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Stability analysis of slot-entry hybrid journal bearings operating with non-Newtonian lubricant

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HIGHLIGHTS

- *Analysis of rheological effects of lubricant on stability parameters of slot-entry hybrid journal bearings system has been carried out.*
- *The stability parameters have been computed of different configurations of slot-entry hybrid journal bearing operating with Newtonian and Non-Newtonian lubricants for wide range of external load.*
- *The variation of viscosity due to non-Newtonian behavior of lubricant affect bearing stability quite significantly*
- *The single row asymmetric slot-entry hybrid journal bearing system has been identified as best configuration from stability point of view.*

ABSTRACT

This paper presents theoretical investigations of rheological effects of lubricant on stability parameters of various configurations of slot-entry hybrid journal bearing system. FEM has been used to solve Reynolds equation governing flow of lubricant in bearing clearance space along with restrictor flow equation using suitable iterative technique. The non-Newtonian lubricant has been assumed to follow cubic shear stress law. The stability parameters in terms of stiffness coefficients, damping coefficients, threshold speed and whirl frequency of different configurations of slot-entry hybrid journal bearing have been computed and presented for wide range of external load while operating with Newtonian and Non-Newtonian lubricants. The computed results reveal that variation of viscosity due to non-Newtonian behavior of lubricant affects bearing stability quite significantly. The results are presented in graphical form and logical conclusions are drawn to identify best possible configuration from stability point of view.

Keywords:
| Slot-entry | Journal bearing | Hybrid | Non-Newtonian lubricant | Stability parameters |

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1.0 INTRODUCTION

The slot-entry hybrid journal bearings have been successfully used in various engineering applications because of their good performance over wide range of speed and load, besides their relative simplicity in manufacturing. The continual growth of technological advances and industry's expanding demands for higher speed applications and ability of hydrostatic/hybrid journal bearings to support heavy loads have necessitated to study performance of these bearings in detail under more realistic conditions. In slot-entry type restrictors are formed by a slotted shim fabricated into bearing. The slot-entry bearing need finer filtration to prevent gradual silting of slot restrictors.

(Shires and Dee, 1967) proposed idea of slot-entry bearing originally for a purely hydrostatic bearing application. Slot-entry bearings are originally developed by Dee and described by (Dee and Shires, 1971). (Rowe et al., 1976) showed that when hole and slot-entry journal bearings are compared with recessed hydrostatic and hydrodynamic bearings on load basis alone, slot-entry journal bearing has a greatly superior performance to recessed hydrostatic bearing for a given supply pressure. The advantages of slot-entry journal bearing over axial groove hydrodynamic bearings were also demonstrated. From studies and initial experimental results (Rowe and Koshal, 1977) found that there are areas of operation where slot-feed journal bearings have considerable advantages over recessed bearings. Slot-entry bearings are more suitable for heavily loaded conditions including high dynamic loading. (Ives and Rowe, 1987) theoretically analyzed slot-entry bearings of various symmetrical and asymmetrical inlet port configurations by finite difference method. (Cheng and Rowe, 1995) presented a computerized selection method for externally pressurized journal bearings including slot-entry journal bearing with respect to bearing type, configuration, fluid feeding device, bearing material and production techniques.

A study by (Sharma et al., 1999) demonstrated that a slot-entry journal bearing configuration may provide enhanced performance compared with other non-recessed bearing configurations. It has been studied by (Sharma et al., 2000) that combined effect of non-linear behavior of lubricant and bearing flexibility alters bearing performance characteristics of a slot entry hybrid journal bearing. The value of minimum film thickness is reduced appreciably due to this combined effect. The maximum decrease in minimum film thickness is found to be of the order of 51.41% with respect to rigid bearing operating with Newtonian lubricant.

(Sharma et al., 2002) developed a theoretical model to take into account the influence of thermal effects in slot-entry hybrid journal bearing by considering variation of viscosity due to temperature rise of lubricant. It has been found out that consideration of thermal effects lowers value of minimum fluid film thickness for both symmetric as well as asymmetric configurations while lubricant supply requirement of a slot-entry hybrid journal bearing system is found to increase. At a constant value of external load, enhancement in bearing flow is found to be of the order of 11% and 4% for symmetric and asymmetric bearing configurations respectively.

(Garg et al., 2006) presented a comprehensive review of developments in design and application of hydrostatic and hybrid journal bearing systems and concluded that more extensive research is needed, both analytically as well as experimental, to consider extension of these bearings into high speed applications. (Duvedi et al., 2006) carried out the theoretical analysis of capillary compensated non-recessed hole-entry hybrid journal bearing lubricated with non-Newtonian lubricant. (Garg et al., 2007) found that change in viscosity of lubricant due to non-Newtonian behavior and rise of temperature affects performance of hole-entry hybrid journal bearing system quite significantly. (Garg et al., 2010a) performed the analysis of hole–entry hybrid journal bearing with capillary valve restrictor by considering the combined effect of temperature increase and non-Newtonian behavior of the lubricant. (Garg et al., 2010b) theoretically investigated thermal and rheological effects of lubricant on performance of symmetric and asymmetric slot-entry hybrid journal bearing system. It was found that stability of slot-entry hybrid journal bearing (symmetric/asymmetric configuration) increases with viscosity variation due temperature rise and non-Newtonian behavior of the lubricant, while operating at higher load and low speed parameter. (Garg et al., 2010c) also investigated the effect of viscosity variation due to temperature rise and non-Newtonian behavior of the lubricant on the performance of hole-entry and slot-entry hybrid journal bearings system and found that bearing performance can be improved by selecting a particular bearing configuration in conjunction with a suitable compensating device. (Khatak and Garg, 2012) studied the applications of Eringen micropolar theory to different configurations of bearings. They showed significant performance variation in bearings with micropolar lubrication. (Garg and Kumar, 2013) found that variation of viscosity due rise in temperature and non-Newtonian behavior of the lubricant affects the performance of asymmetric hole-entry hybrid journal bearing system quite significantly. They investigated that there is an increase in the oil requirement for a hybrid journal bearing with the specified operating and geometric parameters, when the viscosity of the lubricant decreases due to the rise in temperature and non-Newtonian behavior of the lubricant. (Kushare and Sharma, 2014) studied the effect of non-Newtonian lubricant on the stability of a two lobe symmetric hole entry worn hybrid journal bearing, compensated with constant flow valve restrictor. They concluded that the proper selection of parameters such as offset factor, wear depth parameter and the non-linearity factor may provide better bearing stability. (Mehrjardi et al., 2015) presented stability performance characteristics of circular and noncircular two-, three-, and four-lobe journal bearings with micropolar fluids and found that for a constant vertical external load, the stability performance of rotating system can be improved by replacing the circular journal bearing with similar noncircular types. The review of literature presented in this section indicates that non-recessed bearings are best suited for high speed applications; especially slot-entry type bearings show superior performance over recessed bearing and hole-entry bearings. The load carrying capacity of these bearing can be still improved by selecting an optimum value of axial land width ratio, restrictor design parameter, concentric design pressure ratio etc. Most of research work pertaining to non-recessed journal bearing assumes standard symmetric and asymmetric configurations. However, many more configurations are possible by changing position of slot which may improve performance of the slot-entry journal bearing.

In present work, author has done rheological analysis of some alternative configurations of slot-entry hybrid journal bearing from stability point of view which have not been reported in literature till date. The results presented in the study are expected to be quite useful to the bearing designers.

2.0 ANALYSIS

The analysis presented in following subsection uses finite element method to model complete journal bearing system operating with non-Newtonian lubricants. The mathematical model, which includes viscosity variation due to non-Newtonian behaviour of lubricant, involves solution of Reynolds equation. The four configurations of slot-entry hybrid journal bearing have been considered for the analysis. These are referred as A1, A2, A3 and A4 configurations in rest of text in this paper. In slot-entry hybrid journal bearing, the restrictors are formed by a slotted shim fabricated into the bearing. The geometry of these four slot-entry hybrid journal bearings and their coordinate system is shown in Figures 1(a) – 1(e). The A1 is double row symmetric slot-entry hybrid journal bearing configuration having 12 holes per row. The A2 is single row symmetric slot-entry hybrid journal bearing configuration having 12 holes. The A3 is double row asymmetric slot-entry hybrid journal bearing configuration having 6 holes per row. The A4 is single row asymmetric slot-entry hybrid journal bearing configuration having 6 holes.

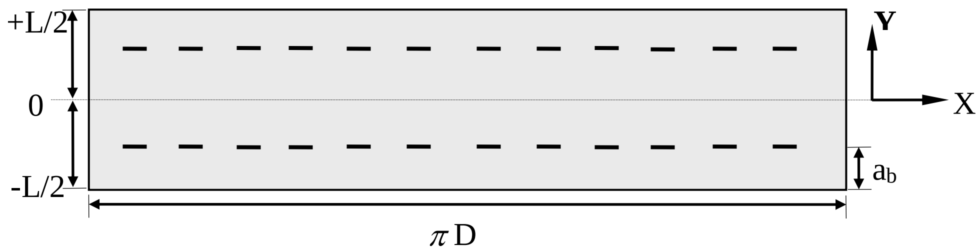
2.1 Reynolds Equation for Fluid Domain

The generalized Reynolds equation governing laminar flow of incompressible lubricant between clearance space of journal and bearing considering variable viscosity and usual assumptions in non-dimensional form is written as [Dowson (1962)]

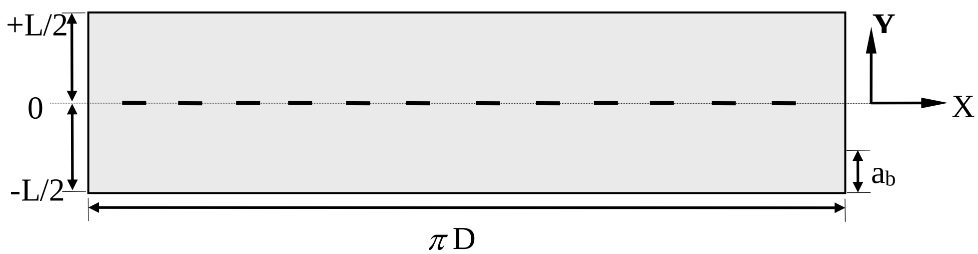
$$\frac{\partial}{\partial\alpha}\left(\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\alpha}\right)+\frac{\partial}{\partial\beta}\left(\bar{h}^3\bar{F}_2\frac{\partial\bar{p}}{\partial\beta}\right)=\Omega\left[\frac{\partial}{\partial\alpha}\left\{\left(1-\frac{\bar{F}_1}{\bar{F}_0}\right)\bar{h}\right\}\right]+\frac{\partial\bar{h}}{\partial\bar{t}} \tag{1}$$

Where $\bar{F}_0$, $\bar{F}_1$, and $\bar{F}_2$ are cross film viscosity integrals and given by following relations:

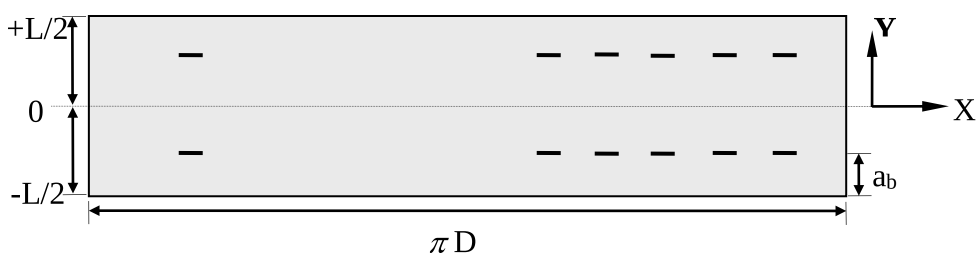
$$\bar{F}_0=\int_0^1\frac{1}{\bar{\mu}}\,d\bar{z}\,,\quad \bar{F}_1=\int_0^1\frac{\bar{z}}{\bar{\mu}}\,d\bar{z}\,,\quad \bar{F}_2=\int_0^1\frac{\bar{z}}{\bar{\mu}}\left(\bar{z}-\frac{\bar{F}_1}{\bar{F}_0}\right)d\bar{z}$$



(a)



(b)



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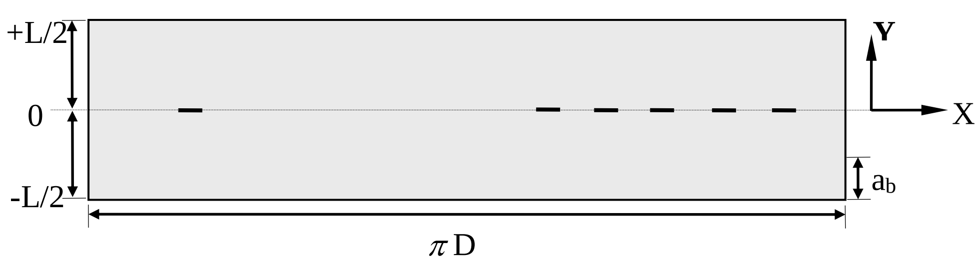

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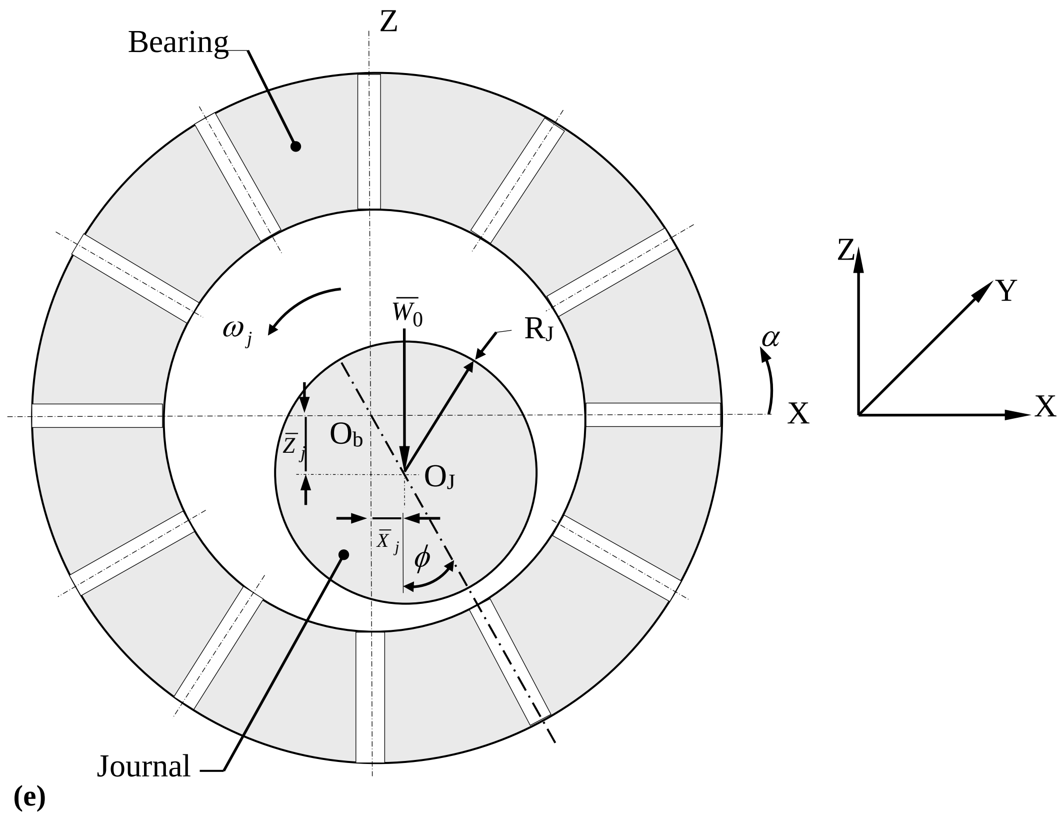


Figure 1: Slot-Entry hybrid journal bearings geometry and coordinate system, (a) Configuration A1 (b) Configuration A2 (c) Configuration A3 (d) Configuration A4 (e) Coordinate System

2.2 Finite Element Formulation

Lubricant flow field is discretized by using 4-noded quadrilateral isoparametric elements. The pressure variation is assumed to vary linearly over an element and is expressed as

$$\overline{p} = \sum_{j=1}^{n_l^e} \overline{p}_j N_j \tag{2}$$

Where N_j is an elemental shape function and n_l^e is number of nodes per element of 2D flow field discretized solution domain.

Using Galerkin's orthogonality conditions and following usual assembly procedure, global system equation is obtained as below (Garg et al., 2010):

$$\left[\overline{F}\right]\left\{\overline{P}\right\} = \left\{\overline{Q}\right\} + \Omega\left\{\overline{R}_H\right\} + \dot{\overline{X}}_J\left\{\overline{R}_x\right\} + \dot{\overline{Z}}_J\left\{\overline{R}_z\right\} \tag{3}$$

For e^{th} element these are given by:

$$\bar{F}_{ij}^{\,e} = \int_{\Omega^e} \bar{h}^3 \left[\bar{F}_2 \frac{\partial N_i}{\partial \alpha} \frac{\partial N_j}{\partial \alpha} + \bar{F}_2 \frac{\partial N_i}{\partial \beta} \frac{\partial N_j}{\partial \beta} \right] d\Omega^e$$

$$\bar{Q}_i^{\,e} = \int_{\Gamma^e} \left\{ \left(\bar{h}^3 \bar{F}_2 \frac{\partial \bar{p}}{\partial \alpha} - \Omega \left(1 - \frac{\bar{F}_1}{\bar{F}_0} \right) \bar{h} \right) l_1 + \left(\bar{h}^3 \bar{F}_2 \frac{\partial \bar{p}}{\partial \beta} \right) l_2 \right\} N_i \, d\bar{\Gamma}^e$$

$$\bar{R}_{H_i}^{e} = \int_{\Omega^e} \left(1 - \frac{\bar{F}_1}{\bar{F}_0} \right) \bar{h} \frac{\partial N_i}{\partial \alpha} d\Omega^e$$

$$\bar{R}_{x_i}^{\,e} = \int_{\Omega^e} N_i \cos\alpha \, d\Omega^e$$

$$\bar{R}_{z_i}^{\,e} = \int_{\Omega^e} N_i \sin\alpha \, d\Omega^e$$

Where l_1 and l_2 are directions cosines and i , j =1,2 … n_l^e (number of nodes per element).

2.3 Restrictor Flow Equations

The flow of lubricant through slot restrictor is expressed as (Sharma et al., 1999)):

$$\bar{Q}_{in} = \frac{1}{12\eta} \left[p_s - p \right] \frac{a_s Z_s^3}{Y_s} \tag{4}$$

The above equation is reduced to non-dimensional form as:

$$\bar{Q}_R = \bar{C}_{SR} (1 - \bar{p}_c \,) \tag{5}$$

The parameter ($\bar{C}_{SR}$) is called as slot-restrictor design parameter and is defined as:

$$\bar{C}_{SR} = \frac{\pi}{36} \frac{SWR}{\lambda} \frac{k}{\bar{a}_b} \frac{a_b}{Y_s} \left[\frac{Z_s}{c} \right]^3$$

Where $\bar{a}_b = \dfrac{a_b}{L}$ is land width ratio and k is number of rows of slots in a bearing. Slot width ratio (SWR) is ratio of actual slot width a_s to maximum possible slot width $(a_s)_{\max}$ as given below:

$$SWR = \frac{a_s}{(a_s)_{\max}} = \frac{a_s n}{\pi D}$$

2.4 Non-Newtonian Model

Most of non-Newtonian oils follow the behavior, which is represented by cubic shear law (Sharma et al., 2000). The constitutive equation for cubic shear law is described in non-dimensional form as:

$$\bar{\tau} + \bar{K}\bar{\tau}^3 = \bar{\dot{\gamma}} \tag{6}$$

Here, $\bar{K}$ is known as non-linearity factor. The viscosity of non-Newtonian lubricant is described by apparent viscosity ($\tilde{\mu}_a$) and is defined as function of shear strain ($\bar{\dot{\gamma}}$).

$$\tilde{\mu}_a = \bar{\tau} / \bar{\dot{\gamma}} \tag{7}$$

In non-dimensional form, shear strain rate ($\bar{\dot{\gamma}}$) at a point in fluid–film is function of velocity gradients $\dfrac{\partial \bar{u}}{\partial \bar{z}}$ and $\dfrac{\partial \bar{v}}{\partial \bar{z}}$, and is expressed as

$$\bar{\dot{\gamma}} = \left[\left\{ \frac{\partial \bar{u}}{\partial \bar{z}} \right\}^2 + \left\{ \frac{\partial \bar{v}}{\partial \bar{z}} \right\}^2 \right]^{\frac{1}{2}} \tag{8}$$

Where,
$$\frac{\partial \bar{u}}{\partial \bar{z}} = \left\{ \frac{\bar{h}}{\bar{\mu}} \left(\bar{z} - \frac{\bar{F}_1}{\bar{F}_0} \right) \frac{\partial \bar{p}}{\partial \alpha} + \frac{\Omega}{\bar{\mu}\bar{h}\bar{F}_0} \right\}$$
$$\frac{\partial \bar{v}}{\partial \bar{z}} = \left\{ \frac{\bar{h}}{\bar{\mu}} \left(\bar{z} - \frac{\bar{F}_1}{\bar{F}_0} \right) \frac{\partial \bar{p}}{\partial \beta} \right\}$$

2.5 Dynamic Performance Characteristics

The bearing dynamic coefficients i.e. stiffness and damping coefficient of fluid-film fall in this category. For two degree of freedom system, there exist four stiffness and four damping coefficients, which can be used to study stability of the system (Ghosh and Majumadar, 1978). These coefficients are defined as below:

2.5.1 Fluid Film Stiffness Coefficients

The fluid-film stiffness coefficients are defined as:

$$\bar{S}_{ij} = -\frac{\partial \bar{F}_i}{\partial q} (i = x, z) \tag{9}$$

Where ‘ i ’ represents direction of force and q represents direction of displacement of journal center coordinate ($\bar{X}_J$ *or* $\bar{Z}_J$).

Stiffness coefficient matrix will be

$$\begin{bmatrix} \bar{S}_{xx} & \bar{S}_{xz} \\ \bar{S}_{zx} & \bar{S}_{zz} \end{bmatrix} = -\begin{bmatrix} \dfrac{\partial \bar{F}_x}{\partial \bar{X}_J} & \dfrac{\partial \bar{F}_x}{\partial \bar{Z}_J} \\ \dfrac{\partial \bar{F}_z}{\partial \bar{X}_J} & \dfrac{\partial \bar{F}_z}{\partial \bar{Z}_J} \end{bmatrix}$$

2.5.2 Fluid Film Damping Coefficients

The fluid film damping coefficients are defined as

$$\bar{C}_{ij} = -\frac{\partial \bar{F}_i}{\partial \dot{q}} (i = x, z) \tag{10}$$

$\dot{q}$ represents the velocity component of journal center ($\dot{\bar{X}}_J$ *or* $\dot{\bar{Z}}_J$).

Damping coefficients matrix is given by:

$$\begin{bmatrix} \bar{C}_{xx} & \bar{C}_{xz} \\ \bar{C}_{zx} & \bar{C}_{zz} \end{bmatrix} = -\begin{bmatrix} \dfrac{\partial \bar{F}_x}{\partial \dot{\bar{X}}_J} & \dfrac{\partial \bar{F}_x}{\partial \dot{\bar{Z}}_J} \\ \dfrac{\partial \bar{F}_z}{\partial \dot{\bar{X}}_J} & \dfrac{\partial \bar{F}_z}{\partial \dot{\bar{Z}}_J} \end{bmatrix}$$

2.6 Stability Parameters

For a very small disturbance from equilibrium position, hydrodynamic forces in journal can be regarded as linear functions of displacements and velocity vectors. The equation of disturbed motion of journal can be written by equating inertia force to stiffness and damping forces. The linearized equation of motion of journal in non-dimensional form is given by:

$$\left[\bar{M}_J\right]\left\{\ddot{\bar{X}}_J\right\} + \left[\bar{C}\right]\left\{\dot{\bar{X}}_J\right\} + \left[\bar{S}\right]\left\{\bar{X}_J\right\} = 0 \tag{11}$$

We can write equation of motion in matrix form:

$$\begin{bmatrix} \bar{M}_J & 0 \\ 0 & \bar{M}_J \end{bmatrix} \begin{Bmatrix} \ddot{\bar{X}}_J \\ \ddot{\bar{Z}}_J \end{Bmatrix} + \begin{bmatrix} \bar{C}_{xx} & \bar{C}_{xz} \\ \bar{C}_{zx} & \bar{C}_{zz} \end{bmatrix} \begin{Bmatrix} \dot{\bar{X}}_J \\ \dot{\bar{Z}}_J \end{Bmatrix} + \begin{bmatrix} \bar{S}_{xx} & \bar{S}_{xz} \\ \bar{S}_{zx} & \bar{S}_{zz} \end{bmatrix} \begin{Bmatrix} \bar{X}_J \\ \bar{Z}_J \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}$$

The stability margin of journal bearing system, in terms of critical mass $\overline{M}_c$, is obtained. The system is stable when $\overline{M}_J < \overline{M}_c$.

Threshold speed is that speed of journal at threshold of instability which can be obtained using relation given below:

$$\overline{\omega}_{th} = \left[\overline{M}_c \big/ \overline{F}_o \right]^{1/2} \tag{12}$$

Where, $\overline{F}_0$ is resultant fluid film force or reaction $\left(\frac{\partial \overline{h}}{\partial \overline{t}} = 0 \right)$

Where, the whirl frequency can be obtained as under:

$$\nu = \left[\left(\overline{S}_{xx}\overline{C}_{xx} + \overline{S}_{xz}\overline{C}_{xz} + \overline{S}_{zx}\overline{C}_{zx} + \overline{S}_{zz}\overline{C}_{zz} \right) \Big/ \left(\left(\overline{C}_{xx} + \overline{C}_{zz} \right) \times \overline{M}_c \right) \right]^{1/2} \tag{13}$$

2.6 Boundary Conditions

The boundary conditions used for lubricant flow field are described as [Sharma et al. (1999)]

1. Nodes situated on external boundary of bearing have zero relative pressure with respect to atmospheric pressure.
   $\overline{p}\,|_{\beta = \mp 1.0}=0.0$
2. The nodal flows are zero at internal nodes except those situated on slots and external boundaries.
3. Flow of lubricant through restrictor is equal to bearing input flow at slot.
4. At trailing edge of positive region $\overline{p} = \frac{\partial \overline{p}}{\partial \alpha}$ =0.0 according to Swift-Stieber cavitation condition.

3.0 SOLUTION PROCEDURE

The flow chart of an iterative scheme used to obtain converged solution for slot-entry hybrid journal bearing system operating with non-Newtonian lubricant has been shown in Fig. 2. The Reynold's equation with restrictor equation is solved after satisfying appropriate boundary conditions to yield pressure distribution in flow-field. The solution for Newtonian lubricant is obtained as initial trial solution to be used for the non-Newtonian case. The element fluidity matrix and right hand side vector of equation 3 are generated and assembled using usual assembly procedure in subroutine FLUID. In this subroutine the cross viscosity integrals $\overline{F}_0$, $\overline{F}_1$, $\overline{F}_2$ and fluid-film viscosity can be computed for both Newtonian and non-Newtonian lubricants. The values of cross viscosity integrals $\overline{F}_0$, $\overline{F}_1$ and $\overline{F}_2$ are obtained at each gauss point using Numerical integration (Simpson's rule). The shear strain rate $\bar{\gamma}$ is computed using equation 8, and corresponding equivalent shear stress $\bar{\tau}$ is obtained from equation 6 using Newton-Rapson's method. The apparent viscosity $\tilde{\mu}_a$ is computed at each gauss point, using equation 7. For continuity of the flow between restrictor and bearing, system equation is modified accordingly. The subroutine BOUNDRY modifies system equation for specified boundary conditions. The modified system equation is solved for nodal pressure in subroutine SOLVER using the Gaussian elimination technique. In BLOCK NON_NWTN iterative procedure is terminated when the difference in nodal pressures at each nod in successive iteration becomes less than predefined tolerance of $ZO \leq 0.001$ for non-Newtonian solution. The pressure field for lubrication flow field for Newtonian and non-Newtonian lubricants can be obtained as shown by BLOCK PRES. The final journal equilibrium position is established using an iterative scheme. Under a given bearing geometric parameters and for a given external vertical load, journal center position ($\overline{X}_J, \overline{Z}_J$) is unique. For a given external load, tentative values of journal center coordinates are fed as input. The corrections ($\Delta\overline{X}_J, \Delta\overline{Z}_J$) on assumed journal center coordinates ($\overline{X}_J, \overline{Z}_J$) are computed using following algorithm. The fluid film reaction components $\overline{F}_x, \overline{F}_z$ are expressed by Taylor's series about i^{th} journal center position. Assuming that alteration in journal center position are quite small and retaining terms only up to first order in Taylor's series expansion, the corrections ($\Delta\overline{X}_J\big|_i, \Delta\overline{Z}_J\big|_j$) on the coordinates are obtained as:

$$\Delta\overline{X}_J\Big|_i = -\frac{1}{D_J}\left[\frac{\partial\overline{F}_z}{\partial\overline{Z}_J}\bigg|_i \qquad -\frac{\partial\overline{F}_x}{\partial\overline{Z}_J}\bigg|_i\right]\begin{Bmatrix}\overline{F}_x\big|_i \\ \overline{F}_z\big|_i - \overline{W}_0\end{Bmatrix} \tag{14}$$

$$\Delta\overline{Z}_J\Big|_i = -\frac{1}{D_J}\left[-\frac{\partial\overline{F}_z}{\partial\overline{X}_J}\bigg|_i \qquad \frac{\partial\overline{F}_x}{\partial\overline{X}_J}\bigg|_i\right]\begin{Bmatrix}\overline{F}_x\big|_i \\ \overline{F}_z\big|_i - \overline{W}_0\end{Bmatrix} \tag{15}$$

Where,

$$D_J = \left[\frac{\partial\overline{F}_x}{\partial\Delta\overline{X}_J}\bigg|_i \frac{\partial\overline{F}_z}{\partial\Delta\overline{Z}_J}\bigg|_i - \frac{\partial\overline{F}_x}{\partial\Delta\overline{Z}_J}\bigg|_i \frac{\partial\overline{F}_z}{\partial\Delta\overline{X}_J}\bigg|_i\right]$$

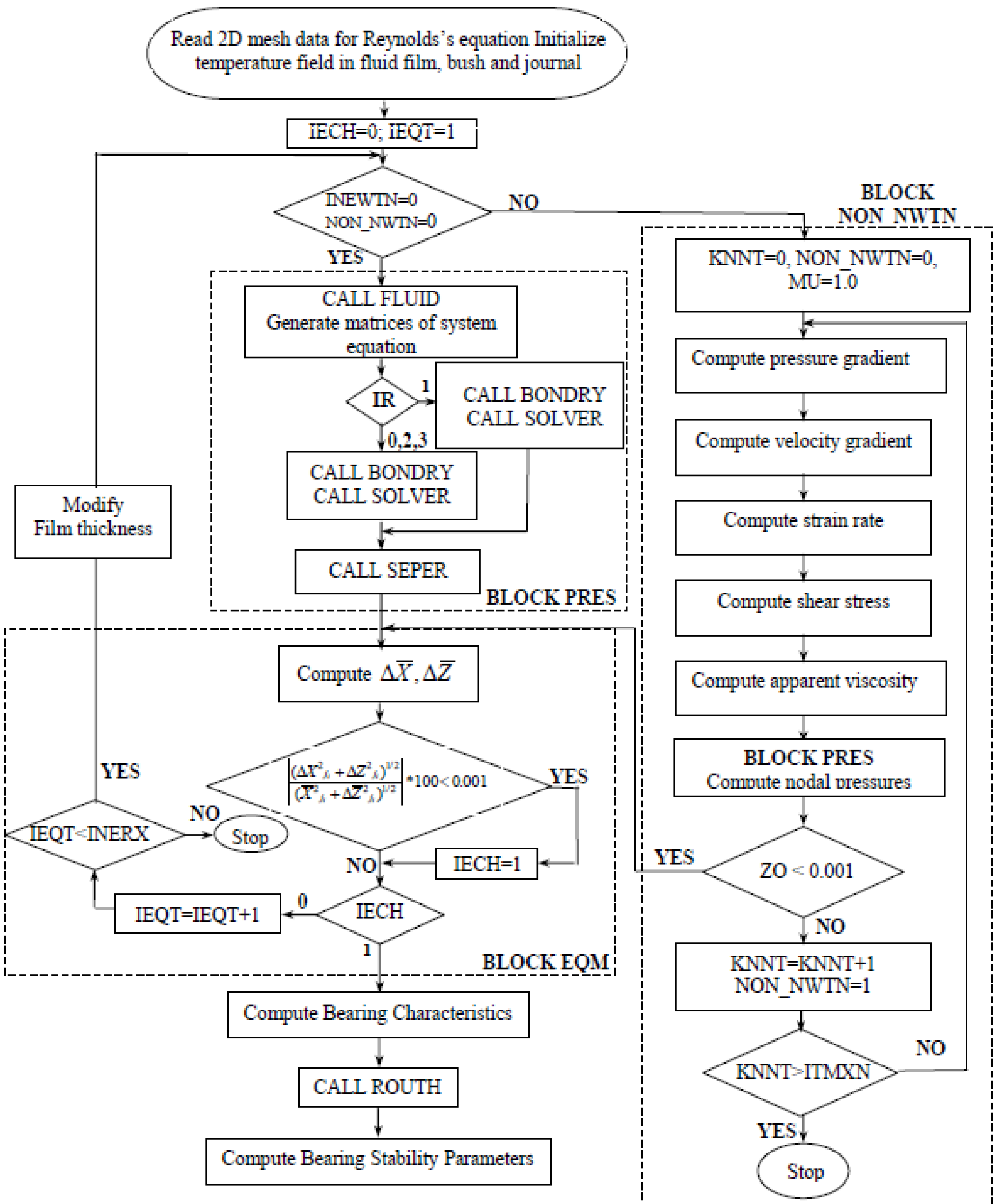
The new journal center position co-ordinate $\left\lfloor \overline{X}_J\big|_{i+1}, \overline{Z}_J\big|_{i+1}\right\rfloor$ are expressed as:

$$\overline{X}_J\Big|_{i+1} = \overline{X}_J\Big|_i + \Delta\overline{X}_J\Big|_i$$

$$\overline{Z}_J\Big|_{i+1} = \overline{Z}_J\Big|_i + \Delta\overline{Z}_J\Big|_i$$

The iterations are terminated when index 'IECH' attains a value equal to unity. The index for convergence 'IECH' attains a value equal to unity, when journal center equilibrium position satisfies criterion $\left[\dfrac{\left[\Delta\overline{X}_j{}^2+\Delta\overline{Z}_j{}^2\right]^{\frac{1}{2}}}{\left[\overline{X}_j{}^2+\overline{Z}_j{}^2\right]^{\frac{1}{2}}}\right]\times 100<0.001$

The performance characteristics are computed. The stability parameters (threshold speed and whirl frequency) are computed in subroutine ROUTH using Routh's criterion of stability.


Read 2D mesh data for Reynolds's equation Initialize temperature field in fluid film, bush and journal
IECH=0; IEQT=1
INEWTN=0 NON_NWTN=0
NO
YES
BLOCK NON_NWTN
KNNT=0, NON_NWTN=0, MU=1.0
CALL FLUID Generate matrices of system equation
IR
1
0,2,3
CALL BONDRY CALL SOLVER
CALL BONDRY CALL SOLVER
CALL SEPER
BLOCK PRES
Modify Film thickness
Compute $\Delta\bar{X}, \Delta\bar{Z}$
$\left|\frac{(\Delta X^2_A + \Delta Z^2_A)^{1/2}}{(\bar{X}^2_A + \bar{Z}^2_A)^{1/2}}\right| * 100 < 0.001$
YES
YES
IEQT<INERX
NO
Stop
NO
IECH=1
IEQT=IEQT+1
0
IECH
1
BLOCK EQM
Compute Bearing Characteristics
CALL ROUTH
Compute Bearing Stability Parameters
Compute pressure gradient
Compute velocity gradient
Compute strain rate
Compute shear stress
Compute apparent viscosity
BLOCK PRES
Compute nodal pressures
YES
ZO < 0.001
NO
KNNT=KNNT+1 NON_NWTN=1
KNNT>ITMXN
NO
YES
Stop

4.0 RESULT AND DISCUSSION

Based on analysis, a computer program is developed to compute dynamic performance characteristics of slot-entry journal bearings of different chosen configurations. The dynamic performance characteristics are computed for representative bearing geometric and operating parameters as shown in Table 1. The performance characteristics are computed for values of concentric design pressure ratio, $\beta^* = 0.5$, slot width ratio, SWR=0.25. The value of land width ratio is taken to be 0.25 in the present work as Rowe et al. (1977,1982) reported that optimum value of a_b / L is 0.25 from point of view of minimum power consumption.

To check validity of analysis, results for a two-row slot-entry non-recessed journal bearing (disregarding non-Newtonian behavior of lubricant) with 6 slots per row are compared with available results of Rowe et al. (1982). The computed results of load carrying capacity corresponding to different eccentricity ratios show a good agreement with maximum percentage deviation of about 2% as shown in Figure 3.

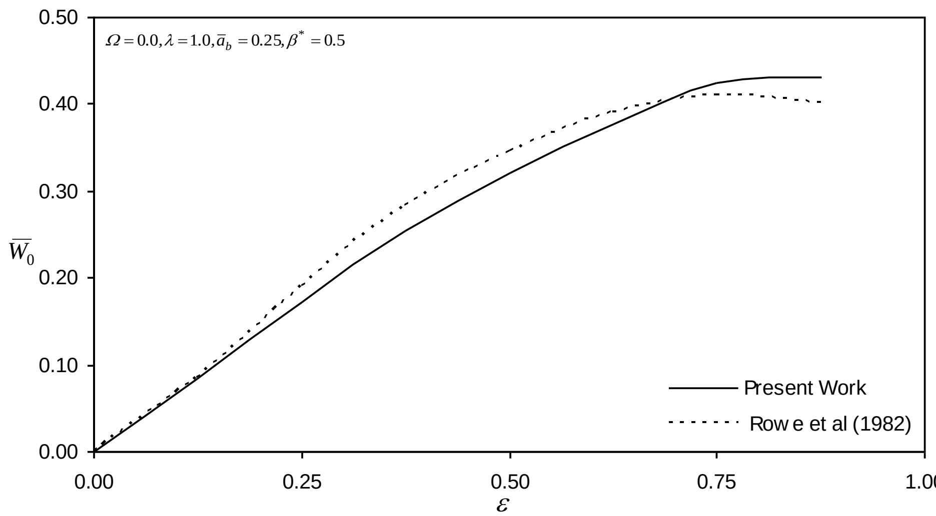


Figure 3: Comparison of load carrying capacity of symmetric slot-entry journal bearing

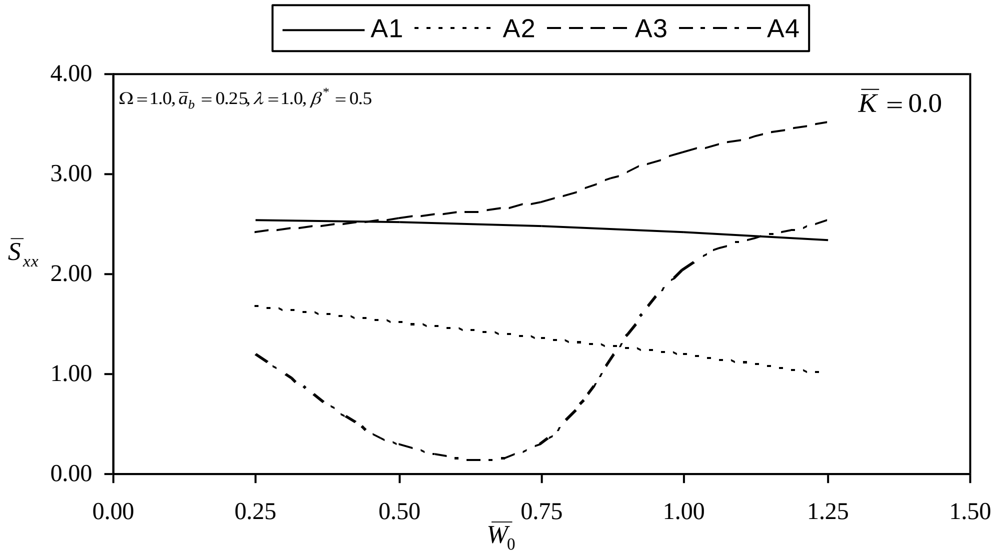
Table 1: Bearing geometric and operating parameters

Bearing Geometric Parameter	Symbol	Bearing Type: Hybrid	Bearing Type: Hydro-dynamic	Unit
Journal Radius	R_J	50	50	mm
Bush External Radius	R_2	55	100	mm
Radial Clearance	c	0.05	0.145	mm
Bearing Length	L	100	80	mm
Bush Thickness	t_b	5	---	mm
Land Width Ratio	a_b/L	0.25	---	---

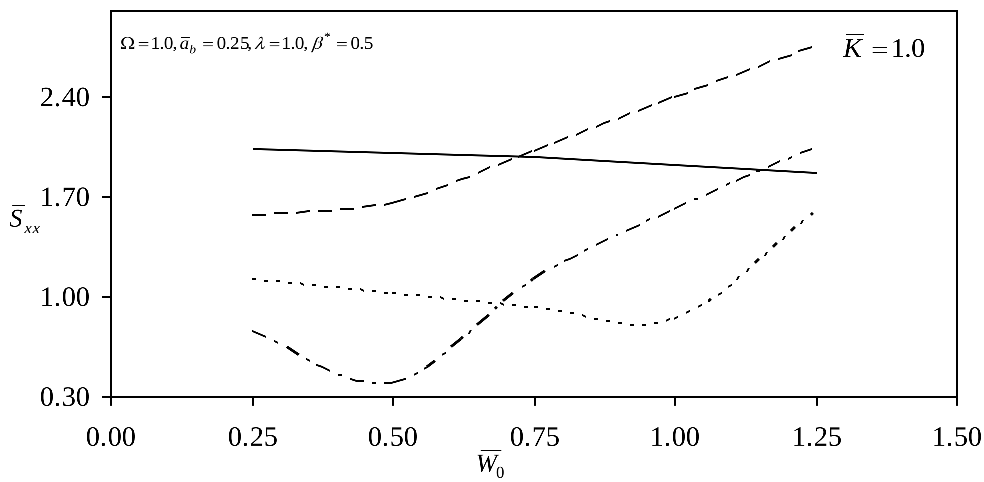
Lubricant Characteristics				
Density	ρ_f	860	860	kg. m^{-3}
Viscosity (at 40 degree Celsius)	μ_{ref}	0.02636	0.0277	Pa. s
Operating Parameters				
Journal Speed	N	2500	2000	rpm
External Load	W_o	22.4	1 to 10	kN
Supply Pressure	p_s	8.96×10^6	70×10^3	N. m^{-2}

4.1 Stiffness Coefficients ($\bar{S}_{XX}, \bar{S}_{ZZ}$)

The variation of direct stiffness coefficient ($\bar{S}_{XX}$) in horizontal direction with respect to load for different value of non linearity factor is shown in Figures 4 (a)-4 (b). At lower load $\overline{W_0} = 0.25$, direct stiffness coefficient ($\bar{S}_{XX}$) of A1 configuration is more and it is less for A4 configuration while operating with both Newtonian and non-Newtonian lubricant. At higher value of load $\overline{W_0} = 1.25$, value of direct stiffness coefficient ($\bar{S}_{XX}$) of A3 configuration is maximum. Figures 5(a)-5(b) show variation direct stiffness coefficient ($\bar{S}_{ZZ}$)in vertical direction with external load while operating with Newtonian and non-Newtonian lubricant respectively. It is clear from figures that for given range of load value of direct stiffness coefficient ($\bar{S}_{ZZ}$) of A1 configuration is maximum while operating with both Newtonian and non-Newtonian lubricant.



(a)



(b)

Figure 4: Variation of fluid-film stiffness coefficient ($\bar{S}_{xx}$) with external load ($\overline{W_0}$)

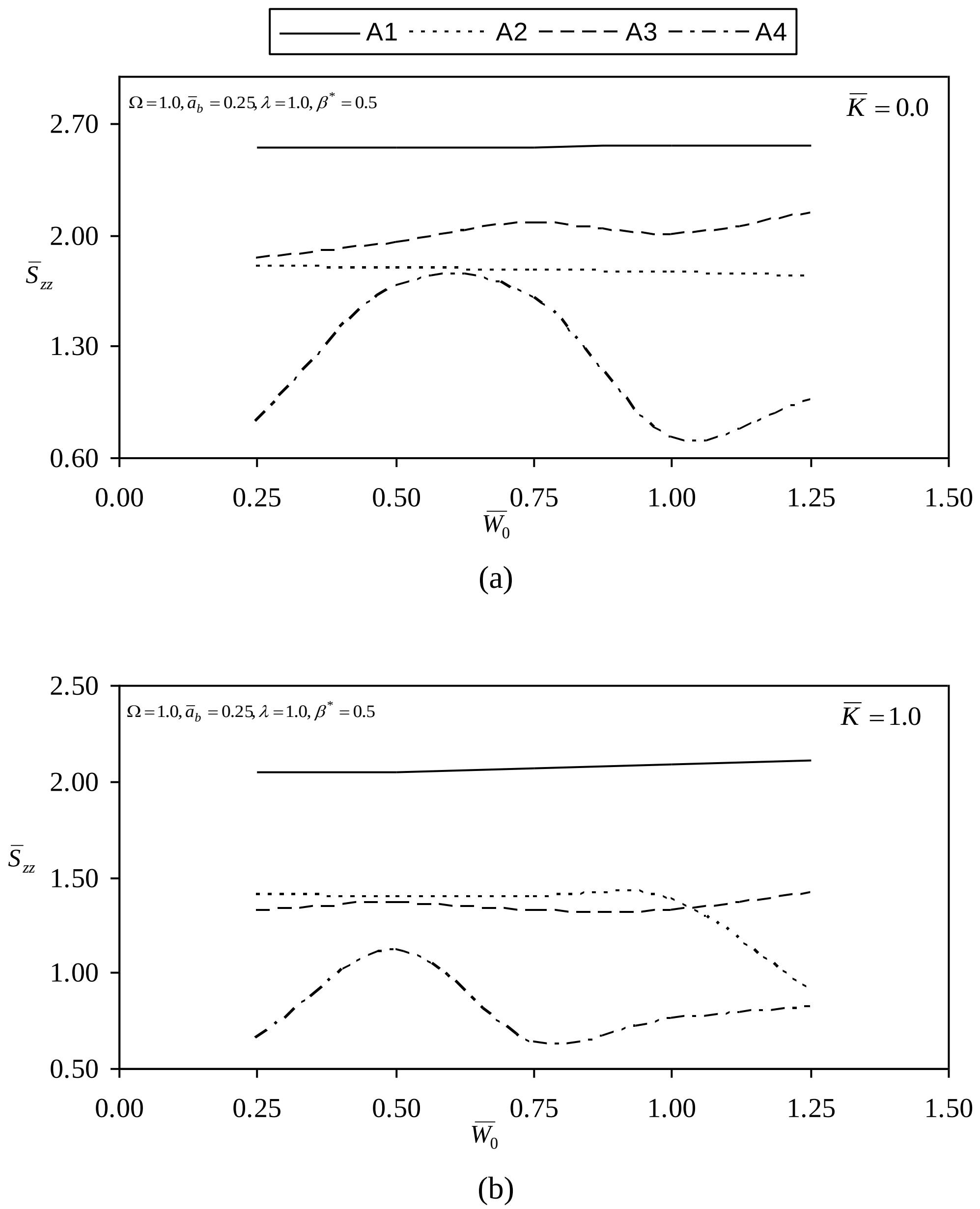
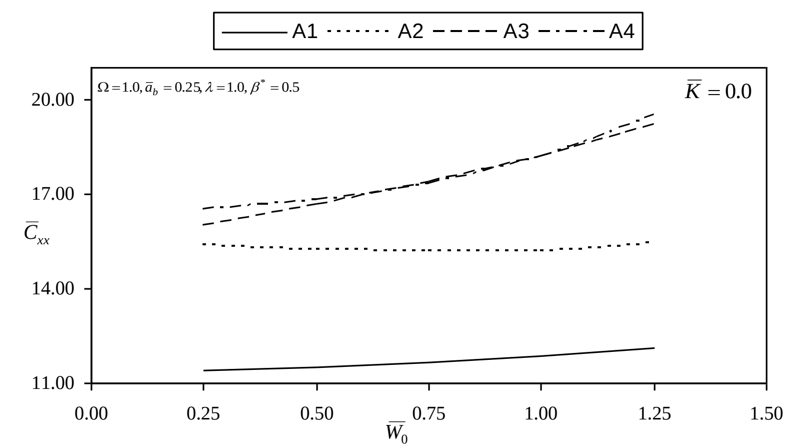


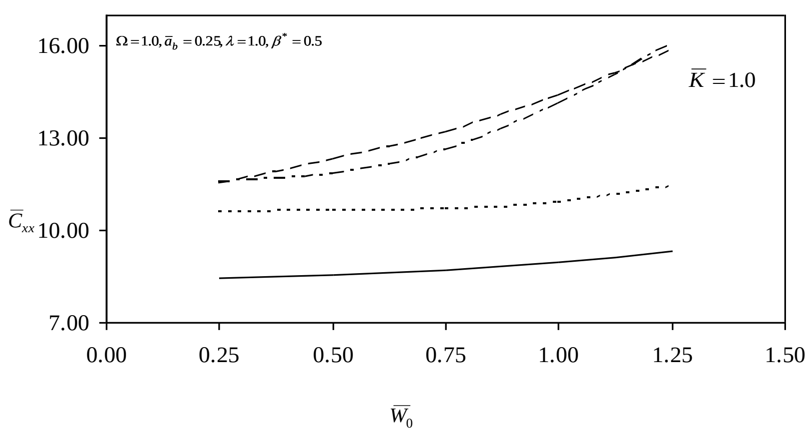
Figure 5: Variation of fluid-film stiffness coefficient ($\bar{S}_{zz}$) with external load ($\overline{W_0}$)

4.2 Damping Coefficients ($\overline{C}_{XX}, \overline{C}_{ZZ}$)

Figures 6(a)-6(b) and 7(a)-7(b) show the variation of direct damping coefficients ($\overline{C}_{XX}$ & $\overline{C}_{ZZ}$) with respect to external load for different value of non linearity factor of lubricant. It can be observed from figures that for all configurations, value of direct damping coefficients ($\overline{C}_{XX}$ & $\overline{C}_{ZZ}$) in horizontal direction and in vertical direction increases with increase in external load while operating with both Newtonian and non-Newtonian lubricant. At a given value of external load direct damping coefficients ($\overline{C}_{XX}$ & $\overline{C}_{ZZ}$) decrease with increase in non linearity factor of lubricant. At constant value of external load variation in value direct damping coefficients ($\overline{C}_{XX}$ & $\overline{C}_{ZZ}$) of A3 and A4 configurations is marginal for both Newtonian and non-Newtonian lubricant. The value of direct damping coefficients ($\overline{C}_{XX}$ & $\overline{C}_{ZZ}$) is minimum for A1 configuration for given load range and non linearity factor of lubricant.

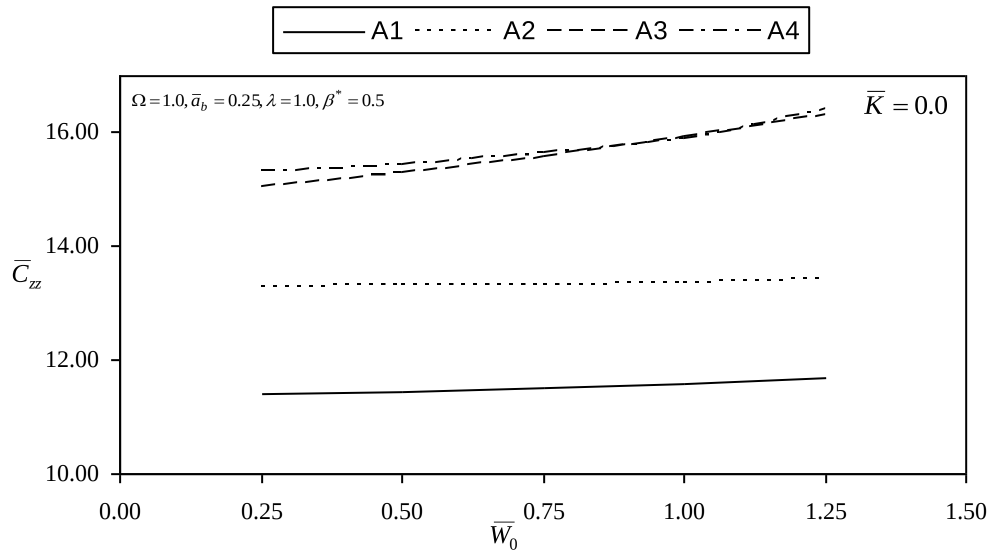


(a)

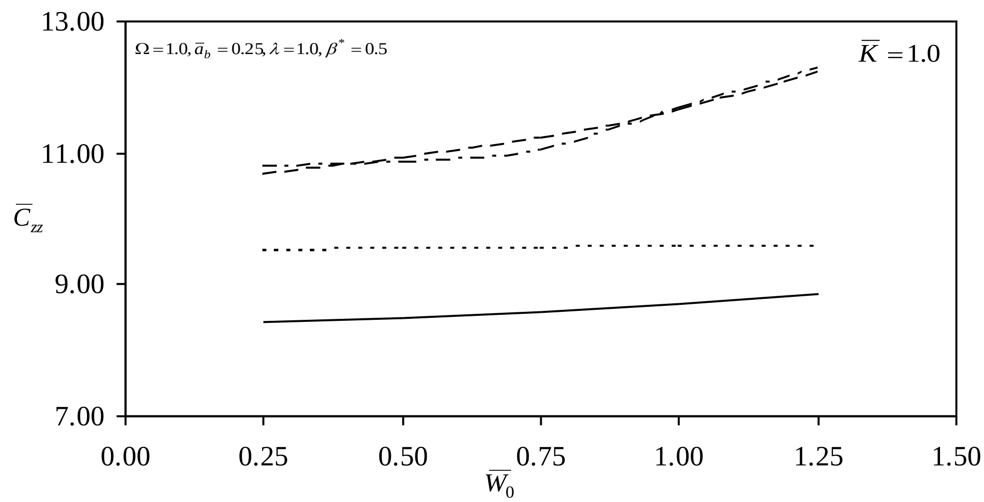


(b)

Figure 6: Variation of direct damping coefficient ($\overline{C}_{xx}$) with external load ($\overline{W}_0$)



(a)

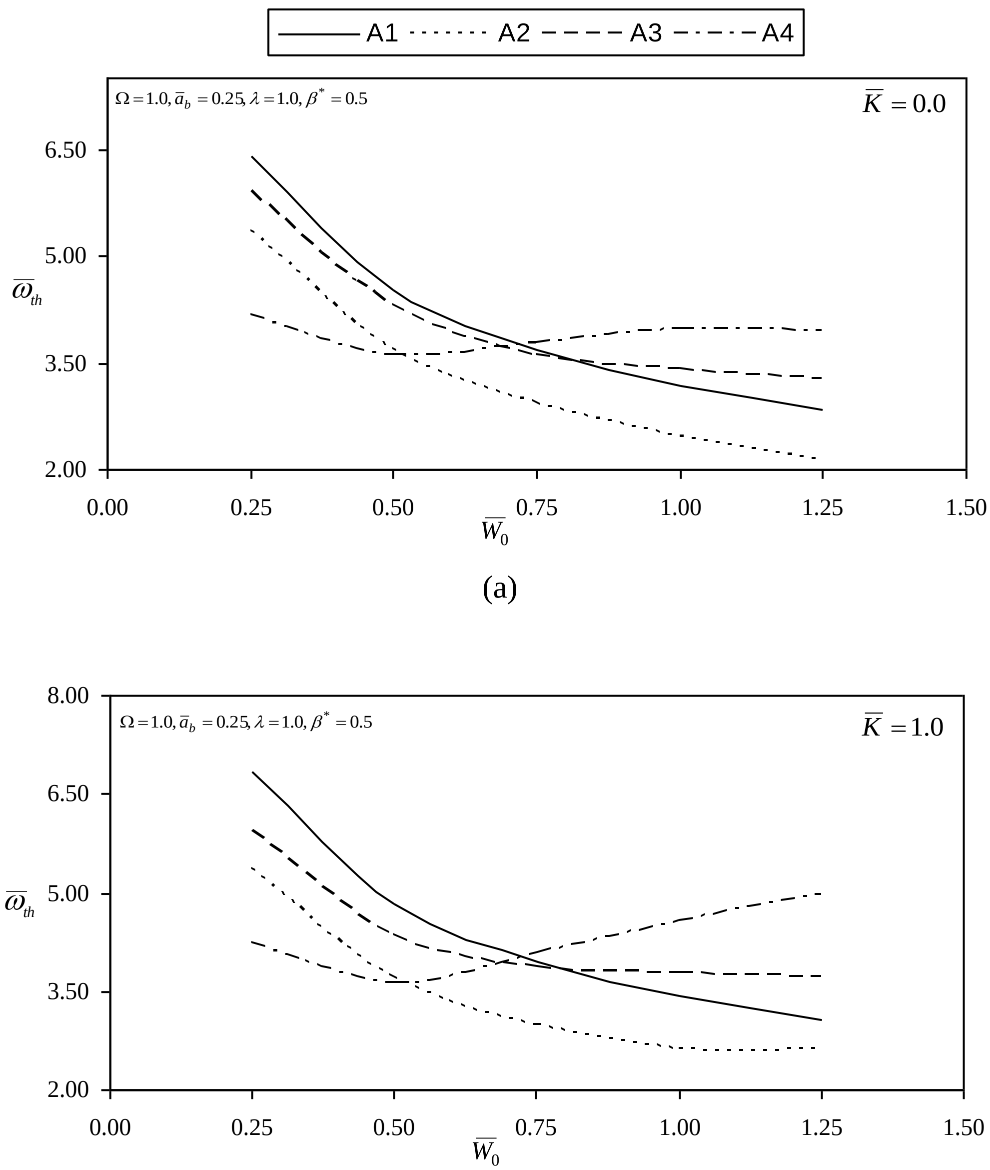


(b)

Figure 7: Variation of direct damping coefficient ($\bar{C}_{zz}$) with external load ($\bar{W}_0$)

4.3 Stability Parameters ($\bar{\varpi}_{th}$ and ν)

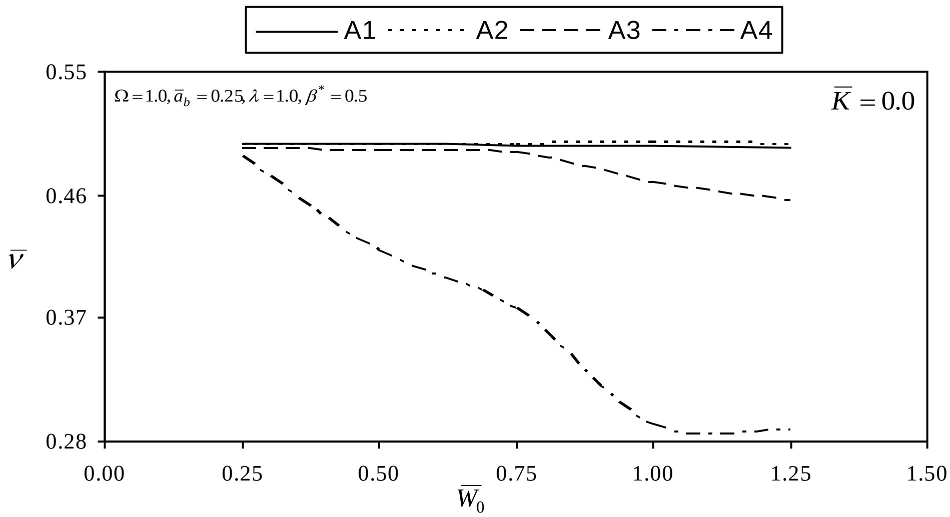
It may be observed from Figures 8 (a) - 8 (b) that value of threshold speed ($\bar{\varpi}_{th}$) for A1, A2 and A3 configurations decreases with increase in external load for given value of non linearity factor of lubricant while threshold speed ($\bar{\varpi}_{th}$) for A4 configuration increases with increase in external load $\bar{W}_0 > 0.5$. At given external load threshold speed ($\bar{\varpi}_{th}$) of all configurations increases with increase in non linearity factor of lubricant. At higher external load $\bar{W}_0 = 1.25$ threshold speed ($\bar{\varpi}_{th}$) of A4 configuration is maximum while it is minimum for A2 configuration while operating with both Newtonian and non-Newtonian lubricant. From Figures 9 (a) - 9 (b) it is clear that at lower value of load $\overline{W}_0 = 0.25$, whirl frequency (ν) is almost same for all configurations while at higher value of load $\overline{W}_0 = 1.25$ whirl frequency (ν) is minimum for A4 configuration for both Newtonian and non-Newtonian lubricant.



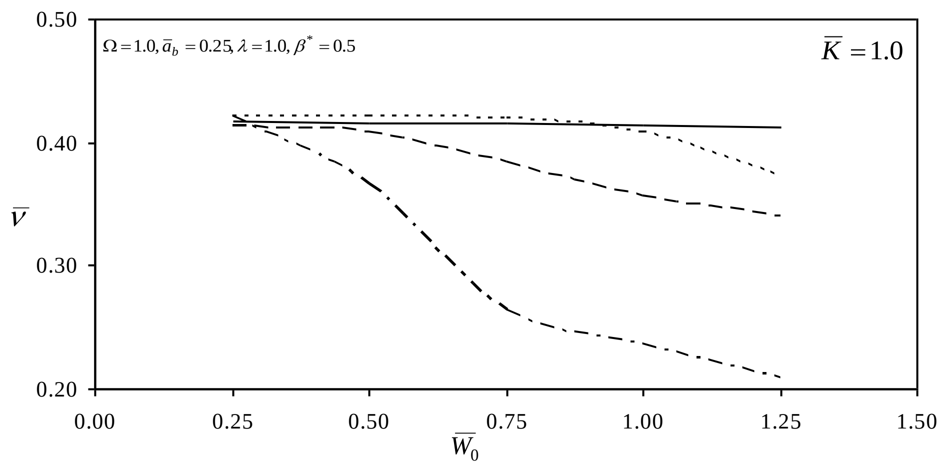
(a)

(b)

Figure 8: Variation of threshold speed ($\overline{\omega}_{th}$) with external load ($\overline{W}_0$)



(a)



(b)

Figure 9: Variation whirl frequency ($\bar{\nu}$) with external load ($\bar{W}_0$)

CONCLUSION

At higher external loads A4 configuration operating with Newtonian and non-Newtonian lubricants has maximum value damping coefficients ($\overline{C}_{xx}$ & $\overline{C}_{zz}$), threshold speed ($\overline{\omega}_{th}$) and minimum value of whirl frequency (ν). So from the stability point of view, A4 configuration is more stable while operating with lubricant with higher non-linearity factor at higher external load.

The effect of the variation of external loads on the values stiffness coefficients ($\overline{S}_{XX}$ & $\overline{S}_{ZZ}$), damping coefficients ($\overline{C}_{xx}$ & $\overline{C}_{zz}$) and whirl frequency (ν) is marginal for A1 configuration operating with Newtonian and non-Newtonian lubricants.

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