1. Introduction

Circular fluid film bearings can only offer a little improvement on the stability of rotor bearing systems. Attention was therefore directed on non-circular forms like lemon bore, offset half, pressure dam and bearings by Akkok et al. [1], Schuller [2,3], and Malik et al. [4,5]. The notable feature of all these types is their contribution to an improvement in performance and therefore is called the anti-whirl, multi-film bearings.

Hargreaves [6] did the first study dealing with the tri-taper bearing, which predicted the influence of manufacturing tolerances, misalignment as well as turbulence on the steady state performance of a tri-taper journal bearing. His study concluded that over-sized ramps results in an increased oil flow rate and a decreased load-carrying capacity. Rao et al. [7] made a comparative study on the effect of Reynolds and JFO cavitation boundary condition on the steady state performance of a tri-taper journal bearing. Theoretical investigations are carried out by Rao et al. [8] to ascertain the whirl instability of a rotor supported on a tri-taper journal bearing, operating under steady, periodic and variable rotating load, using non-linear transient approach as given by Majumdar and Brew [9]. The dynamic equations of motion are solved by fourth order Runge-Kutta method to result in eccentricity ratio, attitude angle and their derivatives, which are used for the next time step. Hydrodynamic forces are then evaluated using these values in the time-dependent Reynolds equation in two dimensions for incompressible fluid. Stability of the rotor bearing system is determined from the journal locus.

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In this paper, the stability characteristics of a tri-taper journal bearing are analyzed theoretically. The time dependent form of Reynolds equation in two dimensions for incompressible fluid is solved numerically by finite difference method, satisfying the Reynolds boundary conditions. Dynamic coefficients are determined using the method of linearization of bearing reactions. A study with various ramp size’s and L/D ratio’s predicts that a bearing with higher ramp size and higher L/D ratio results in an increased mass parameter and improved stability margin of the bearing.

KEY WORDS: Fluid film bearing; Linearized bearing reactions; Stiffness and damping; Stability; Tri-taper journal bearing.

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In this paper, a linearized perturbation approach is used to study the stability characteristics of tri-taper journal using the Reynolds boundary condition. The time-dependent Reynolds equation in two dimensions is solved for this purpose. A first order perturbation method is used to find the dynamic pressures in the clearance space. The stiffness and damping coefficients (both direct and cross components) are calculated from the real and imaginary parts of the integrated pressures. These coefficients are then used in the equations of motion of a rigid rotor for estimating the mass parameter, a measure of stability. The effect of ramp size and L/D ratio on the stability of these bearings is studied. The results are in dimensionless form and can be conveniently used in the design of these bearings.

2. Bearing Geometry

Geometrical arrangement of a tri-taper bearing is similar to that of a three-lobe journal bearing. However, this bearing is akin to three tapered land composite thrust bearings arranged around a bearing circle. There are three lubricant supply grooves, equispaced at 120º apart, around the bearing periphery. Each of the three sections consists of equal lengths of the bearing base circle and a section offset from the base circle, such that it merges smoothly with the base circle and provides a predetermined depth at or near to the lubricant supply groove. The sectional view of the tri-taper journal bearing is shown in Fig. 1.

3. Theory

The basic differential equation for pressure distribution in the bearing clearance under dynamic conditions for incompressible fluid may be written as

$$\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) + h^3 \frac{\partial^2 p}{\partial z^2} = 6 \eta U \frac{dh}{dx} + 12 \eta \frac{\partial h}{\partial \tau} \quad (1)$$

The lubricant film profile is determined by superimposing the effect of ramp on the conventional expression for the film thickness. Film thickness equation for all configurations of the bearing can be given by:

$$h = C + e \cos(\theta^* - \psi) + \text{ramp} \quad (2)$$

Using the following substitutions:

$$\theta = \frac{x}{R}, \bar{z} = \frac{z}{L}, \bar{p} = \frac{pC^2}{\eta_o R^2}, \tau = \omega_p \tau, h = \frac{h}{C}$$

$$= 1 + e \cos(\theta^* - \psi) + R_{\text{ramp}}$$

Equation 1 can be non-dimensionalized as:

$$\frac{\partial}{\partial \theta} \left( \bar{h}^3 \frac{\partial \bar{p}}{\partial \theta} \right) + \frac{1}{4} \left( \frac{\partial \bar{p}}{\partial \theta} \right)^2 \bar{h}^3 \frac{\partial^2 \bar{p}}{\partial \bar{z}^2} = 6 \frac{d\bar{h}}{d\theta} + 12 \bar{\lambda} \frac{\partial \bar{h}}{\partial \tau} \quad (3)$$

where

$$\lambda = \frac{\omega_p}{\omega} \frac{d\bar{h}}{d\theta} = -e \sin(\theta^* - \psi) + \frac{\partial (\text{ramp})}{\partial \theta}$$

and

$$\frac{d\bar{h}}{d\tau} = e \cos(\theta^* - \psi) + \frac{\partial (\text{ramp})}{\partial \tau}$$

4. Pressure Boundary Condition

Pressure along the bearing edges was set to zero.

$$\bar{p}_o = 0.0 \text{ at } \bar{z}=0 \text{ and } \bar{z}=1 \text{ for all } \theta$$

Cavitation was allowed to occur at ambient pressure by setting all calculated negative pressure equal to zero throughout the iterative solution scheme. This implies that the lubricant film ruptures and reforms when:

$$p_o = 0, \frac{\partial p}{\partial \theta} = 0$$

The non-dimensionalized supply pressure \(\bar{p}_s = 0.1362\) was set at the grooves.

5. Steady State Characteristics

Under steady state conditions, the time-dependent term is dropped. The governing equation will then be:

$$\frac{\partial}{\partial \theta} \left( \bar{h}^3 \frac{\partial \bar{p}_o}{\partial \theta} \right) + \frac{1}{4} \left( \frac{\partial \bar{p}_o}{\partial \theta} \right)^2 \bar{h}^3 \frac{\partial^2 \bar{p}_o}{\partial \bar{z}^2} = 6 \frac{d\bar{h}_o}{d\theta} \quad (4)$$

In the conventional cylindrical bearing the coordinate \(\theta\) in the circumferential direction is taken from the position of maximum film thickness. In the grooved bearing this position needs to be found beforehand. This is done by assuming an arbitrary value of attitude angle \(\psi\) and the coordinate \(\bar{\theta}\) is measured from the vertical position. Using \(\psi\) the film thickness is obtained from Eqs. 2 and 4 and is solved for pressure using the boundary conditions. The force components along and perpendicular to the line of centres are found from:

$$W_c = \frac{W_o C^2}{\eta U R^2 L} = -\int_0^1 \int_0^{2\pi} \bar{p}_o \cos(\theta^* - \psi) \ d\theta \ d\bar{z} \quad (5)$$
\[ \vec{W}_t = \frac{W_t C^2}{\eta U R^2 L} = \int_0^1 \int_0^{2\pi} \bar{p}_0 \sin(\theta^* - \psi) \, d\theta \, d\bar{z} \]  
(6)

The load capacity and attitude angle are:
\[ \vec{W} = [\vec{W}_t^2 + \vec{W}_t^2]^{\frac{1}{2}} \]  
(7)
\[ \phi_0 = \tan^{-1} \left( \frac{\vec{W}_t}{\vec{W}_t} \right) \]  
(8)

After each calculation, the attitude angle calculated from Eq. 8 is compared with the assumed value of attitude angle (\(\psi\)). The value of \(\psi\) is modified with a small increment of \(\psi\) and the Reynolds equation is solved using this modified value until \(\psi\) is equal to \(\phi_0\).

6. Dynamic Characteristics

The Reynolds equation under dynamic conditions, Eq. 3 is solved assuming that the journal performs a periodic motion of small amplitude \(\Re(\epsilon e^{i\theta})\) along the line of centres and \(\Re(\Phi e^{i\theta})\) perpendicular to the line of centres around its steady state position \(\epsilon_0\) and \(\Phi_0\). It is assumed at the onset of instability that the position of the journal centre can be defined as a steady state value \((\epsilon_0, \Phi_0)\) together with a harmonic vibration of frequency \(\omega_\psi\). Thus:

\[ \epsilon = \epsilon_0 + \epsilon_1 e^{i\theta} \]
\[ \Phi = \Phi_0 + \Phi_1 e^{i\theta} \]

where

\[ |\epsilon_1| < |\epsilon_0| \]
\[ |\Phi_1| < |\Phi_0| \]

Under this condition (i.e. for small amplitude of vibration) first-order perturbation will be valid.

Pressure and film thickness can be written as:
\[ \bar{p} = \bar{p}_0 + \epsilon_1 e^{i\theta} \bar{p}_1 + \epsilon_0 \Phi_1 e^{i\theta} \bar{p}_2 \]  
(9)
\[ \bar{h} = \bar{h}_0 + \epsilon_1 e^{i\theta} \cos \theta + \epsilon_0 \Phi_1 e^{i\theta} \sin \theta \]  
(10)

Substitution of Eqs. 9 and 10 in Eq. 3 and retaining up to first linear terms gives the following equations:

\[ \left( \bar{h}_0 \frac{\partial^2 \bar{p}_0}{\partial \theta^2} \right) + 3 \bar{h}_0 \frac{\partial \bar{p}_0}{\partial \theta} \frac{\partial \bar{p}_0}{\partial \theta} + \frac{1}{4} \left( \frac{D}{L} \right)^2 \bar{h}_0 \frac{\partial^2 \bar{p}_0}{\partial \bar{z}^2} - 6 \frac{d \bar{h}_0}{d \theta} = 0 \]  
(11)

\[ \left( \bar{h}_0 \frac{\partial^2 \bar{p}_1}{\partial \theta^2} \right) + 3 \bar{h}_0 \cos \theta \frac{\partial^2 \bar{p}_0}{\partial \theta^2} - 3 \bar{h}_0 \sin \theta \frac{\partial \bar{p}_0}{\partial \theta} \]
\[ + 6 \bar{h}_0 \cos \theta \frac{\partial \bar{p}_0}{\partial \theta} + 3 \bar{h}_0 \frac{\partial \bar{p}_0}{\partial \theta} \frac{\partial \bar{p}_1}{\partial \theta} \]  
(12)
\[ + \frac{1}{4} \left( \frac{D}{L} \right)^2 \bar{h}_0 \frac{\partial^2 \bar{p}_1}{\partial \bar{z}^2} + \frac{3}{4} \left( \frac{D}{L} \right)^2 \bar{h}_0 \cos \theta \frac{\partial^2 \bar{p}_0}{\partial \bar{z}^2} \]
\[ + 6 \sin \theta - 12 i \lambda \cos \theta = 0 \]

Knowing the steady state pressure distribution, Eqs. 12 and 13 are solved numerically satisfying the boundary conditions.

7. Stiffness and Damping Coefficients

It is found that the fluid film, which supports the rotor, is equivalent to a spring and a dashpot system. Since the journal executes small harmonic oscillations about its steady state position, the dynamic load-carrying capacity can be expressed as a spring and a viscous damping force. This results in four stiffness and damping coefficients as given below:

\[ \bar{K}_{rr} = -\Re \left( \int_0^1 \int_0^{2\pi} \bar{p}_1 \cos \theta \, d\theta \, d\bar{z} \right) \]
\[ \bar{K}_{\Phi r} = -\Re \left( \int_0^1 \int_0^{2\pi} \bar{p}_1 \sin \theta \, d\theta \, d\bar{z} \right) \]
\[ \bar{D}_{rr} = -\Im \left( \frac{\int_0^1 \int_0^{2\pi} \bar{p}_1 \cos \theta \, d\theta \, d\bar{z}}{\lambda} \right) \]
\[ \bar{D}_{\Phi r} = -\Im \left( \frac{\int_0^1 \int_0^{2\pi} \bar{p}_1 \sin \theta \, d\theta \, d\bar{z}}{\lambda} \right) \]

where

\[ \bar{K}_{ij} = \frac{K_{ij} C}{L D P_s} \text{ and } \bar{D}_{ij} = \frac{D_{ij} C \omega}{L D P_s} \]

Considering the dynamic displacement of the journal centre along \(\Phi\) direction, the following expressions result:

\[ \bar{K}_{\Phi \Phi} = -\Re \left( \int_0^1 \int_0^{2\pi} \bar{p}_2 \sin \theta \, d\theta \, d\bar{z} \right) \]
\[ \bar{K}_{\Phi r} = -\Re \left( \int_0^1 \int_0^{2\pi} \bar{p}_2 \cos \theta \, d\theta \, d\bar{z} \right) \]
\[ \bar{D}_{\Phi \Phi} = -\Im \left( \frac{\int_0^1 \int_0^{2\pi} \bar{p}_2 \sin \theta \, d\theta \, d\bar{z}}{\lambda} \right) \]
\[ \bar{D}_{\Phi r} = -\Im \left( \frac{\int_0^1 \int_0^{2\pi} \bar{p}_2 \cos \theta \, d\theta \, d\bar{z}}{\lambda} \right) \]

The stiffness and damping coefficients thus obtained can be used to study the stability of a rigid rotor supported on this bearing.
8. Stability Analysis

The stability of the journal is analyzed by combining the equations of motion and the film forces $W_r$ and $W_{\phi r}$ assuming the rotor to be rigid and having a mass $M$. The equations of motion in non-dimensional form are written as:

\begin{align}
\bar{M} &= \frac{1}{\lambda^2(D_{\phi r} + D_{rr})} \left[ (\bar{D}_{rr}K_{\phi \phi} + \bar{K}_{rr}D_{\phi r}) - (\bar{K}_{\phi r}D_{rr} + \bar{D}_{\phi r}K_{rr}) + \frac{W}{\epsilon_0} (\bar{D}_{rr} \cos \phi_0 - \bar{D}_{\phi r} \sin \phi_0) \right] \\
\bar{M}^2 \lambda^4 - \lambda^2 \left[ \bar{M} \left( \frac{W \cos \phi_0}{\epsilon_0} + \bar{K}_{\phi r} + \bar{K}_{rr} \right) (\bar{D}_{rr} \bar{D}_{\phi r} - \bar{D}_{\phi r} \bar{D}_{rr}) \right] + \left( \bar{K}_{rr} \bar{K}_{\phi r} - \bar{K}_{\phi r} \bar{K}_{rr} \right) + \frac{W}{\epsilon_0} (\bar{K}_{rr} \cos \phi_0 - \bar{K}_{\phi r} \sin \phi_0) &= 0 \tag{16}
\end{align}

\begin{align}
\bar{M}^2 \lambda^4 - \lambda^2 \left[ \bar{M} \left( \frac{W \cos \phi_0}{\epsilon_0} + \bar{K}_{\phi r} + \bar{K}_{rr} \right) (\bar{D}_{rr} \bar{D}_{\phi r} - \bar{D}_{\phi r} \bar{D}_{rr}) \right] + \left( \bar{K}_{rr} \bar{K}_{\phi r} - \bar{K}_{\phi r} \bar{K}_{rr} \right) + \frac{W}{\epsilon_0} (\bar{K}_{rr} \cos \phi_0 - \bar{K}_{\phi r} \sin \phi_0) &= 0 \tag{17}
\end{align}

Eqs. 16 and 17 are linear algebraic equations in $\bar{M}$ and $\lambda$ and the solution of these will give $\bar{M}$ and $\lambda$. The speed of the journal calculated from this value of $\bar{M}$ is the threshold speed, above which the bearing system will be unstable. The whirl speed (speed of the journal centre) can be found from the whirl ratio for the journal speed. It must be noted that the journal centre rotates in the same direction as that of the journal.

9. Results and Discussion

The computer code is validated by comparing the dynamic characteristics namely, direct and cross stiffness and damping coefficient for a plain journal bearing with those obtained by Stachowiak and Batchelor\cite{10} (Figs. 2 and 3). Influence of ramp and L/D ratio on dynamic performance characteristics of a tri-taper journal bearing is studied. The computed values of direct stiffness, cross stiffness and damping co-efficient are plotted with eccentricity ratio and presented below.

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Fig. 2 Comparison of non-dimensional direct stiffness coefficient.

Fig. 3 Comparison of non-dimensional direct damping coefficient.

Fig. 4 Comparison of non-dimensional direct stiffness coefficient, $K_r$, for a tri-taper bearing with different L/D ratios.
From Fig. 5 it is observed that the range of values of non-dimensional cross stiffness coefficient, $K_{rr}$, increases with increase in L/D ratio. For bearing with L/D = 1.0, or 1.225 and ramp sizes 0 and 0.25, an increase in eccentricity ratio decreases the value of $K_{rr}$. However, for a bearing with $R_{amp} = 0.58$, increase in eccentricity ratio increases the value of $K_{rr}$, peaks at eccentricity ratio of 0.6, and decreases with further increase in eccentricity ratio.

From Fig. 6 it is observed that the range of values of non-dimensional direct damping coefficient, $D_{rr}$, decreases with increase in L/D ratio. At lower eccentricity ratios ($\epsilon < 0.5$), bearing with $R_{amp} = 0.58$ has lesser value of $D_{rr}$ than bearings with ramp size 0 and 0.25. However at higher eccentricity ratios, bearing with $R_{amp} 0.58$ has higher value of $D_{rr}$. For L/D = 1.0 or 1.225 and bearings with ramp sizes of 0 and 0.25, an increase in eccentricity ratio results in increase in the value of $D_{rr}$. For a bearing with $R_{amp} = 0.58$, the $D_{rr}$ value at eccentricity ratio less than 0.6 is considerably lower compared to that for a bearing of ramp size 0 and 0.25.

From Fig. 6 it is observed that the range of values of non-dimensional direct damping co-efficient, $D_{rr}$, for a tri-taper bearing with different L/D ratios.

From Fig. 7 it is seen that the range of values of non-dimensional direct damping co-efficient $D_{eq}$, increases with increase in L/D ratio. For L/D = 1.0, or 1.225 as the eccentricity ratio increases, the value of $C_{rr}$ decreases for bearings with ramp size of 0 and 0.25. For bearing with $R_{amp} = 0.58$, increase in eccentricity ratio results in increase in $D_{eq}$, attains a maximum value at eccentricity ratio of 0.6; and further increase in eccentricity decreases the value of $C_{rr}$. From Fig. 8 it is observed that the range of values of non-dimensional cross damping co-efficient $D_{rr}$, increases with increase in L/D ratio. For a bearing with L/D = 1.0 or 1.225 and $R_{amp} = 0$ and 0.25, increase in eccentricity ratio decreases the value of $D_{eq}$. For a bearing with $R_{amp} = 0.58$, an increase in eccentricity ratio increases the value of $D_{eq}$, peaks at an eccentricity ratio of 0.6, and further increase in eccentricity ratio decreases the value of $D_{eq}$. For lower eccentricity ratio ($\epsilon < 0.5$), the value of $D_{eq}$ is lower for bearings with higher ramp sizes; while for higher eccentricity ratios ($\epsilon < 0.5$), $R_{amp} = 0.58$ shows higher value of $D_{eq}$ than bearings with ramp sizes of 0 and 0.25.

Fig. 5 Comparison of non-dimensional cross stiffness coefficient, $K_{rr}$, for a tri-taper bearing with different L/D ratios.

Fig. 6 Comparison of non-dimensional direct damping co-efficient, $D_{rr}$, for a tri-taper bearing with different L/D ratios.
10. Stability

Clearance plays an important role in deciding bearing stability. Fig. 9 shows the effect of ramp size on critical mass parameter and Fig. 10 depicts its effect on whirl frequency ratio for a bearing with L/D = 1.0. The bearing with $R_{\text{amp}} = 0.58$ offers better stability than $R_{\text{amp}} = 0.25$. Fig. 9 depicts that increase in the ramp size increases the critical mass parameter, resulting in an improvement in the stability margin.

11. Conclusions

It is the dynamic load characteristics of the bearings of a rotor-bearing system that help determine the critical speeds and the threshold of instability of the rotor-bearing system.

At higher eccentricity ratios, the ramp has least influence on values of $K_{\text{bb}}$ and $K_{\varphi\varphi}$. Dimensionless $D_{\text{rr}}$ is higher in the case of circular bearings and dimensionless $D_{\text{ro}}$ is greater for a bearing with $R_{\text{amp}} = 0.58$. Dimensionless $D_{\varphi\phi}$ and $D_{\varphi r}$ are greater for circular bearings at lower eccentricity ratios, however, a bearing with $R_{\text{amp}} = 0.58$ has greater $D_{\varphi\phi}$ and $D_{\varphi r}$ at higher eccentricity ratios. Fig. 9 depicts that increase in the ramp size increases the critical mass parameter, resulting in an improvement in the stability margin. Variation in film thickness significantly affects the performance characteristics of the bearing. The bearing with $R_{\text{amp}} = 0.58$ offers better stability than $R_{\text{amp}} = 0.25$. 
References


