Research Article
Stability Analysis of Journal Bearing: Dynamic Characteristics

[1]S. Dinakaran and [2]S. Ramesh
[1]Department of Mechanical Engineering, Pavai College of Technology, Namakkal-637018,
[2]Department of Mechanical Engineering, Vel Tech High Tech Dr. Rangarajan Dr. Sakunthala
Engineering College, Avadi, Chennai-600062, Tamilnadu, India

Abstract: The purpose of the present study is to provide the dynamic characteristics of Journal bearings lubricated with couple stress fluids. Based upon the dynamic Reynolds governing equation for the different L/D ratio, Eccentricity ratio, the stiffness and damping coefficient of the system are evaluated. At Eccentricity ratio after 0.7, the small disturbances results affect in a higher stability in the journal bearing system. Using the classical Newton's multivariate Interpolation the empirical relation has derived for eccentricity ratio L/D ratio and the dynamic coefficient for finding the experimental stability characteristics.

Keywords: Couple stress fluid, damping co-efficient, eccentricity ratio, journal bearing, L/D ratio, stability, stiffness co-efficient

INTRODUCTION

In designing Journal bearings a first consideration is their static characteristics such as hydrodynamic film force, load-carrying capacity and co-efficient of friction etc. But under certain unexpected disturbances the bearing system is prone to an oscillating behavior. If the amplitude of oscillation grows too much, it may cause the undesired Journal-bearing contact and endanger safe operation of the whole machine system. Stability analysis of journal bearing system is a topic of intensive research. Several papers are available which deals with the performance enhancement both due to geometrical changes of the Journal bearing system and due to the changes in the lubricating fluid.

Many researchers have investigated the stability of hydrodynamic Journal Bearings. In Pai and Majumdar (1991) analyzed the stability characteristics of submerged plain journal bearings under a unidirectional constant load and variable rotating load. In Raghunandana et al. (1999) analyzed the effects of surface roughness on the stability of submerged plain journal bearings and studied the effects on Non-Newtonian lubricant on the stability of oil film journal bearings under a unidirectional constant load. In Kakoty and Majumdar (2000) analyzed the stability of journal bearings under the effects of fluid Inertia. These studies however were restricted to the stability of plain Journal bearings.

On the other hand, In Dwivedi (2012) analyzed the dynamic behavior of fluid film journal bearing by using Fuzzy logics. In Dhande et al. (2013) journal bearings models are developed for different speeds and eccentricity ratios to study the interaction between the fluid and elastic behavior of the bearing. In Mukesh et al. (2012), CFD technique used to predict the performance characteristics of a plain journal bearing. In Daniel and Cavalca (2013), the thermal effects have analyzed to find the behavior of hydrodynamic Journal. In Cai-Wan and Hsiang-Chen (2011) studied a systematic analysis of the dynamic behavior of a gear-bearing system with the turbulent flow effect, nonlinear suspension, nonlinear oil-film force and nonlinear gear mesh force. In Sarangi et al. (2004) analyzed the stiffness and damping characteristics of lubricated ball bearings. In Zirkelback and San Andres (1998) investigated the variation of the dynamic co-efficient for a hydrodynamic journal bearing with rotating grooves at a fixed journal position.

This study presents an analytical method to investigate the stability of a hydrodynamic journal bearing are calculated using the Newton's Multivariate Interpolation. The Linear equations of motion can be represented as a parametrically excited system because the dynamic co-efficient have time-varying components due to the rotating journal even in steady state. Their solution can be assumed as a Fourier series expansion so that the equations of motion can be rewritten as simultaneous algebraic equations with respect to the Fourier Coefficients. The validity of this research is proved by the comparison of the stability chart with the L/D and ε obtained from the equations of motion.

Corresponding Author: S. Dinakaran, Department of Mechanical Engineering, Pavai College of Technology, Namakkal-637018, Tamilnadu, India, Tel.: +91 9994777692
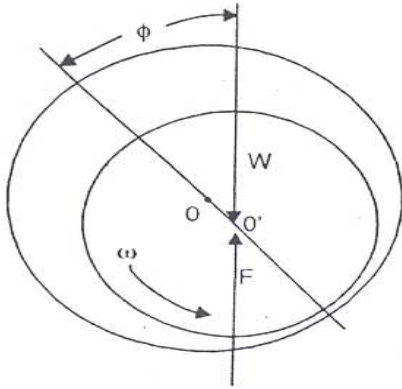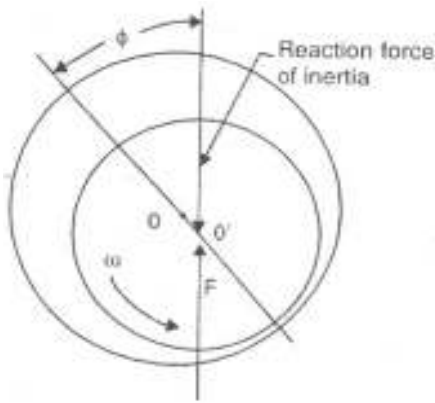
Fig. 1: Journal bearing
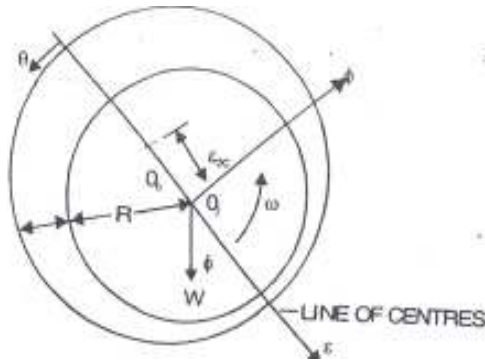

Fig. 2: Stable position

Fig. 3: Unloaded shaft in motion

MATERIALS AND METHODS

Basic equations: Consider the physical configuration of the cross section for a rigid rotor supported on long bearings shown in Fig. 1. In this position the resultant force on the shaft must be equal and opposite to the applied radial load W (Fig. 2). Let us now consider the motion of an unloaded shaft which is momentarily displayed from 'O' to O' as shown in Fig. 3. Due to this the film exerts a resultant force F on the shaft. As there is no opposing force W, the force F will now be spent

in acceleration the shaft. The shaft centres will not e toward the bearing center 'O'. The shaft will be forced to move in an orbit around the bearing centre and the whirl will continue. This whirling motion may be accelerating stable or may die out depending on the attitude angle and the damping characteristics of the system. It should be kept in mind that an important assumption in this explanation is that the bearing should be operating under an unloaded or lightly loaded condition. Half frequency whirl, resonant whirls, oil whirl are other instability in some rotating machines. These also a self-excited vibration. Here the vibration is supported by the fluid-film action, while the frequency is controlled by the system critical speed.

Assuming that the journal whirls about its mean steady state position, the pressure and film thickness are expressed as:

$$\bar{p} = \bar{p}_1 \varepsilon_1 \, e^{i\tau p} + \bar{p}_2 \varepsilon_0 \, \Phi \, e^{i\tau p} \qquad (1)$$

$$h = h_0 + \varepsilon_1 \, e^{i\tau p} \cos \theta + \varepsilon_0 \, e^{i\tau p} \sin \theta \qquad (2)$$

where,

$$\varepsilon = \varepsilon_0 + \varepsilon_1 \, e^{i\tau p} \qquad (3)$$

$$\psi = \psi_0 + \psi_1 \, e^{i\tau p} \qquad (4)$$

Substituting (1) to (4) in Reynolds equation, we get steady state and perturbed Eq. (5):

$$\frac{\partial}{\partial \theta}\left[\bar{h}_0^3 \frac{\partial \overline{p_0}}{\partial \theta}\right] + \frac{\partial}{\partial \bar{z}}\left[\bar{h}_0^3 \frac{\partial \overline{p_0}}{\partial z}\right] = 6 \frac{\partial \bar{h}}{\partial \theta}$$

That perturbed equations are solved using finite difference method using the steady state pressure distribution and appropriate boundary conditions for $\bar{p}_1$ and $\bar{p}_2$.

Stiffness and damping co-efficient are $\bar{k}_{xx}$, $\bar{k}_{xz}$, $\bar{k}_{zx}$, $\bar{k}_{zz}$ in N/m and $\bar{D}_{xx}$, $\bar{D}_{xz}$, $\bar{D}_{zx}$, $\bar{D}_{zz}$ in N/m/sec.

p = Pressure in bearing clearance in N/m^2
p_0 = Feed Pressure in N/m^2
ε = Eccentricity Ratio (e/c)
Φ = Attitude angle in deg
θ = Angular Co-ordinate in deg
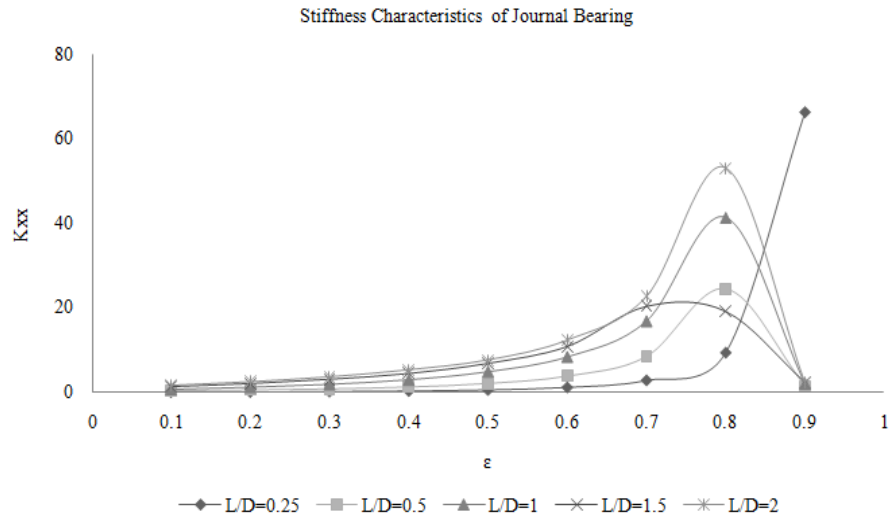h = Film Thickness in mm
h_0 = Minimum Film Thickness in mm
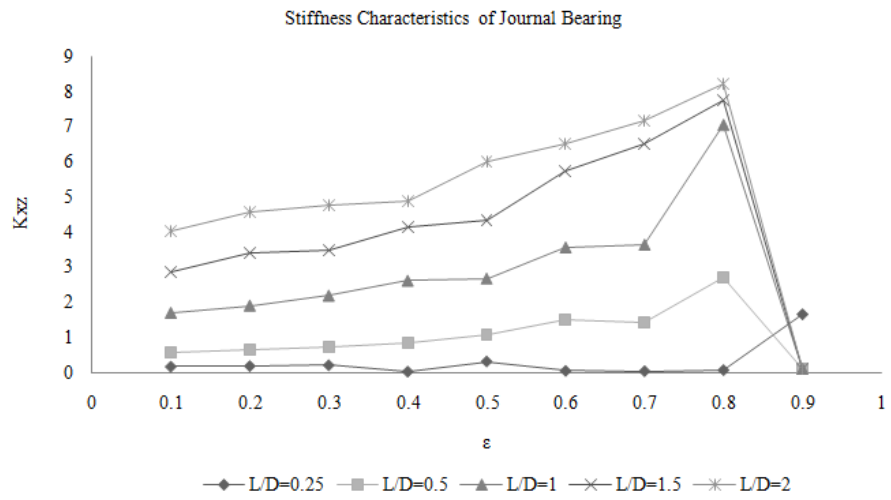
Bearing performance characteristics:
Load carrying capacity: The fluid film reaction components in 'y' and 'x' directions are given by:

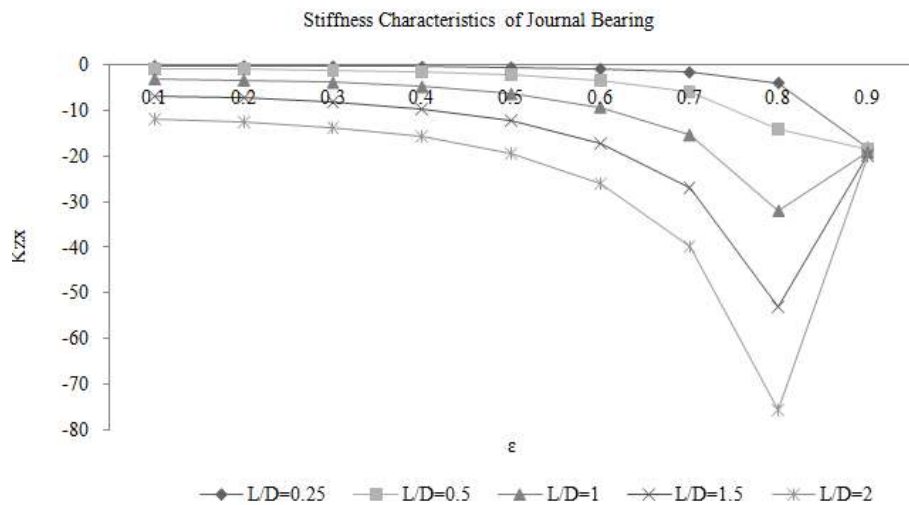$$\overline{W} \, x = \int \bar{p} \cos \theta . d\bar{\theta}, \, d\bar{y}$$

$$\overline{W} \, y = \int \bar{p} \sin \theta . d\bar{\theta}, \, d\bar{y}$$

Stiffness Characteristics of Journal Bearing

(a)

Stiffness Characteristics of Journal Bearing

(b)

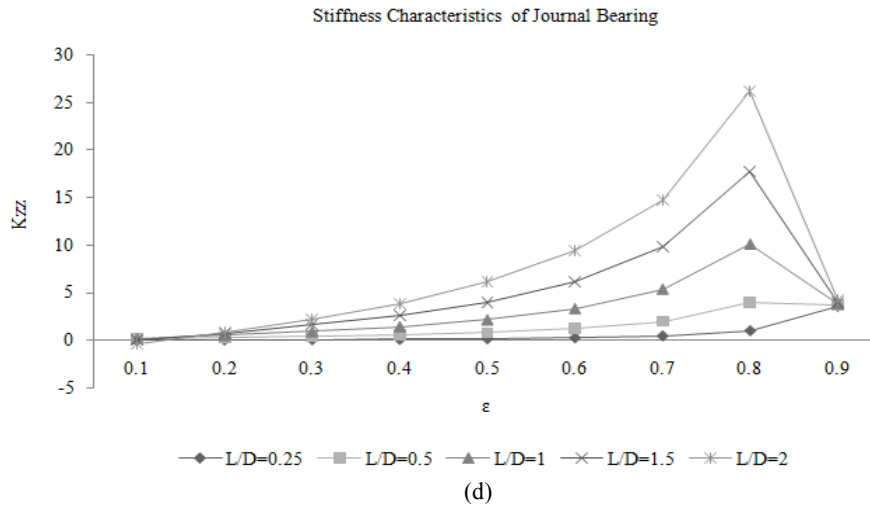Stiffness Characteristics of Journal Bearing

(c)

(d)

Fig. 4: Stiffness co-efficient vs. eccentricity (for different L/D)

For vertical support, the following conditions are satisfied journal centre equilibrium position:

$$\overline{W}_v = W \text{ and } \overline{W}_h = 0$$

Attitude angle: Angle between the load line and the line joining the bearing and journal centers is defined as the attitude angle. The attitude angle is established by satisfying the condition of vertical load support.

Stiffness co-efficient: If the journal center is displaced from its static equilibrium position, then in general, the difference in the components of the fluid-film reactions in the disturbed position and in the static equilibrium position gives the out of balance components of the fluid-film force:

$$\begin{array}{cc} \overline{k}\,xx & \overline{k}\,xz \\ \overline{k}\,zx & \overline{k}\,zz \end{array} = \begin{array}{cc} \dfrac{\partial \overline{W}\,x}{\partial \overline{x}} & \dfrac{\partial \overline{W}\,x}{\partial \overline{z}} \\ \dfrac{\partial \overline{W}\,z}{\partial \overline{x}} & \dfrac{\partial \overline{W}\,z}{\partial \overline{z}} \end{array}$$

The first subscript of the stiffness coefficients denotes the direction of force and the second, the direction of displacement.

Damping co-efficient: The damping co-efficient are defined as:

$$\begin{array}{cc} \overline{C}\,xx & \overline{C}\,xz \\ \overline{C}\,zx & \overline{C}\,zz \end{array} = \begin{array}{cc} \dfrac{\partial \overline{W}\,x}{\partial \overline{u}} & \dfrac{\partial \overline{W}\,x}{\partial \overline{v}} \\ \dfrac{\partial \overline{W}\,z}{\partial \overline{u}} & \dfrac{\partial \overline{W}\,z}{\partial \overline{v}} \end{array}$$

The first subscript of the damping co-efficient denotes the direction of force and the second, the velocity.

RESULTS AND DISCUSSION

The dynamic characteristics of journal bearing have calculated for different L/D ratio and ε. The stiffness and damping co-efficient have calculated and the various relationships have plotted in graph. From the result the relationship between the eccentricity ratio and L/D ratio have identified from the theoretical results.

The Variation of the stiffness co-efficient with L/D Ratio and eccentricity ratio has shown in Fig. 4a to d. ε≥0.7 the parameters have varied differently, its affect the stability.

The Variation of the damping co-efficient with L/D Ratio and Eccentricity has shown in Fig. 5a to d. Similar variation has got in damping co-efficient.

CONCLUSION

In the present study the effect of dynamic characteristics is analyzed. From this analysis the following points has been identified. These are:

- Load carrying capacity has the vital role for increasing the eccentricity ratio. Particularly after reaching ε = 0.7, it is observed that the load capacity increased very drastically. So the stiffness and damping co-efficient may involve the major role to disturbing the bearing systems.
- At that particular eccentricity ratio, the stability of a journal bearing system will disturb the system due to varying the load capacity and dynamic coefficient. This empirical relation will helps to identify the dynamic coefficient in practical manner. It will assist to resist the instability condition in the system.
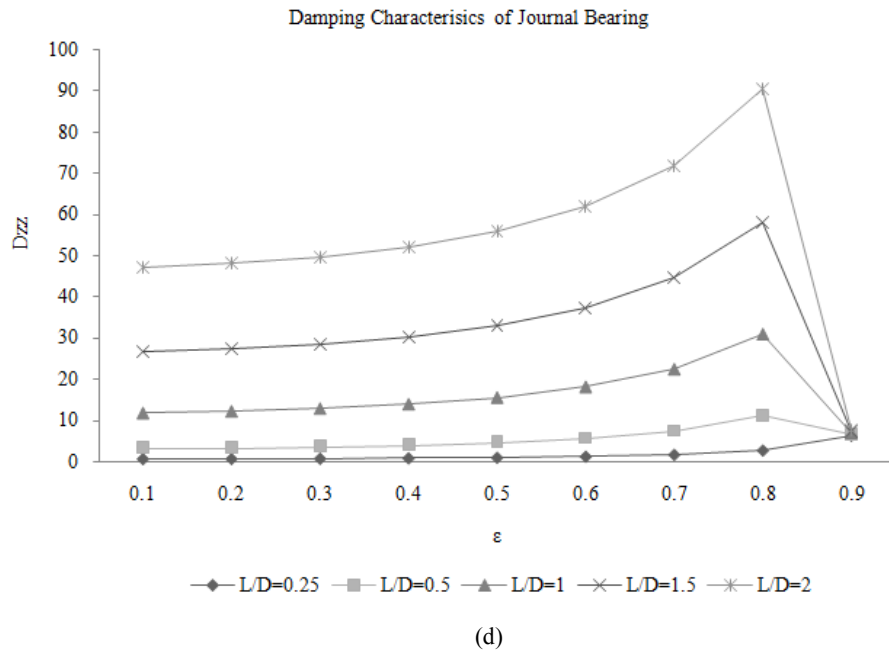
(a)

(b)

(c)

(d)
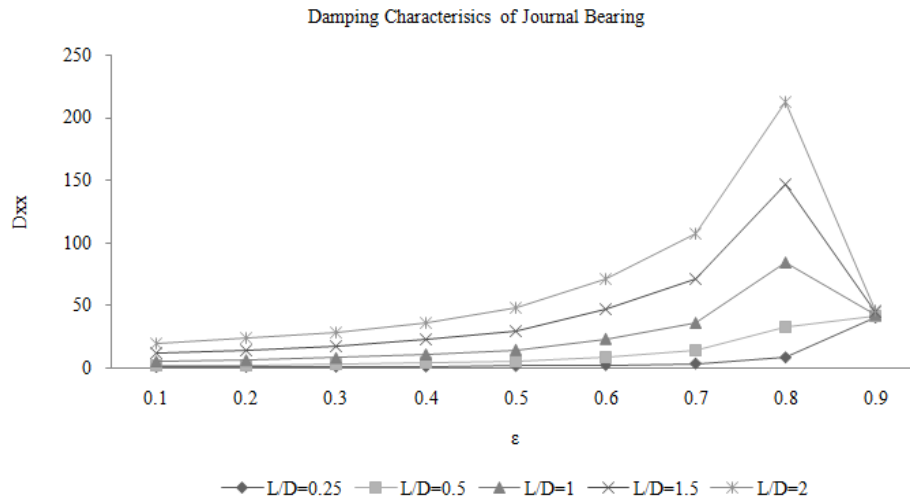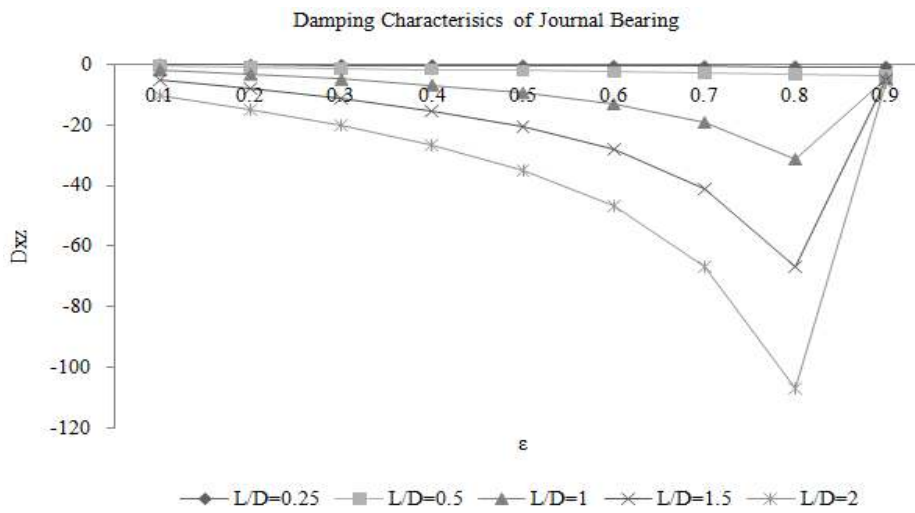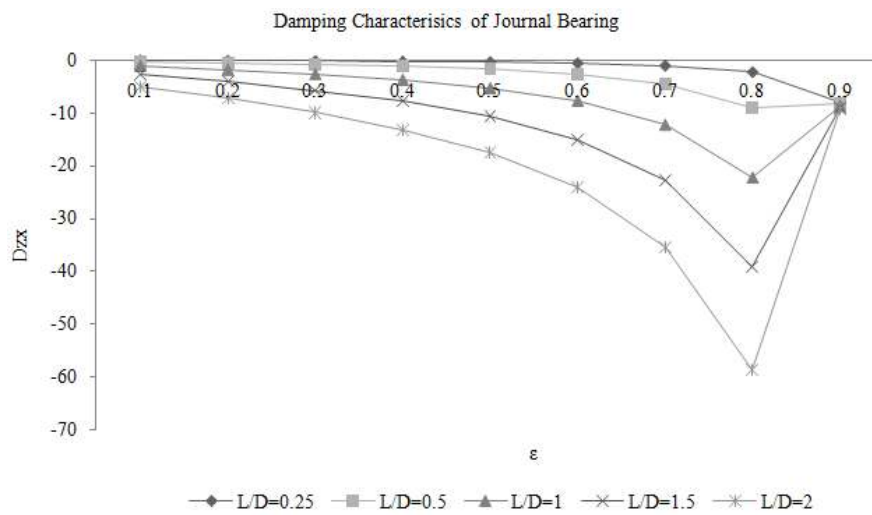
Fig. 5: Damping characteristics vs. eccentricity (for different L/D)

REFERENCES

Cai-Wan, C.J. and H. Hsiang-Chen, 2011. Chaotic responses on gear pair system equipped with journal bearings under turbulent flow. Appl. Math. Model., 36: 2600-2613.

Daniel, G.B. and K.L. Cavalca, 2013. Evaluation of the thermal effects in tilting pad bearing. Int. J. Rotat. Mach., 2013: 1-17, Article ID 725268.

Dhande, D., D.W. Pande and V. Chatarkar, 2013. Analysis of hydrodynamic journal bearing using fluid structure interaction approach. Int. J. Eng. Trends Tech., 4(8): 3389-3392.

Dwivedi, V.K., 2012. Fuzzy based decision making for selection of fluid film journal bearing. Int. J. Fuzzy Logic Syst., 2(3).

Kakoty, S.K. and B.C. Majumdar, 2000. Effect of fluid inertia on stability of oil journal bearings. ASME J. Tribol., 122: 741-745.

Mukesh, S., K.G. Ashish, D. Ashish, 2012. Thermohydrodynamic analysis of a journal bearing using CFD as a tool. Int. J. Sci. Res. Publ., 2(9).

Pai, R. and B.C. Majumdar, 1991. Stability analysis of flexible supported rough submerged oil journal bearings. Tribol. T., 40(3): 437-444.

Raghunandana, K., R. Jonnadula and B.C. Majumdar, 1999. Stability of journal bearing systems using non-newtanion lubricants: A non-linear transient analysis. Tribol. Int., 32: 179-184.

Sarangi, M., S., B.C. Majumdar and A.S. Sekhar, 2004. Stiffness and damping characteristics of lubricated ball bearings, Part I: Theorical formulation. J. Eng. Tribol., IMechE Part-J., 218: 529-538.

Zirkelback, N. and L. San Andres, 1998. Finite element analysis of herringbone groove journal bearings: A parametric study. ASME J. Tribol., 120: 234-240.