Ram (2016b).

4.0 RESULTS AND DISCUSSION

To determine the bearing performance characteristics, a computer source code in Fortran 77 has been developed. For the validation, the computed results are correlated by the results of Wang et al. (2002) and these are shown in Figure 2. The results are computed for the parameters of bearings as revealed in Table 1 for the couple stress parameter values $\bar{l} = 5, 10, 15$. In the present work, the influence of couple stress lubricant on the performance of journal bearings is studied theoretically by defining the non-dimensional parameter of couple stress lubricant $\left(\bar{l} = c/l\right)$, where l has the dimension of length and is a property of lubricant. It has a dependence on the polar additive's molecule size. The effect of couple stress lubricant on the static and dynamic performance characteristics of an orifice compensated hole-entry hybrid journal bearing system is presented in this work. The values of bearing's characteristics due to couple stress lubricant have been compared with Newtonian fluid lubricated bearing in each figure.

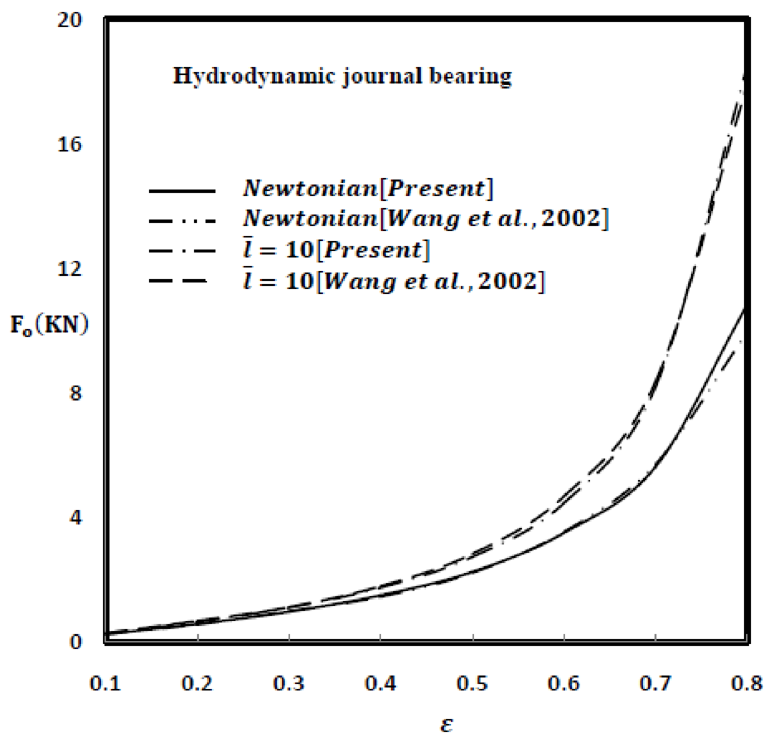


Figure 2: Variation of F_o with ε

Table 1: Parameter of bearings under couple stress and Newtonian lubricants

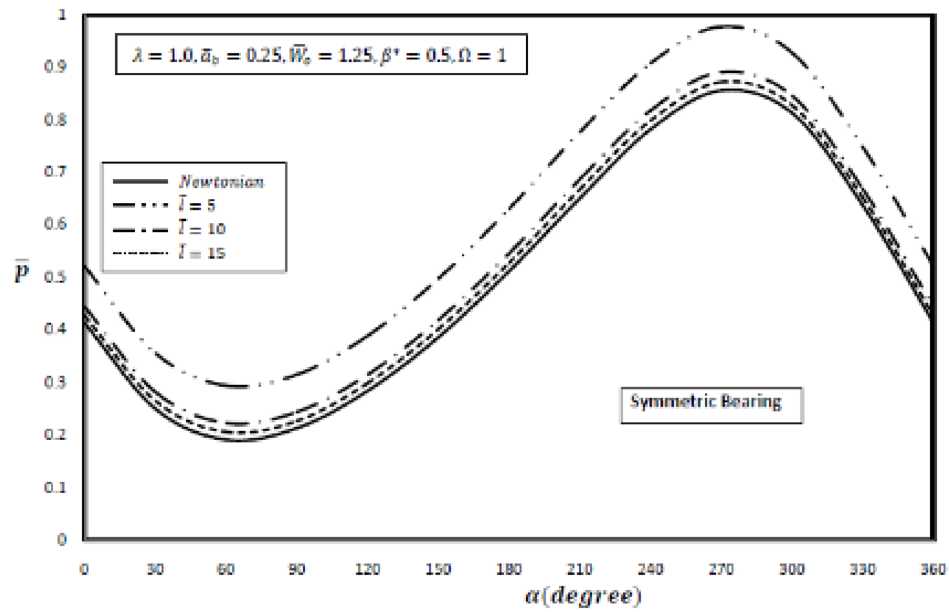
Parameters	Values
Bearing aspect ratio (λ)	1.0
Land width ratio ($\bar{a}_b$)	0.25
External load ($\bar{W}_o$)	0.25 – 1.25 for symmetric bearing 1.0 – 2.0 for asymmetric bearing
No. of rows of holes	2
No. of holes per row	6 for symmetric bearing 12 for asymmetric bearing
Concentric design pressure ratio (β^*)	0. 5
Speed parameter (Ω)	1.0
Type of restrictor	Orifice

4.1 Distribution of Fluid-Film Pressure $(\bar{p})$

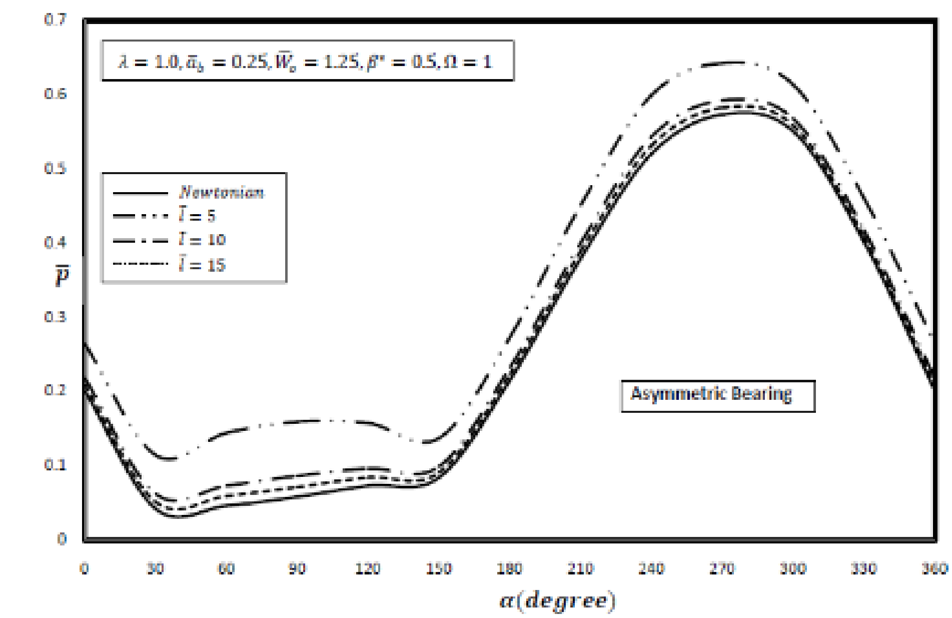
The effect of couple stress lubricant on the distribution of fluid-film pressure in circumferential direction on mid plane of bearing is revealed in Figure 3(a) and 3(b). Observations shown in Figure 3(a) and 3(b) that as the value of couple stress parameter $\bar{l}$ decreases, the value of fluid film pressure distribution increases for the entire profile of the symmetric and asymmetric bearings than the bearings under Newtonian fluid. It can be concluded that the additives with larger chain length molecules, enhance the load carrying capacity of bearings. The distribution of fluid film pressure for asymmetric bearing is shown in Figure 3(b). It may be noticed that fluid film pressure distribution increase due to the influence of couple stress lubricant when compared with same bearing operating under Newtonian lubricant. For a lower half of the asymmetric bearing (i.e. between 180^o to 360^o), the trend in the variation of the fluid-film pressure distribution is almost similar to that observed in a symmetric bearing.

4.2 Minimum Fluid Film Thickness $\left(\bar{h}_{min}\right)$

A decreasing trend for minimum fluid film thickness $\left(\bar{h}_{min}\right)$ with load $(\bar{W}_o)$ has been given in Figure 4(a) and (b) for symmetric and asymmetric bearings. It has been observed that the fluid film thickness reduces for both the bearings, operate under couple stress and Newtonian lubricants. However, an increase in fluid film thickness value is noted for the bearing operates under couple stress lubricant. It may be noted that the $\bar{h}_{min}$ value is higher at constant value of $\bar{W}_o$ when bearing operates with couple stress fluid than Newtonian fluid. Therefore, a bearing with couple stress lubricant can sustain higher value of external loads and permits the larger value of $\bar{h}_{min}$. Further, an order of maximum 5.16% increase is found in the value of $\bar{h}_{min}$ at $\bar{W}_o = 1.5$ for asymmetric hybrid bearing corresponding to couple stress parameter $\bar{l} = 5$ as compared to the bearing lubricated with Newtonian fluid.



3(a)



3(b)

Figure 3 (a) and (b): Circumferential fluid film pressure distribution at the axial mid-plane at $\beta = 0.0$

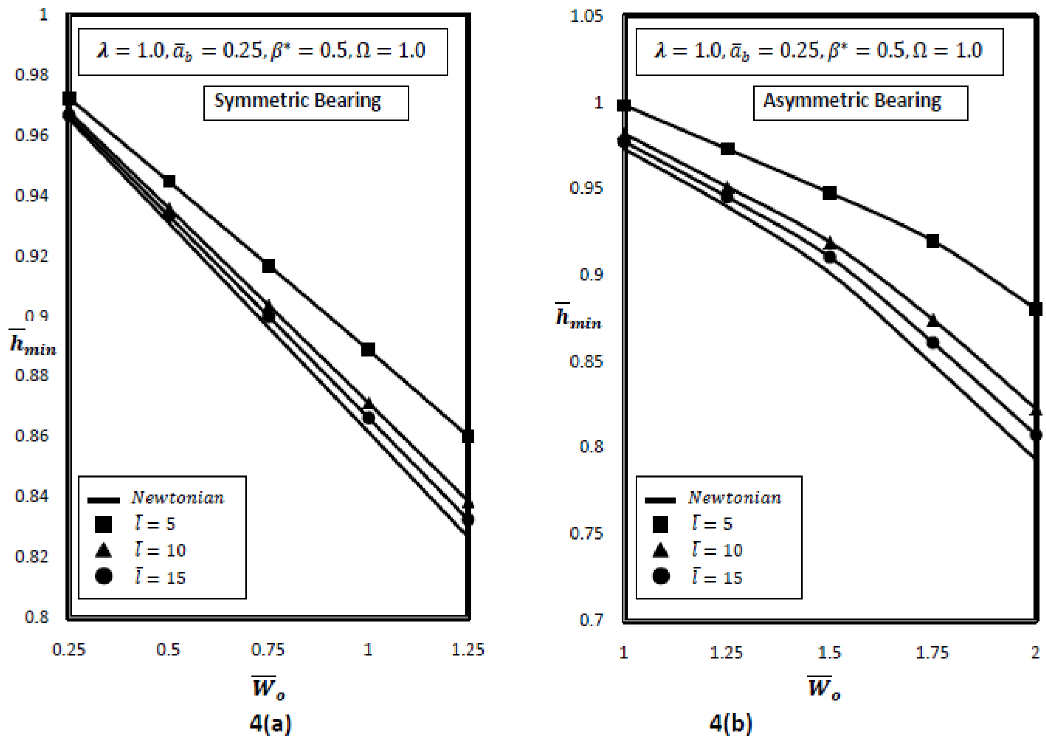


Figure 4(a) and (b): Variation of minimum fluid film thickness $\bar{h}_{min}$ with $\bar{W}_o$

4.3 Bearing Lubricant Flow($\bar{\boldsymbol{Q}}$)

Figure 5(a) and 5(b) illustrate the variation of lubricant flow ($\bar{Q}$) for various values of couple stress parameter. It may be observed that the bearings operate under couple stress and Newtonian lubricants, as the external load($\bar{W}_o$) increases, the lubricant's flow ($\bar{Q}$) demand decreases. Despite that, observations also show that there is more reduction in lubricant flow ($\bar{Q}$) value for constant load ($\bar{W}_o$) for symmetric and asymmetric hybrid bearing operating with couple stress lubricant when compared with Newtonian lubricant. Further observations show that for a couple stress lubricants bearing the $\bar{Q}$'s value reduces as couple stress parameter $\bar{l}$ reduces keeping the concentric design pressure ratio fixed. For symmetric and asymmetric hybrid bearing, a reduction in $\bar{Q}$ value is found to be of order of 12.93% at $\bar{W}_o = 1.0$ and 8.72% at $\bar{W}_o = 1.5$ respectively corresponding to couple stress parameter at $\bar{l} = 5$ as compared to the bearing lubricated with Newtonian fluid.

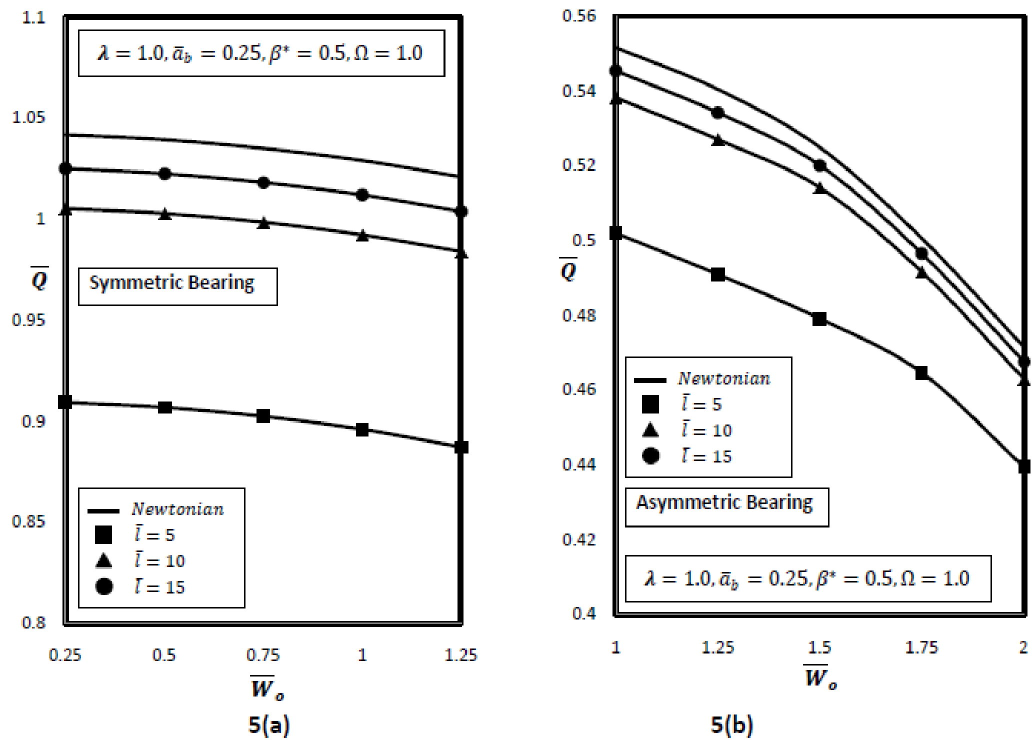


Figure 5 (a) and (b): Variation of bearing flow $\bar{Q}$ with $\bar{W}_o$

4.4 Fluid-Film Stiffness Coefficients($\bar{S}_{11}, \bar{S}_{22}$)

The effect of couple stress lubricant on the bearing's stiffness coefficients is shown in Figures 6 and 7. It has been observed from Figure 6 and Figure 7 that, the value of stiffness coefficient ($\bar{S}_{11}$) for symmetric and asymmetric bearing decreases for both the couple stress and Newtonian lubricants. However, for constant external load ($\bar{W}_o$), the value of $\bar{S}_{11}$ is found to increase when the bearings are lubricated with couple stress fluid. At a chosen value of $\beta^* = 0.5$, an increase in the value of $\bar{S}_{11}$ is observed 16.86% at $\bar{W}_o = 1.0$ and 30.14% $\bar{W}_o = 1.5$ for symmetric and asymmetric bearings respectively for couple stress parameter $\bar{l} = 5$ as than Newtonian fluid. Figure 7(a) and (b) indicate that the stiffness coefficient ($\bar{S}_{22}$) value is large at lesser value of load when the bearing is lubricated with couple stress lubricant in comparison to Newtonian fluid. At couple stress parameter $\bar{l} = 5$, the stiffness coefficient ($\bar{S}_{22}$) is 21.38% for $\bar{W}_o = 1.5$ for asymmetric bearing in comparison to the bearing under Newtonian fluid. As the value of parameter $\bar{l}$ decreases, the stiffness coefficient ($\bar{S}_{22}$) increases for symmetric bearing than Newtonian lubricant as shown in Figure 7(b). It is noted that there exists a particular value of load $\bar{W}_o$ for which the stiffness $\bar{S}_{22}$ decreases and after that the value of $\bar{S}_{22}$ increases for asymmetric bearing lubricated with couple stress lubricant.

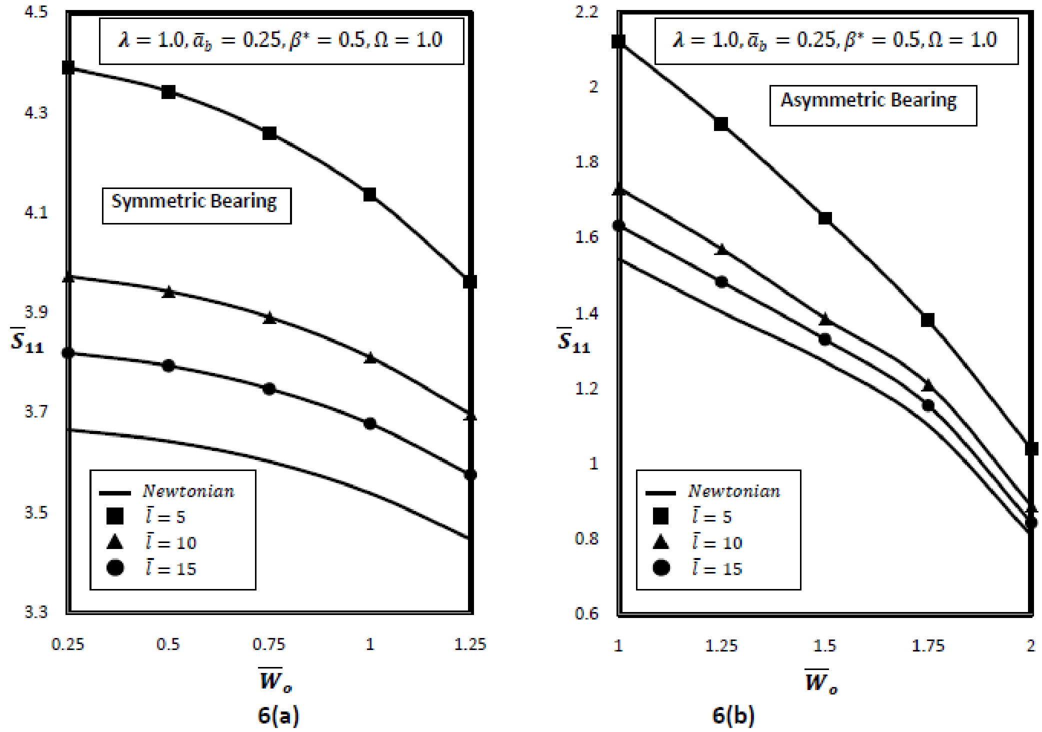


Figure 6 (a) and (b): Variation of direct fluid film stiffness coefficient $\bar{S}_{11}$ with $\bar{W}_o$

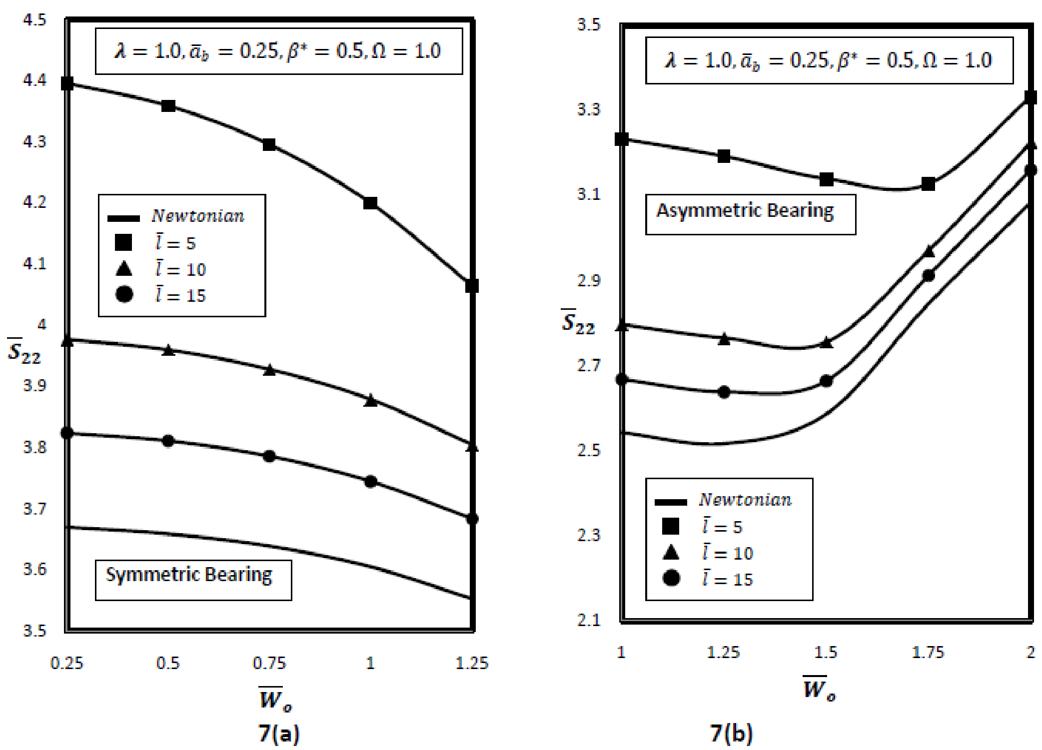


Figure 7 (a) and (b): Variation of direct fluid film stiffness coefficient $\bar{S}_{22}$ with $\bar{W}_o$

4.5 Fluid-Film Damping Coefficients($\bar{C}_{11}, \bar{C}_{22}$)

Variation of damping coefficients ($\bar{C}_{11}, \bar{C}_{22}$) for couple stress lubricated bearings is shown in Figures 8 and 9. From Figure 8(a) and 8(b), it is found that the damping coefficient ($\bar{C}_{11}$) gets increased by reducing the value of couple stress parameter $(\bar{l})$ at every load for the both bearings. The value of damping coefficient ($\bar{C}_{11}$) is 23.13% for $\bar{W}_o = 1.0$ and 33.94% for $\bar{W}_o = 1.5$ at couple stress parameter $\bar{l} = 5$ when symmetric and asymmetric bearings respectively operate under couple stress lubricant. The similar observation can be made from Figure 9(a) and 9(b) for damping coefficient $\bar{C}_{22}$ at constant load for the bearings. The coefficient ($\bar{C}_{22}$) is found 31.18% at $\bar{W}_o = 1.5$ for couple stress lubricant having $\bar{l} = 5$ for asymmetric bearing than Newtonian lubricant.

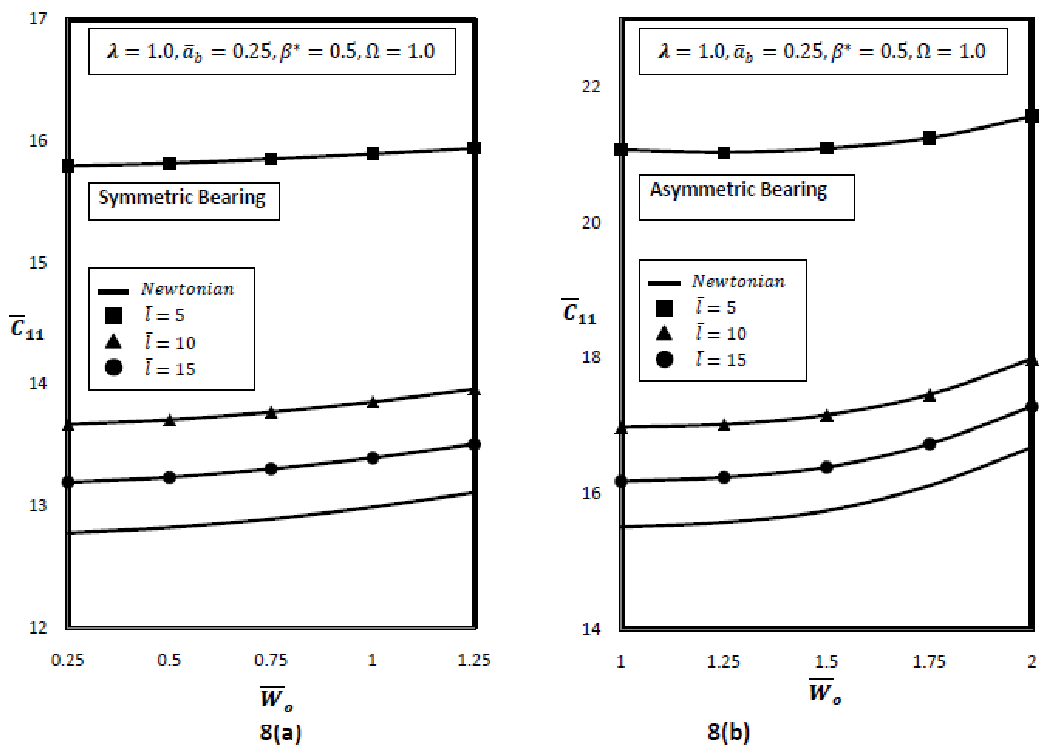


Figure 8 (a) and (b): Variation of direct fluid film damping coefficient $\bar{C}_{11}$ with $\bar{W}_o$

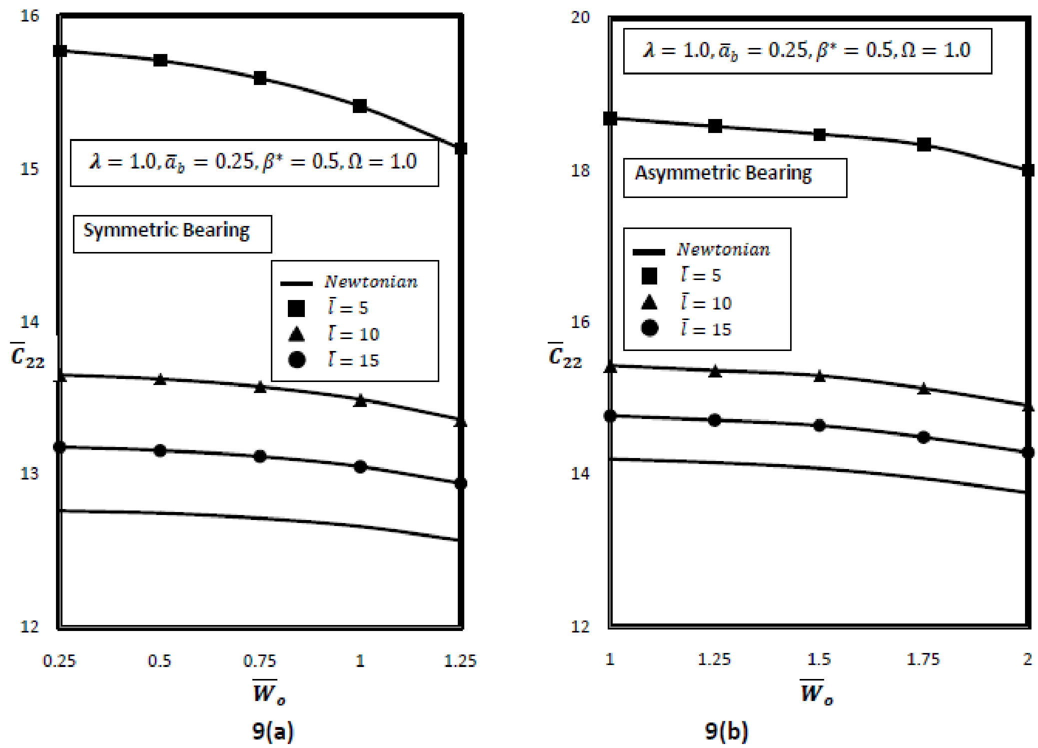


Figure 9 (a) and (b): Variation of direct fluid film damping coefficient $\bar{C}_{22}$ with $\bar{W}_o$

4.6 Stability Threshold Speed Margin ($\bar{\omega}_{th}$)

The variation in the value of stability threshold speed $\bar{\omega}_{th}$ with external load $\bar{W}_o$ for different values of $\bar{l}$ is shown in Figure 10. The value of $\bar{\omega}_{th}$ is found to increase with a reduction in the value of couple stress parameter $\bar{l}$ for constant value of external load as compared to Newtonian lubricant. The stability threshold speed margin $\bar{\omega}_{th}$ depends on the values of bearing's stiffness and damping coefficients. As the values of bearing rotor dynamic coefficients get affected due to couple stress parameter $\bar{l}$. A maximum increase of 7.83% at $\bar{W}_o = 1.0$ in the value of $\bar{\omega}_{th}$ is observed for the couple stress parameter $\bar{l} = 5$ for asymmetric bearing as compared to Newtonian lubricant.

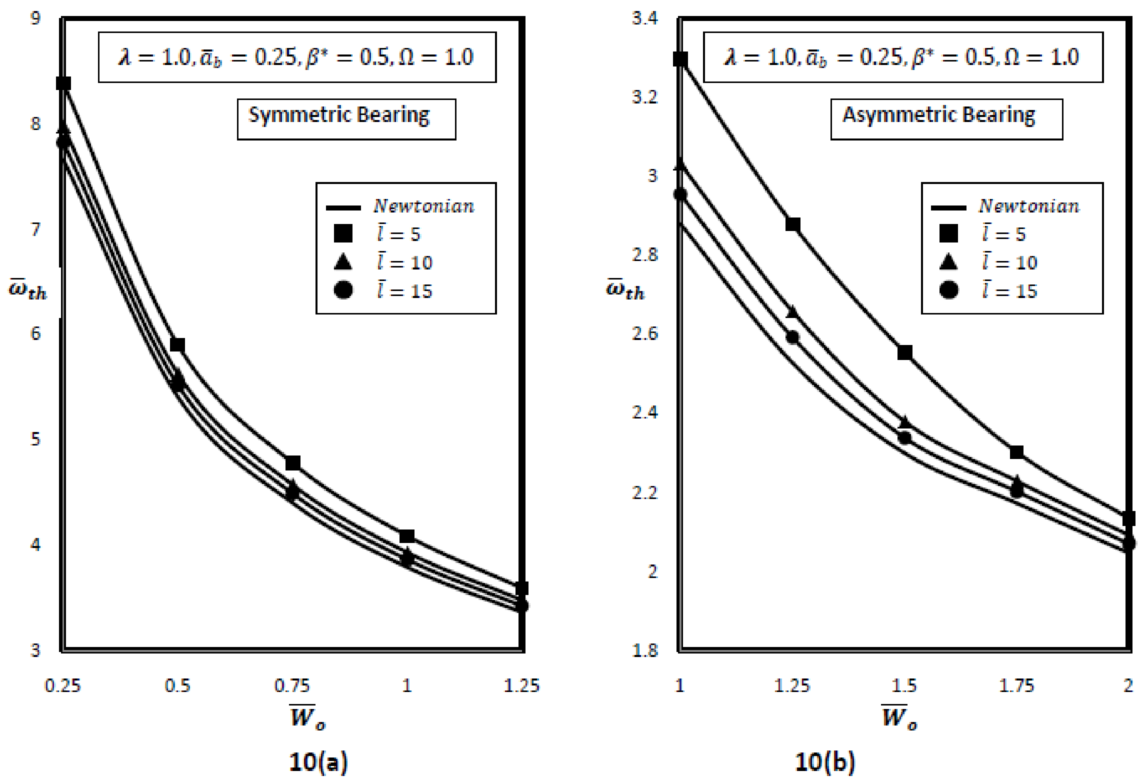


Figure 10 (a) and (b): Variation of stability threshold speed margin $\bar{\omega}_{th}$ with $\bar{W}_o$

CONCLUSION

The conclusions from computed results are:

1. The minimum fluid film thickness for asymmetric bearing is more in comparison to symmetric bearings when operates under couple stress lubricant.
2. The value of bearing flow decreases in case of symmetric bearing operates under couple stress lubricant than asymmetric bearing.
3. At couple stress parameter $\bar{l} = 5$, the values of stiffness coefficients are larger for the bearing lubricated under couple stress lubricant as compared to Newtonian lubricant.
4. The damping coefficients increases significantly for couple stress fluid lubricated bearing as compared to similar bearing under Newtonian lubricant.
5. The threshold speed $\bar{\omega}_{th}$ is obtained 11.08% at $\bar{W}_o = 1.5$ for $\bar{l} = 5$ when the asymmetric bearing lubricated under couple stress lubricant than similar bearing under Newtonian lubricant

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